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Benefits of Replacing Residential Central Air Conditioning Systems

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OPERATED BY
MARTIN MARIETTA ENERGY SYSTEMS, INC.
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ENERGY DIVISION

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ABBREVIATIONS AND ACRONYMS

ARI	Air-Conditioning and Refrigeration Institute
Btu	British thermal unit
COP	coefficient of performance
DOE	U.S. Department of Energy
EER	energy efficiency ratio
EIR	electric input ratio
EMS	energy management system
EPA	U.S. Environmental Protection Agency
HIR	heating input ratio
HVAC	heating, ventilating, and air conditioning
NECPA	National Energy Conservation Policy Act
ORNL	Oak Ridge National Laboratory
RCS	Residential Conservation Service
SEER	seasonal energy efficiency ratio
SHR	sensible heat ratio
TECH	Tennessee Energy Conservation in Housing
TMY	typical meteorological year

ABSTRACT

The energy efficiency ratios (EERs) of marketed residential air conditioning equipment have increased during recent years. This investigation examined the benefits of replacing a unit having an EER of 6 with a unit having an EER of 10 in a prototypical two-story house located in each of 32 cities in the United States. The U.S. Department of Energy building simulation model, **DOE-2.1A**, was used to predict the energy savings associated with this action. The reasonableness of the model for this study was confirmed by comparing the DOE-2.1A predicted energy use data with measured energy use data for the Tennessee Energy Conservation in Housing (TECH) control house in Knoxville, Tennessee, and four specially metered houses in Little Rock, Arkansas.

It was predicted that the seasonal energy efficiency ratios (SEERs) of correctly sized units would vary from 0.6 of the rated EERs in the northern part of the country, to 0.8 of the rated EERs in the middle part of the country, and to about the rated EERs in the lower southern part of the country. Oversized units were predicted to have lower SEERs. These values reflect the total annual energy consumed by the air conditioning units, including the energy used by the crankcase heaters. From these results, a simplified Residential Conservation Service Model Audit procedure for estimating SEERs from EERs was developed.

Using 1982 capital and electrical energy costs, simple payback periods were calculated to be as low as 5 years in the lower south regions to about 10 to 15 years in the upper south regions. If the air conditioning unit needs replacement, the simple payback period for the incremental cost of installing a high-efficiency unit was calculated to be about 2 to 5 years in these regions. Further savings would be realized if existing oversized units were replaced with properly sized, high-efficiency units.

1. INTRODUCTION

In response to the series of energy emergencies occurring in the 1970s, Congress enacted the National Energy Conservation Policy Act (NECPA) of November 9, 1978. One part of NECPA mandated the establishment of the Residential Conservation Service (RCS). RCS was to promote energy conservation by encouraging installation of conservation and renewable resource measures in residential buildings.

RCS requires large gas and electric utilities (annual sales of at least 10 billion ft^3 of natural gas or 750 million kWh of electricity) to implement the service. Such "covered utilities" must advertise the service to their customers, offer to supply an on-site audit for a fee not to exceed $15, provide a list of energy-saving practices and measures to the homeowner based on audit results, and offer to help arrange for loans and contractors.

The U.S. Department of Energy (DOE) issued the original RCS regulations, in response to NECPA, in the *Federal Register* of November 7, 1979.[1] Amended RCS regulations were issued in the *Federal Register* of June 25, 1982,[2] in response to the Energy Security Act of June 30, 1980.

The RCS regulations set forth 17 energy conservation measures, and one of these is "replacement central air conditioner." The measure is defined by DOE in the *Federal Register* as follows:

> **Replacement Central Air Conditioner.** The term "replacement central air conditioner" means a central air conditioner which replaces an existing central air conditioner of the same fuel type and which reduces the amount of fuel consumed due to an increase in efficiency.

1.1 Background

The ever-increasing cost of energy has had a marked effect on the efficiency of air conditioning equipment that is marketed. Demand for higher efficiency equipment is illustrated in Fig. 1.1 by industry shipments of residential units having increasing energy efficiency ratios (EERs)* during recent years. EERs increased from an average of about 7 in 1977 to an average of 8–9 in 1982. Prior to 1977, an average EER of about 6 or under was predominant. Today, there are air conditioning units on the market having EERs in excess of 12; however, these are limited. There are a number of units available having EERs of 10 or greater. Therefore, the evaluation of replacing an air conditioning unit with an EER of 6 with a newer model with an EER of 10 is appropriate.

When replacement of an existing air conditioning unit is considered, there are many different scenarios. The extreme scenarios are envisioned to be (1) the existing unit is economically beyond repair and must be replaced in any case and (2) the existing unit continues to operate with a history of few or no failures but has a low EER rating (<6).

*EER is a common figure of merit used by industry, which is defined as Btu/h of heat extracted per watt of energy input to the air conditioning unit.

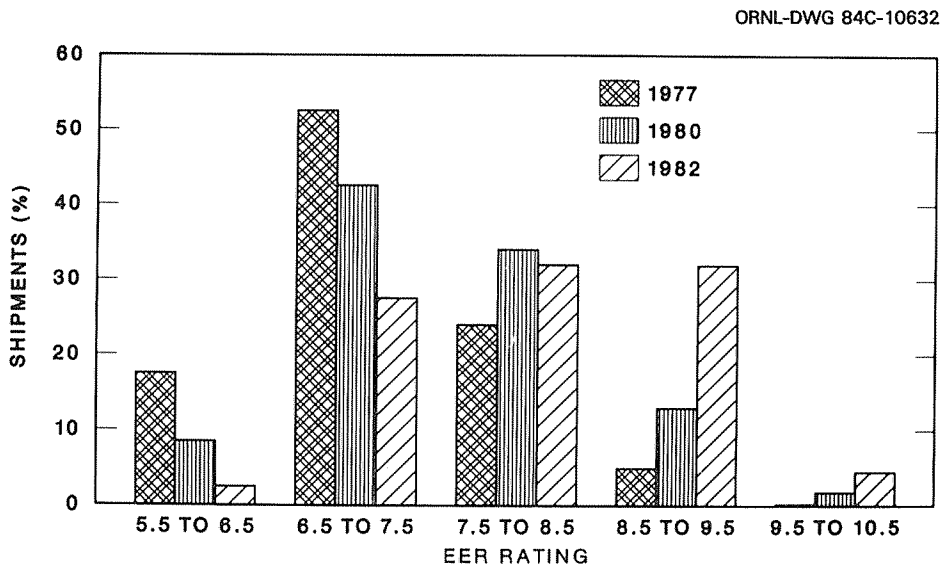

Fig. 1.1. Industry shipments of single-package and split air conditioning units under 65,000 Btu/h showing EER changing with time (year). Source: Richard Denny, Air-Conditioning and Refrigeration Institute, Arlington, Va., personal communication to David Goldenberg, March 1983.

These two scenarios are addressed in this study. Between these extremes, there can be scenarios such as the existing unit has a limited life and needs major repair—for example, a compressor replacement. The results of this study could be interpolated for this in-between scenario.

In the scenario of replacing a unit that is economically beyond repair, an energy audit can be helpful in defining the proper size unit to be purchased and to encourage the owner to pay the difference for the higher efficiency unit. Proper size considerations are especially significant to prevent energy-wasteful oversizing if other energy conservation measures are exercised. Selecting the proper size air conditioning unit (not oversized) financially benefits the homeowner in two ways: reduced initial investment cost and lower operating costs.

Many existing air conditioning units have been operating more than 5 years with a history of little or no maintenance. If these units have low EERs, they can be targets for replacement regardless of being service free. In this case, a homeowner could benefit by having a properly sized, high-efficiency unit installed. An added incentive would be the longer expected life of the replacement unit.

A typical costly repair to an air conditioning unit is the replacement of a burned-out compressor. This type of failure often requires cleanout of the entire refrigerant circuit and possible replacement of other components such as capillary tubes. Although this repair cost is less than a total replacement cost, the homeowner faced with this problem should investigate paying the difference for a properly sized, high-efficiency unit. Again, there would be the additional benefit of the longer expected life for the replacement unit.

1.2 Purpose

The primary purpose of this study was to evaluate the economic benefits of replacing an existing residential air conditioning unit having a relatively low efficiency rating with a new high-efficiency unit. A secondary purpose was to evaluate the impact of air conditioning unit oversizing on efficiency and operating costs. The results of this study provided the data necessary to improve the procedures and data base of the RCS *Model Audit Manual*.[3]

1.3 Scope

The scope of this study is the evaluation of costs and benefits of replacing an existing air conditioning system having the assumed predominant EER of 6 for a system that is older than five years with a high-efficiency unit readily available today (EER = 10 assumed).* An evaluation was made for a single prototypical house in each of 32 cities representing different geographical (weather) areas using average electrical energy costs for the state in which each city is located. The evaluation was done for the two scenarios that (1) the existing unit is fully operational needing no repairs and (2) the existing unit is economically beyond repair.

This study was limited to electric drive, vapor compression, air-to-air central air conditioning systems designed primarily for residential application. These systems include all single-package and split systems rated under 65,000-Btu/h capacity. The majority of these manufactured units are listed in semiannual directories published by the Air-Conditioning and Refrigeration Institute (ARI), which include capacity ratings and efficiencies [EER or seasonal energy efficiency ratio (SEER)†][4].

Economic benefits refer to benefits to the homeowner. No attempt was made to evaluate benefits to the utility, such as reduction of the peak load.

1.4 Approach

The approach to evaluating the costs and benefits of replacing existing residential air conditioners with high-efficiency units was to (1) select and verify an existing dynamic predictive building energy system model, (2) select a prototypical house, and (3) evaluate the performance benefits of comparative air conditioning units in selected distinct weather zones. The installed replacement air conditioner costs were then used to evaluate the simple payback for each zone.

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*Although air conditioning units with EER = 10 are readily available, this may be an unacceptably high value in some extremely high humidity regions (see Sect. 2.5).

†Starting in 1980, air conditioning units are rated by the industry on the basis of the seasonal energy efficiency ratio (SEER), which is defined differently from that used in this study. The industry definition of the SEER does not include energy used for the crankcase heaters and is based on dry-coil cycling tests. The differences are discussed further in Appendix C. With the current industry method of rating, the SEER generally does not vary from the EER by more than about 0.5, except for units with multispeed compressors.

The computer model selected was the DOE-2.1A building energy use program because of its relatively sophisticated analysis procedure and the ability to easily vary input data to simulate conditions of specific interest.[5] The DOE-2.1A simulation program evaluates the transient building heat loads and air conditioning system characteristics using hourly meteorological data. The program has the flexibility of simulating changes in cooling load requirements representing differences in occupant life-style (e.g., thermostat settings, internal load, window shading, and window/door openings).

To demonstrate the reasonableness of the use of the DOE-2.1A model for this study, it was necessary to verify its accuracy for simulation of the residential air conditioning application. This was accomplished by comparing the model prediction to measured field data. Two sources of measured field data were used: (1) the control house from the Tennessee Energy Conservation in Housing (TECH) complex[6,7] and (2) selected houses from the DOE Little Rock Electric Energy Systems Load Management Demonstration program.[8]

The test of the simulation model on the TECH control house was performed because of the availability of detailed house construction information, life-style data (actually, the TECH house was normally unoccupied with simulated internal loads), and on-site meteorological data. Another important advantage of the TECH house was its use of a variety of heating and cooling systems. It was thus possible to precisely identify a normally unknown variable—heating system efficiency—during the first winter's operation when the house was heated entirely by electric resistance elements. The model was first verified by comparing the predictions to measured data obtained during the 1977–1978 winter operation of the TECH control house heated by electric resistance heat. This was followed by a verification in both cooling and heating modes during the following year using a known and well-maintained heat pump system.

The four Little Rock houses with central air conditioning systems were selected to obtain repeatability, check another location, and use different house configurations and life-styles. Although the Little Rock data were not always complete and airport meteorological data were used, it was found that the unknowns (including life-style) could be evaluated from the preliminary results, and rational corrections could be made to obtain agreement with the measured data. The results of the comparisons between the simulation model evaluation and the measured field data are described in Sect. 4.2.

A two-story prototypical house was then selected and evaluated using the simulation model with typical meteorological year (TMY) weather data in 32 locations in the nation representing geographically significant weather conditions. The results of the energy consumption thus derived in each location with a low-efficiency and a high-efficiency air conditioning system were compared with the estimated installed cost of replacing the existing low-efficiency air conditioning unit with a high-efficiency unit, and the simple payback period was evaluated. The results of this evaluation are shown in Sect. 5.2.

2. AIR CONDITIONING UNIT DESCRIPTION

Residential air conditioning is defined here as a process of cooling and dehumidifying the conditioned areas of the home. Heat and moisture are removed from the living spaces by mechanical refrigeration. Unlike commercial air conditioning techniques, central residential air conditioning systems generally treat the entire residence as a single zone controlled by a single thermostat operating a single air conditioning unit. Thus, the complications of controlling multiple zones (individually controlled spaces) are eliminated. Generally, the cooling distribution requirements of one room relative to another do not vary greatly, and a one-time adjustment of chilled air supply dampers creates a satisfactory compromise for load distribution among the individual rooms. This temperature equalization is also aided by the tendency for internal doors in the residence to remain open. When a residence is divided into two or more controlled areas, or zones, each zone is usually supplied with a separate air conditioning unit. (This dual-zone configuration was not addressed in this study.)

Residential central air conditioning systems are manufactured in two basic configurations: (1) single-package and (2) split systems. In the single-package system, all components of the system are located in a single housing with indoor conditioned air and outdoor heat sink air individually ducted to the evaporator section and condenser section, respectively. These units are often located near an outside wall of a utility room, garage, basement, or the like, so that minimal ducting, if any, of outside air is required. The condenser (outdoor air) fan must deliver significantly more air than the the evaporator (conditioned air) fan, and it is desirable to keep the transport losses (pressure drop) in outside air ducts to a minimum.

The split air conditioning system divides the components into two distinct packages. One package is located indoors at a location convenient for space availability and for ease in connection to the conditioned air delivery ducting. This package consists of the evaporator coil, expansion device, air filter, air delivery blower, and controls. The second package is located outdoors and contains the compressor, condenser coil, and air-moving fan. The outside location provides more design flexibility from the standpoint of heat sink (air) interface, allows the design of low outside air flow pressure losses, and locates the noisiest components outdoors. The indoor and outdoor components of a split air conditioning system are interconnected by refrigerant tubing (liquid line and suction line).

To better understand the implications of air conditioning system operation, weather conditions, and occupant life-style on the value of upgrading an existing system with one of improved performance characteristics, a brief description of the basic refrigeration cycle and air conditioning performance characteristics is presented in the following subsections.

2.1 Refrigeration Cycle

The essential components of the refrigeration cycle (Fig. 2.1) are:

- compressor,
- condenser,
- evaporator, and
- expansion device.

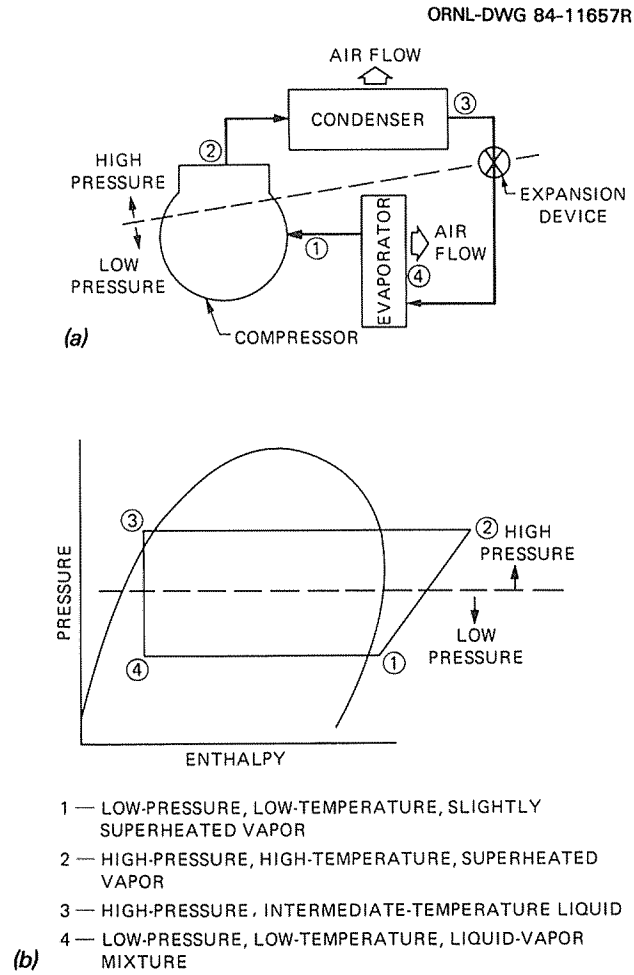

Fig. 2.1. Simple refrigeration cycle: (*a*) schematic flow diagram, (*b*) pressure-enthalpy diagram.

The refrigerant leaves the evaporator as a low-pressure, low-temperature, slightly superheated vapor (point 1) and enters the compressor where it is compressed to a high-temperature, high-pressure, superheated vapor (point 2). The hot vapor enters the condenser where it is desuperheated and condensed, at constant pressure, to a high-pressure, intermediate-temperature liquid (point 3). The heat of condensation is given up to the outdoor air. This liquid is expanded through the expansion device to a low-pressure, low-temperature, liquid-vapor mixture (point 4) before entering the evaporator. In the expansion device, the bulk temperature of the liquid refrigerant is cooled to evaporator temperature through release of the sensible cooling heat to provide latent heat of evaporation for a portion of the liquid (flash gas). In the evaporator, the remaining liquid is evaporated (by absorbing heat from the air being cooled) to the original slightly superheated vapor starting point (point 1) to start the cycle again. The conditioned space heat is thus transferred to the refrigerant in the evaporator and then rejected to the outdoor air by the condenser.

2.1.1 Compressor

The refrigeration compressor on a residential air conditioning unit is generally a reciprocating compressor represented by a vapor flow diagram as shown in Fig. 2.2 (some compressors use the rotary concept, but the principles of compression are the same). Refrigerant vapor, slightly superheated, enters the compressor through the suction valve as the piston moves in the suction stroke toward bottom dead center (point 1). As the piston begins movement toward top dead center (compression stroke), the suction valve closes and the vapor is compressed until the pressure in the cylinder is slightly in excess of the system discharge pressure, at which time the discharge valve is forced open against static spring pressure (point 2). During the remainder of the compression stroke, the high-pressure vapor is forced out of the cylinder (point 3). As the piston reverses into the suction stroke, the discharge valve closes and the residual vapor trapped in the cylinder clearance

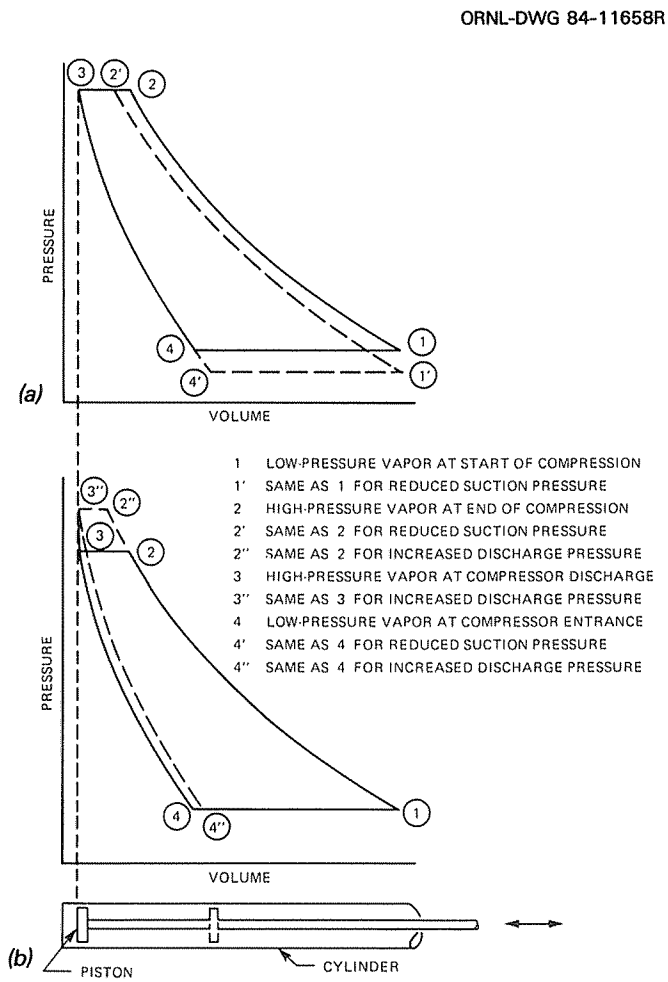

Fig. 2.2. Schematic representation of a refrigeration compressor performance: (*a*) pressure-volume diagram showing change in suction pressure, (*b*) pressure-volume diagram showing change in discharge pressure.

volume is reexpanded to a pressure slightly below the system suction pressure when the suction valve opens (point 4). As the piston proceeds toward bottom dead center, it fills with low-pressure vapor to repeat the cycle (point 1).

The capacity of the compressor (pounds refrigerant per hour pumped) is a function of the density of the refrigerant vapor (pounds per cubic foot) and the compression ratio, which are controlled by the discharge and suction operating pressures. As shown in Fig. 2.2a, if the suction pressure is reduced, suction stroke reexpansion continues past point 4 to point 4′, and the volume of vapor admitted into the cylinder is reduced by the piston-swept volume from point 4 to point 4′. In addition, the refrigerant vapor density is reduced so that the mass of refrigerant pumped is further reduced. The reverse is true at increased suction pressures.

The effect of change in discharge pressure is demonstrated in Fig. 2.2b. In this case, if the discharge pressure is increased, reexpansion of the refrigerant vapor begins at point 3″ rather than point 3, and vapor is admitted into the cylinder at point 4″ rather than point 4. Thus, the volume of refrigerant pumped is reduced as before. In this example, however, the refrigerant density remains unchanged (assuming no change in suction pressure).

The power required to drive the compressor is a function of the capacity (pounds per hour pumped) and compression ratio. Since the system EER equals the capacity (output) divided by the power (input), it is desirable to keep the capacity as high as possible and the compression ratio as low as possible. Both high capacity and low compression ratios are benefited by high suction pressure and low discharge pressure. These conditions (high suction pressure and low discharge pressure) result from high condenser and evaporator performance (low temperature difference between air and refrigerant). The compressor power consumption is especially important because it constitutes the major portion (~80%) of the total air conditioning system input power.

Air conditioning units are usually fitted with a number of devices to protect the compressor. One such device is the crankcase heater, which is a resistance heater located in the compressor sump to prevent liquid accumulation. Although the wattage of this heater is generally low, it can account for a significant portion of the annual energy use, particularly in the northern climates (Appendix C).

2.1.2 Condenser/Evaporator

Both the condenser and evaporator are refrigerant-to-air heat exchangers. (There are also a small number of units with water-cooled condensers but they are not addressed here). Typically, they consist of copper (or aluminum) tubing bent in serpentine fashion, through which the refrigerant passes. For a more effective heat transfer surface, plate or spine fins (usually aluminum) are extended between and thermally bonded to the outer surfaces of the tubes. Air is forced over the outside tube/fin surface by a fan or blower.

The overall thermal effectiveness of either the condenser or the evaporator is dependent upon the amount of heat exchanger surface area and the quantity of air moving past the surface. The ultimate design considerations include both performance and cost factors. Increased heat exchanger surface increases thermal performance at an increased

first cost (selling price). Increased air flow also increases thermal performance at an increased first cost plus increased power consumption or operating cost.

2.1.3 Expansion Device

The expansion device (see Fig. 2.1) regulates the flow of refrigerant through the system by metering liquid refrigerant to the evaporator and serves as a pressure-reducing device. The goal is to meter sufficient flow of liquid to the evaporator so that the last drop of liquid will evaporate to a vapor just before it leaves the evaporator. Thus, the maximum surface area of the evaporator is effectively used, and no liquid is returned to the compressor. Two basic types of expansion devices are used: (1) dynamic and (2) static.

The dynamic device is generally a thermostatic expansion valve. This valve senses the vapor temperature leaving the evaporator as compared to the saturation temperature (corresponding to the suction pressure) and continuously modulates the flow to maintain a constant vapor superheat temperature of about 6–10°F. If the flow rate is too low, evaporation will be complete before the refrigerant reaches the evaporator outlet and the excess heat transfer surface remaining will increase the superheat temperature, signaling the expansion valve to open. The reverse is true if the refrigerant flow is too high.

The static expansion device is generally a fixed, small-diameter capillary tube sized to reduce the pressure from condenser pressure to evaporator pressure. The primary advantage of this device over the thermostatic expansion valve is lower first cost and less maintenance. Although the capillary tube does not modulate mechanically, it has a degree of modulating capability. The metering rate of the capillary tube is dependent upon the pressure difference across it and the amount of subcooling of the high-pressure liquid entering the tube. Because the tube has a small-diameter passage, the resistance to flow is highly sensitive to high specific volume vapor, as opposed to much denser liquid. When the high-pressure liquid is subcooled to a greater degree, it takes longer to reach the flash point,* and a greater portion of the capillary tube is occupied by liquid rather than vapor, thus allowing greater flow. As the air conditioning system load changes with change in indoor and/or outdoor temperature, the changes in pressures and degree of subcooling will tend to properly adjust the flow to a satisfactory degree. For example, if the thermal load on the evaporator is reduced, the refrigerant will boil at a slower rate, and the evaporator will tend to fill with liquid. This liquid, in turn, will reduce the amount of refrigerant in the condenser and result in less subcooling, which will cause the flash point to move upstream in the capillary tube and reduce the flow rate in response to the reduced thermal load.

2.2 Air Conditioning System Performance

Most of the major air conditioning unit manufacturers in the United States are members of the Air-Conditioning and Refrigeration Institute (ARI). ARI publishes

*The flash point is located along the length of the capillary tube where the refrigerant just begins to boil (or flash).

semiannual directories that include listings of performance and efficiency ratings of the members' air conditioning products under 135,000-Btu/h capacity, which covers all residential sizes.[4] The method of listing has changed over the years. Older directories listed system capacity in Btu's per hour and power input in watts at one operating point: 95°F air delivered to the condenser (outdoor air) and 80°F dry bulb/67°F wet bulb air delivered to the evaporator (indoor air). The EER can be calculated by simply dividing the capacity (Btu's per hour) by the power (watts). Beginning in 1980, the ARI directories also list either the EER or the SEER of the air conditioning units. The SEER, as defined in these directories, is a figure of merit that is intended to represent the average operation over an entire cooling season. It is derived from a test that considers only one cycling rate (on for 6 minutes and off for 24 minutes) at one outdoor temperature (82°F) for dry conditions (no latent heat removal). Both the indoor and outdoor fans are cycled in the test, but the crankcase heater energy use is not included.[9]

In the RCS *Model Audit Manual,*[3] default SEER values as defined in this study are listed to provide the auditor with data to obtain at least a rough idea of the seasonal cooling energy use in the residence. When the SEER value of a given air conditioning unit is unknown, the audit procedure gives the auditor the choice of using either the ARI-published value[4] or a value developed from the results of this study (Sect. 5). The latter SEER value is generally lower than the ARI-published value in most regions of the country. It was developed from the results of the simulation of a prototype house using the DOE-2.1A program with that program's default relations describing the equipment performance characteristics at off-design conditions. The default relations assume that the crankcase heater operates when the unit is off and the outdoor temperature is below 65°F. Also, the default relation for performance degradation is somewhat different from the relation commonly used in industry. The effect of these assumptions is discussed in Appendix C. It was found in this study (Sect. 4.2) that these default relations are generally satisfactory for some houses having measured data. However, future work could show that the air conditioning system's characteristics are different from the program default relations. When these data become available, these simulations should be updated using the new performance data.

2.3 Air Conditioning System Sizing

When replacement of an air conditioning system is required, there is an opportunity to properly size the unit to the requirements of the residence. Existing air conditioning units are often oversized. This is due to conservative estimating procedures and the tendency to select the "next larger size" to be "safe." If, in addition, other energy-saving measures are incorporated into the residence, the air conditioning unit becomes even more oversized, as does the heating unit. This oversizing has two adverse effects related to energy use. First, the cycling losses are increased, reducing the actual operating efficiency (SEER). Second, the humidity level rises during the off-cycle periods, resulting in a tendency for the occupant to compensate for the discomfort by lowering the thermostat setting, thus increasing the cooling load.

Another consideration, which may become more important with time, is the use of "energy management systems" (EMSs) by home owners. One common type EMS unit is essentially an electronically controlled timer that cycles the compressor while the thermostat calls for cooling. The cycling period can vary but averages on the order of 5 minutes off-cycle during each 15-minute period. Evaluation of the energy-saving effectiveness of the EMS is beyond the scope of this study. However, this type EMS may further amplify the humidity problem by operating the indoor fan during compressor off-cycle, resulting in reevaporation of condensed moisture on the evaporator surface.

If the replacement air conditioning unit is properly sized—preferably slightly undersized—the cost recovery time is shortened because (1) the initial investment cost is less and (2) the actual SEER is greater.

2.4 Status of Existing Air Conditioning Systems

To properly evaluate the energy savings that could occur by replacing an air conditioning unit with a properly sized, high-efficiency unit, the house cooling load requirements and the condition of the existing air conditioning unit must be determined. The RCS *Model Audit Manual*[3] provides two methods for predicting cooling loads. The preferable method is based on the use of historical utility load data. If these data are not available or are inconsistent, an analytical heat gain procedure is provided. The use of historical load data is preferable because it incorporates unknown life-style conditions that can greatly affect the cooling load. However, the analytical procedure may also be desirable in some cases by better predicting the peak load for proper unit sizing.

A more difficult problem concerns determining the SEER of the existing air conditioning unit. It is recommended that an energy auditor be provided with a readily accessible means of obtaining ARI-rated data for the unit as a point of departure. The auditor must then estimate unit degradation, if any, on the basis of age and condition of the unit. A refrigeration unit does not degrade significantly with age by "wearing out." Poor maintenance, however, can be detrimental to both output capacity and efficiency of a unit. Obvious problems observed during inspection include dirty filters, dirt-clogged coils, severely distorted coil fins, leaky or undersized ducts or other causes of restricted air flow, and air recirculation on the condenser coil. Less obvious but equally important problems include partial loss of refrigerant charge, noncondensables (air) in the refrigerant circuit, and partial plugging of the refrigerant circuit (usually at the expansion device). The less obvious problems can often be suspected from homeowner complaints of insufficient cooling or unusually high summer utility bills. In the latter case, the auditor must be careful to consider the contribution of a basically low EER of the existing unit or unusual life-style to high electric energy consumption.

2.5 Geographical Limitations

Industry's response to increased demand for air conditioning units with higher efficiencies has sometimes resulted in some adverse effects on the overall comfort—and

therefore on the actual energy savings—resulting from design changes. One specific means of increasing air conditioning unit efficiency—increased evaporator effectiveness—has apparently reached its limit in further contribution to efficiency improvement; in many applications it has exceeded its limit.

As the evaporator effectiveness is increased (usually by increasing the heat transfer surface area), the temperature difference between conditioned space air and refrigerant is decreased, thus raising the refrigerant suction temperature and pressure. This increase in suction pressure reduces the compression ratio of the compressor, requiring less work with greater performance (total heat removal) and resulting in increased system efficiency. This same increase in suction pressure (and temperature), however, limits the minimum dew point of the conditioned air. The result is increased total heat removal and efficiency *but* a decrease in the proportion of latent heat removal; that is, the unit has a higher sensible heat ratio (SHR).

The impact of high SHR units was first felt in regions where the ambient humidity is high, such as the southeastern United States; indeed, this condition has already created some negative influence towards upgrading efficiency. The consumer reaction is discomfort, since the equilibrium humidity level in the conditioned space is uncomfortably high because of a combination of high equipment SHR and infiltration of humid air. More important, from the standpoint of overall energy consumption by the facility being air conditioned, this resultant humidity discomfort leads to occupant compensation to regain comfort by lowering the thermostat control point. Two adverse energy-related effects are thus encountered, reducing or possibly totally negating the potential benefits of the high-efficiency air conditioning unit: (1) the decreased temperature level in the conditioned space increases the heat gain through the structure envelope, thus increasing the air conditioning load; and (2) the air conditioning unit operating suction temperature (and pressure) is reduced, increasing the compressor compression ratio (and power input) and decreasing the performance (system capacity) and efficiency (EER).

Evaluation of practical limitations on EER selections by regions was beyond the scope of this study. An upper limit EER = 10 was selected for analysis purposes.

3. TEST HOUSE AND PROTOTYPICAL HOUSE CHARACTERISTICS

Five test houses with recorded air conditioning system energy use data were selected as a basis for evaluating the reasonableness of the DOE-2.1A computer simulation model for this study: the TECH control house in Knoxville, Tennessee, and four of the occupied houses in the DOE Little Rock, Arkansas, Electric Energy Systems Load Management Demonstration program. The TECH control house has been under the direct control of Oak Ridge National Laboratory (ORNL) since the conception of TECH.[6,7] The construction and operating data for the TECH control house are known in great detail, including on-site weather data, measured infiltration rates, and recorded energy use data. The data are generally accurate, but any inaccuracies due to instrumentation or other problems are readily identifiable. The TECH control house has another distinct advantage: the type of heating and cooling systems with known performance characteristics were periodically changed and always maintained; thus, the equipment data are considered reliable. Further, one heating season's operation used electric resistance heat. This heating type completely eliminated the heating system efficiency variable, which added more confidence to verification of the building load evaluation portion of the simulation model.

The four houses selected from the DOE Electric Energy Systems Load Management Demonstration program[8] for additional simulation model verification were those that had the most consistent recorded air conditioning system energy use data. Although the details of house construction and use (life-style) and air conditioning unit efficiency were not as complete as for the TECH house, they did provide a valuable history of cooling energy consumption. It was possible to estimate the efficiency (EER) of the air conditioning units with reasonable accuracy from just the unit model number and the assumption that there were no unusual operating abnormalities such as refrigerant leaks or unusually dirty filters or coils. The most significant unknown was life-style, but that being the sole variable of real concern, it was possible to predict important living habits from the monthly results of the simulated analysis as compared to actual energy consumption. Through a process of iteration and correction for life-style that became apparent from seasonal differences between calculated and actual results, the real life operation for each house was approximated. A sensitivity analysis identified the insignificance of the questionable data.

3.1 TECH Control House

The TECH control house is one of three 1800-ft^2, single-family residences (see Fig. 3.1) consisting of three bedrooms, two baths, a living room, and a kitchen.[6,7] The houses are of frame construction and have much better insulated and tighter thermal envelopes than normally found in this climate zone. Figure 3.2 shows the first- and second-floor plans for these houses. The control house is built over a crawl space and initially had a medium-efficiency heat pump. This unit had ARI ratings of 31,050 Btu/h and 2.46 coefficient of performance (COP) for heating and 30,000 Btu/h and 6.96 EER for cooling. During the first heating season (1977–1978), electric resistance heaters having a 36,000-Btu capacity instead of the heat pump were used to heat the house.

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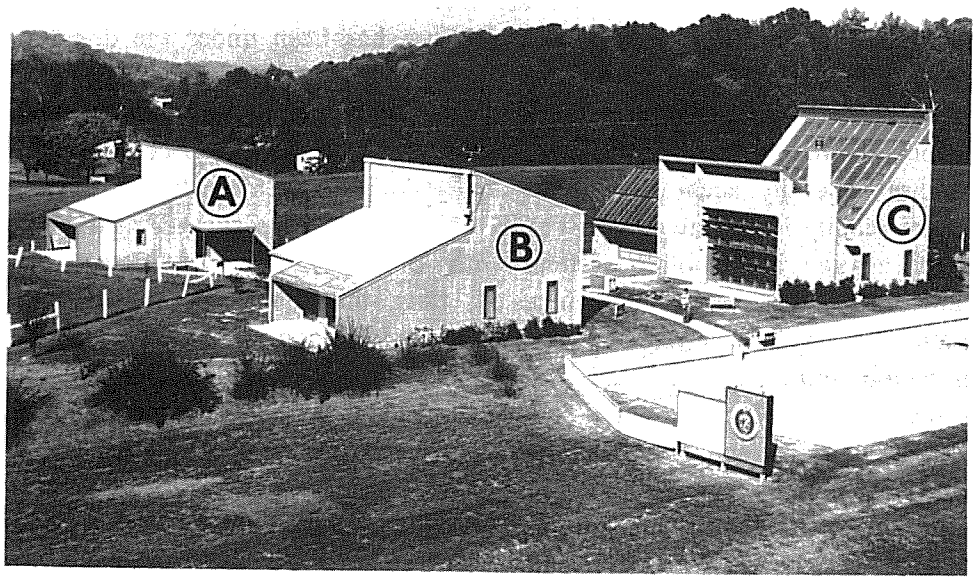

Fig. 3.1. Tennessee Energy Conservation in Housing complex in Knoxville, Tennessee: (A) control house, (B) annual cycle energy system house, (C) solar house.

3.2 Little Rock House Series

Photographs and simplified floor plans for the four houses in the Little Rock, Arkansas, area that have air conditioning energy use data are presented in Figs. 3.3 through 3.6. These data plus general construction data for these houses were obtained from Arkansas Power and Light Company.[10] All four houses are single-story dwellings and all except House No. 2 are heated by gas furnace and cooled by central conditioning units. House No. 2 is heated and cooled by a central heat pump unit.

Although fewer details were available than for the TECH control house, the information on general construction and insulation for these houses was sufficient to provide a reasonable description to the simulation model. A reasonable estimate of air conditioning unit performance was made from the unit model number. A particular area of concern for this analysis was the lack of confidence in evaluating internal loads, air infiltration, and natural ventilation, because of the limited information. On-site weather data for these houses were not available, and the surface data obtained at the Little Rock airport were assumed to represent the weather conditions.

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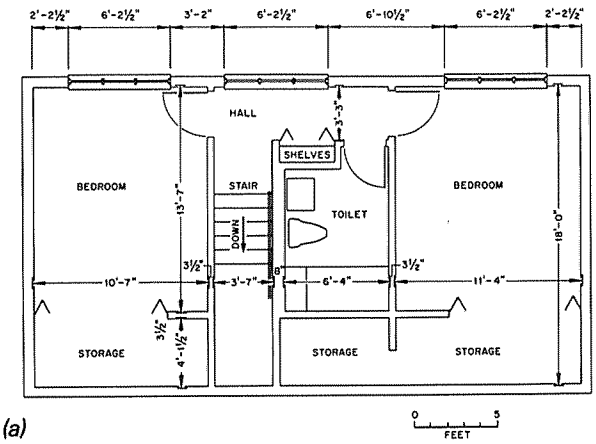

(a)

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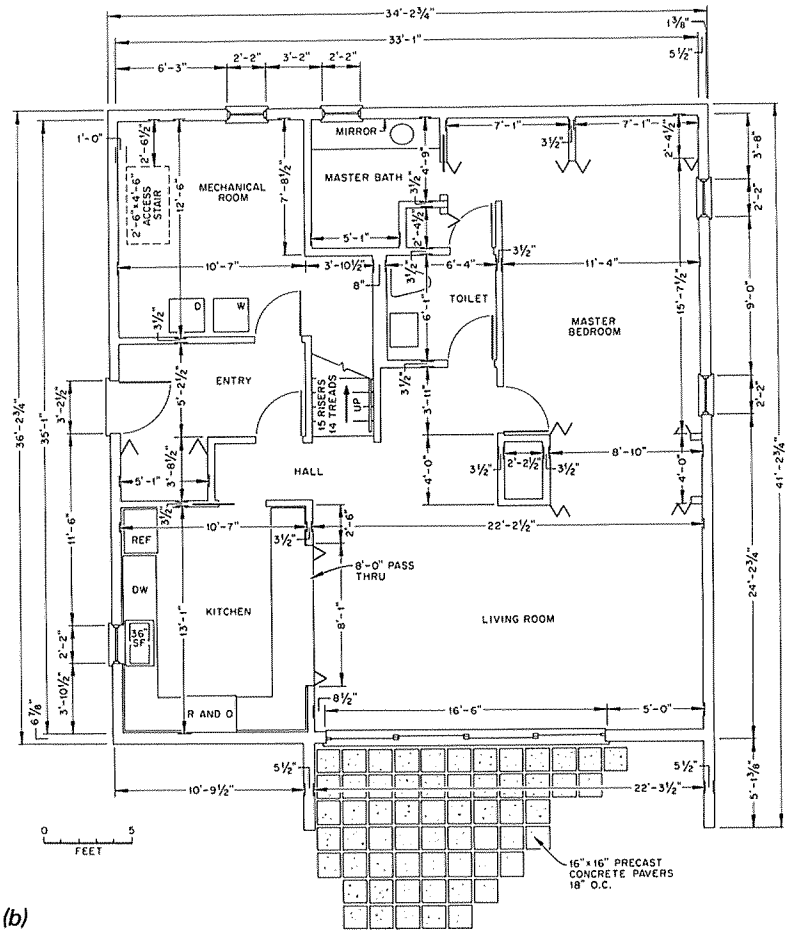


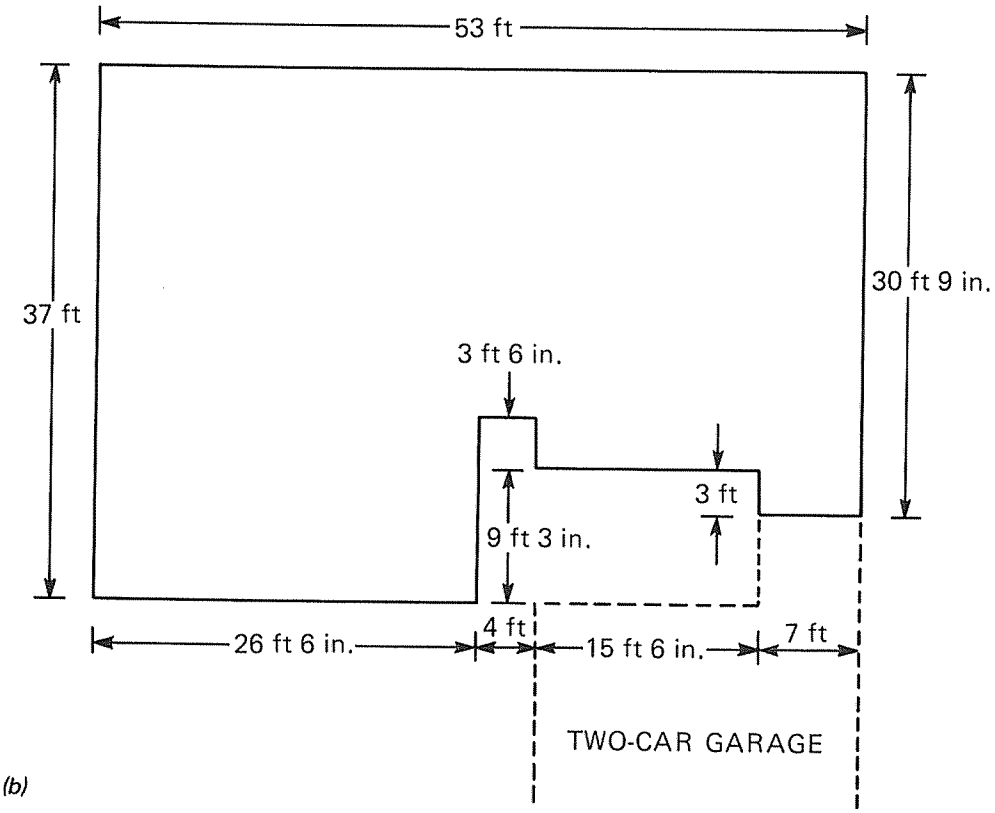
(b)

Fig. 3.2. TECH control house floor plans: (*a*) second floor, (*b*) first floor.

ORNL-PHOTO 0474-83

(a)

ORNL-DWG 84-11659



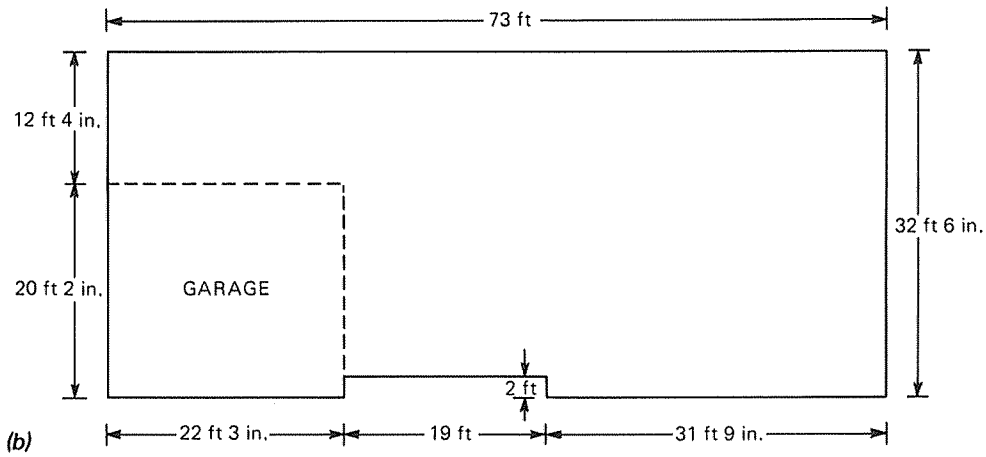
(b)

Fig. 3.3. Little Rock House No. 1: 1720-ft^2, single-story house on concrete slab; central air conditioning unit having ARI ratings of 37,000 Btu/h cooling capacity and 6.96 EER: (*a*) photo of house, (*b*) floor plan.

ORNL-PHOTO 0476-83

(a)

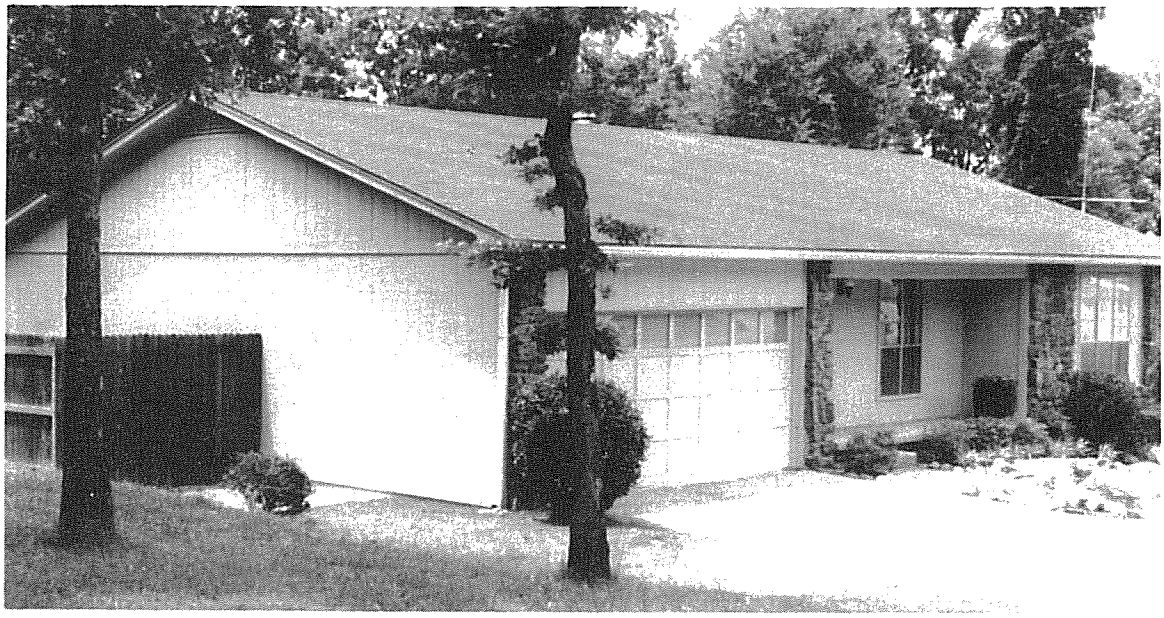
ORNL-DWG 84-11660



(b)

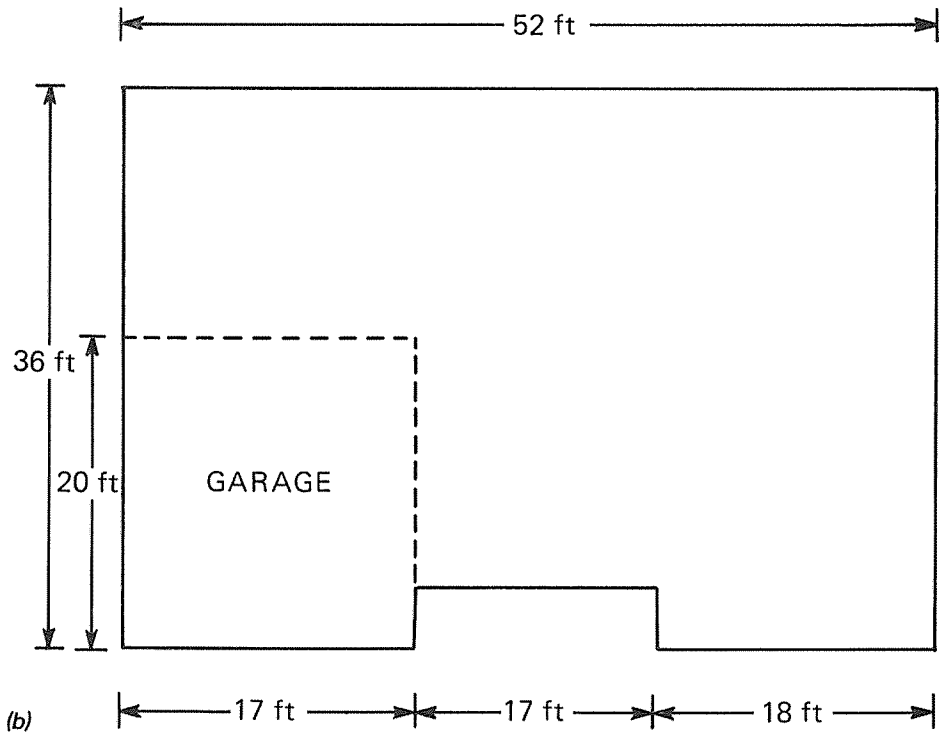
Fig. 3.4. Little Rock House No. 2: 1890-ft^2, single-story house on concrete slab; central heat pump having ARI ratings of 38,000 Btu/h and 2.50 COP for heating and 36,000 Btu/h and 6.83 EER for cooling: (*a*) photo of house, (*b*) floor plan.

ORNL-PHOTO 0477-83



(a)

ORNL-DWG 84-11661



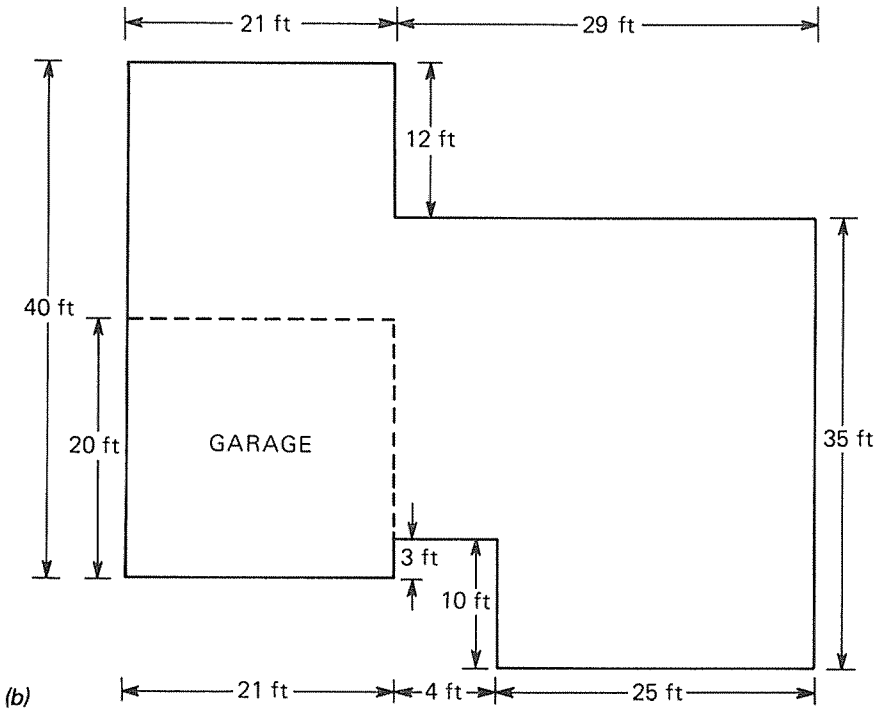
(b)

Fig. 3.5. Little Rock House No. 3: 1470-ft^2, single-story house with crawl space; central air conditioning unit having ARI ratings of 30,000 Btu/h cooling capacity and 6.38 EER: (*a*) photo of house, (*b*) floor plan.

ORNL-PHOTO 0475-83

(a)

ORNL-DWG 84-11662



(b)

Fig. 3.6. Little Rock House No. 4: 1410-ft^2, single-story house with crawl space; central air conditioning unit having ARI ratings of 30,000 Btu/h cooling capacity and 6.38 EER: (*a*) photo of house, (*b*) floor plan.

3.3 Prototypical House

For comparing the performance of air conditioning units throughout the country, the two-story house defined by Hastings[11] was selected as the prototypical house. This house is one of the three houses designed by Hastings that are typical of those now being constructed in this country. It has 1994 ft^2 of finished floor area and is assumed in this study to be built over an unconditioned basement. An elevation of the house is shown in Fig. 3.7, and the floor plan is shown in Fig. 3.8. Construction details of the thermal envelope and the material thermal resistances assumed for the study are listed in Table 3.1.

It was assumed that this house is heated by a gas furnace and cooled by a central air conditioning unit. The capacities selected for these units are dependent on the weather conditions, and they together with the operating characteristics of the units are discussed further in Sect. 5.

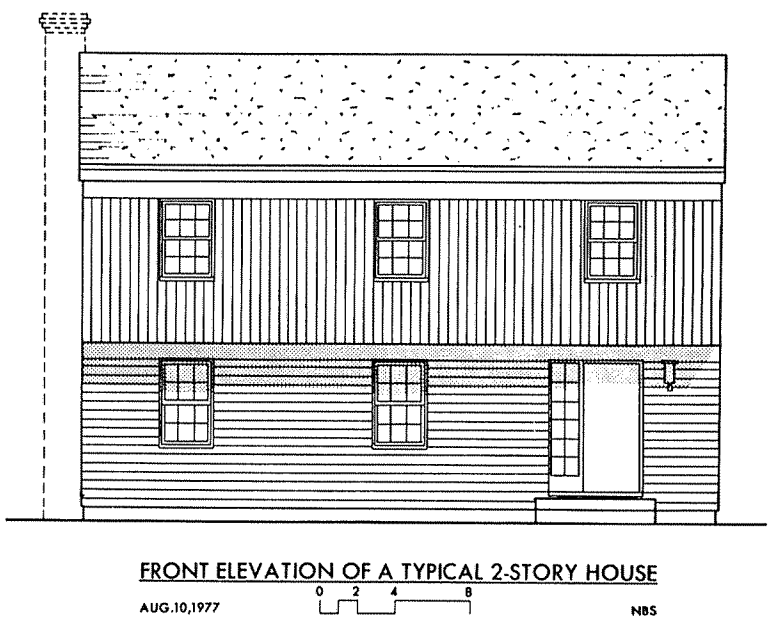


Fig. 3.7. Front elevation of the prototypical two-story house. Source: S. R. Hastings, *Three Proposed Typical House Designs for Energy Conservation Research*, NBSIR 77-1309, October 1977.

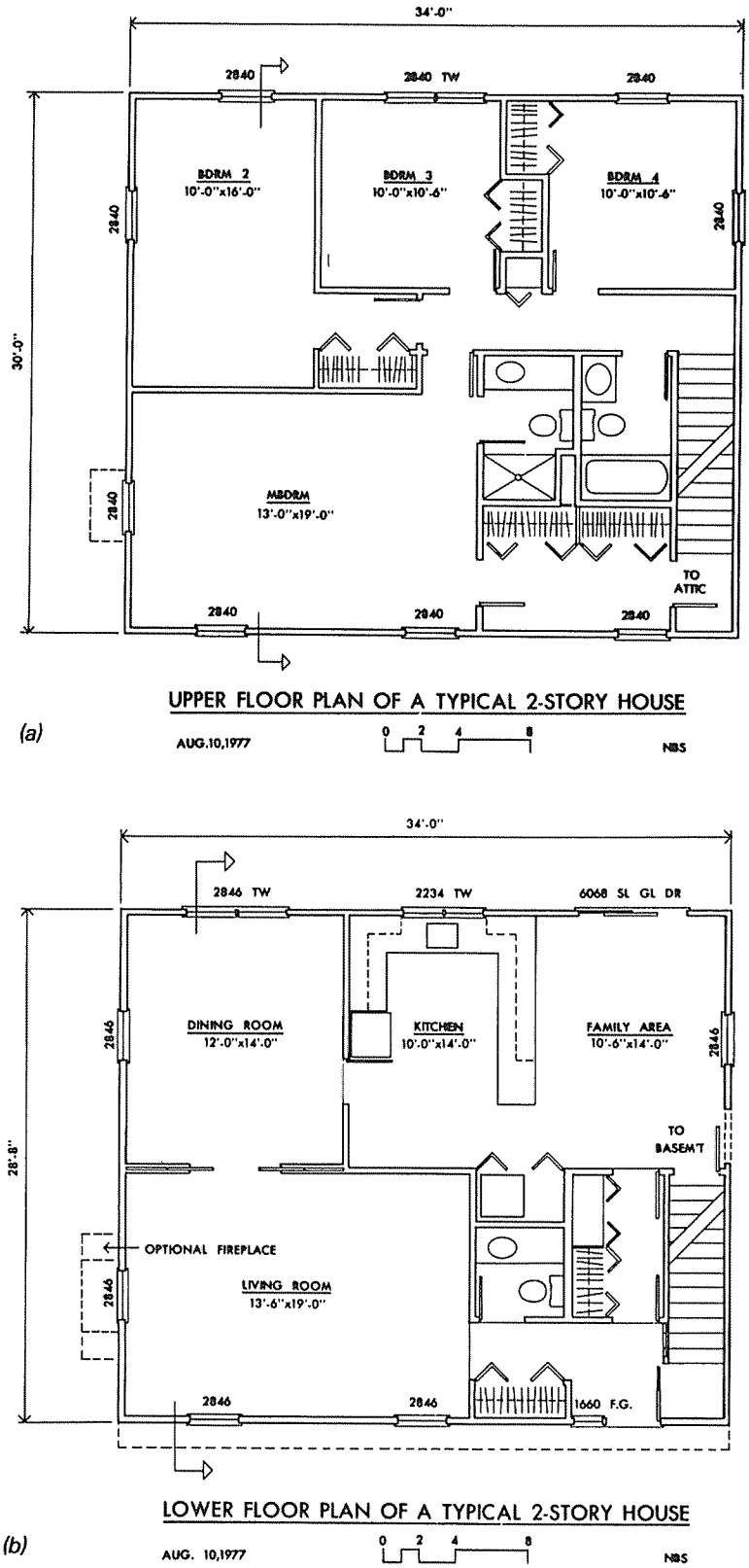


Fig. 3.8. Floor plan of the prototypical two-story house: (*a*) second floor, (*b*) first floor. Source: S. R. Hastings, *Three Proposed Typical House Designs for Energy Conservation Research*, NBSIR 77-1309, October 1977.

Table 3.1. Assumed thermal resistance of materials used in the prototypical house

Component		R-value (for thickness listed)
Basement floor (unconditioned)		
Poured concrete slab (4 in.)		0.32
Horizontal air film (still, heat flow down)		0.32
	Total	0.64
Basement walls (unconditioned)		
Concrete block		1.07
Vertical air film		0.68
	Total	1.75
First floor (above basement)		
Horizontal air film (still, heat flow up)		0.61
Plywood (3/4 in. on 2- by 10-in. joists)		1.60
Carpet		1.18
Horizontal air film (still, heat flow up)		0.61
	Total	4.00
Walls (insulation area: 75% of total, front and rear; 85% of total, sides)		
Vertical air film (15-mph wind)		0.17
Aluminum siding		0.60
Intermediate-density insulating sheathing (1/2 in.)		1.22
Cavity insulation		11.00
Gypsum wallboard (1/2 in.)		0.45
Vertical air film (still)		0.68
	Total	14.12
Walls (frame area: 25% of total, front and rear; 15% of total, sides)		
Vertical air film (15-mph wind)		0.17
Aluminum siding		0.60
Intermediate-density insulating sheathing		1.22
2- by 4-in. wood frame		4.35
Gypsum wallboard (1/2 in.)		0.45
Vertical air film		0.68
	Total	7.47
Windows		
Vertical air film (15-mph wind)		0.17
Architectural glass		0.03
Vertical air film (still)		0.68
	Total	0.88

Table 3.1 (continued)

Component		R-value (for thickness listed)
Doors (front entry)		
Vertical air film (15-mph wind)		0.17
Solid hardwood (1 3/4 in.)		1.59
Vertical air film (still)		0.68
	Total	2.44
Ceiling (insulation area: 90% of total)		
Horizontal air film (still, heat flow up)		0.61
Gypsum drywall (1/2 in.)		0.45
Insulation		19.00
Attic air space		3.10
Plywood		0.63
Shingle		0.44
Horizontal air film (15-mph wind)		0.17
	Total	24.40
Ceiling (truss cord area: 10% of total)		
Horizontal air film (still, heat flow up)		0.61
Gypsum drywall (1/2 in.)		0.45
2- by 6-in. wooden truss cords		6.87
Attic air space		3.10
Plywood		0.63
Shingle		0.44
Horizontal air film (15-mph wind)		0.17
	Total	12.27

R-value = thermal resistance level, $h \cdot ft^2 \cdot °F/Btu$

4. SIMULATION MODEL AND CONFIRMATION

The analyses of cost/benefits of replacing an existing air conditioning system with one of high efficiency and/or proper size was accomplished with the use of the DOE-2.1A simulation model.[5] This section describes this model and the verification procedures and results used to confirm that this model predicts residential air conditioning energy use with sufficient reasonableness.

4.1 DOE-2.1A Simulation Model Description

The DOE-2.1A program describes the flow of heat in a building and the associated space conditioning equipment on an hourly basis. The program uses detailed data for the building geometry and construction, for the space conditioning equipment, and for the weather to predict the energy flow in the building. Internal heat loads in the form of people, lights and equipment, as well as any infiltration and ventilation air are incorporated in the energy flow description. Heat flow through all the internal and external building surfaces is assumed to be one-dimensional. Delay of heat flow through the external surfaces having significant mass is described by the use of time series having constants called *response factors.*

The program uses a sequential approach to calculate the energy use by the heating and air conditioning equipment. It first determines the heating or cooling loads in each zone of the building, assuming that the interior temperature in each zone is fixed. The part of the program where this is done is called LOADS, and the method used in LOADS to account for the time delays for heat transfer from the zone mass to the interior air is the *weighting factor method* described in Appendix A. These fixed temperature loads are then passed on to the next part of the program, called SYSTEMS, where the actual temperatures in the zones and the amount of heat added or extracted by the HVAC system is determined. In SYSTEMS, the weighting factor method, accounting for the time delays in removing or adding heat to the zones by the HVAC equipment, is used again together with the equipment operating characteristics to predict these values. The amount of energy required to operate the HVAC system is also calculated in the SYSTEMS part of the program. The calculation procedure used in SYSTEMS is described further in Appendix A.

For residences, the DOE-2.1A program also allows for natural ventilation cooling of the residence, depending on the indoor-outdoor air enthalpy. For this case, air is assumed to be exchanged at a specified rate, independent of the wind speed and indoor-outdoor temperature difference. This option was used in the simulation of the Little Rock, Arkansas, houses, described in Sect 4.2.2.

Hourly weather data must be compressed into a format readable by the DOE-2.1A program for it to be usable. This data compression is done by a weather processor package furnished with the program.[5] This package is capable of converting weather data in a number of formats, such as TDF 1440 (airport surface observations), TRY (test reference year), and TMY (typical meteorological year), into the desired format. The compressed data can then be modified using the EDIT routine incorporated in the weather processor

package. This routine was used to incorporate on-site weather data in the simulation of the TECH control house, described in Sect. 4.2.1.

The DOE-2.1A program is designed to accept detailed input data regarding the building geometry and construction and the HVAC equipment design and performance characteristics. Many of these data do not have to be specified to use the program. In these cases, the program can draw upon its library of default data and routines to fill in the missing input data. The relations and the default values of the factors used in these relations to describe the performance characteristics of residential HVAC systems examined in this study are given in Appendix A. In this study, the total HVAC equipment capacities and circulating air flow rates at design conditions were specified, but default relations of the program were used to modify these quantities at other conditions.

4.2 Simulation Model Confirmation

Although DOE-2.1A is a highly detailed computer program, it is still a simplified model of the complex building and HVAC system behavior. It was thus felt desirable to confirm its use for this specific application of residential structures and air conditioning equipment behaviors. The lack of detailed air conditioning system performance data and, in some cases, housing construction data was especially concerning. Reliance on the default relations of the program to describe the performance of the air conditioning system had to be proven viable.

The confirmation of the reasonableness of the DOE-2.1A program for this study was done by comparing the predicted HVAC system energy use with data measured at five different houses. These houses were the TECH control house in Knoxville, Tennessee, and four occupied houses in Little Rock, Arkansas. Descriptions of these houses are given in Sect. 3, and the comparisons of the predicted and measured HVAC energy use data are given below in Sects. 4.2.1 and 4.2.2.

It will be seen that the comparisons of the HVAC system energy consumptions are within experimental error. Rational explanations can be given for any significant differences between the measured and predicted values. The best comparisons were obtained for the TECH control house, which had the most precise detailed data. The analysis of the Little Rock houses showed the importance of life-style on energy consumption.

4.2.1 Data Comparison of TECH Control House

The DOE-2.1A comparison of the ACES control house data was done for the winter of 1977–1978, the summer of 1978, and the winter of 1978–1979.[6,7] During this entire time, the house had a medium-efficiency heat pump for heating and cooling, as discussed in Sect. 3.1. However, during the winter of 1977–1978, the electric resistance heaters instead of the heat pump were used to heat the house. Since the performance characteristic of these heaters is a known constant value, the comparison for this season is for the actual energy added to the house by the HVAC system.

The construction data for this house were well defined, and the internal loads were artifically added on a precise schedule during the tests. This minimized many of the uncertainties that often are encountered in the simulation of buildings. The air infiltration rates were measured using the tracer gas decay technique, and the results were correlated as:

$$ACPH = 0.1159 + 0.048W + 0.012|\Delta T| \,, \tag{4.1}$$

where $ACPH$ is the number of air changes per hour, W is the wind speed in knots, and ΔT is the indoor-outdoor temperature difference in degrees Fahrenheit.[12]

On-site weather data, except for the cloud cover and solar radiation data, were used in this comparison. Cloud cover data measured at the Knoxville, Tennessee, McGee-Tyson Airport, located within five miles of the test site, were assumed to be valid for the simulation. Solar radiation was estimated by the program using the time of day and cloud cover data.

Results of the DOE-2.1A simulation and the measured data for the 1977–1978 winter season are presented in Fig. 4.1. It can be seen that the agreement between the predicted and measured cumulative seasonal energy use data is within 3%.The measured cumulative value is the sum of the data actually obtained during this season, but there were periods in the season when data were not obtained. The number of days of each month when data were obtained are shown in Fig. 4.1. Agreement of the average energy use rates was very good for all months, except for January. The reason for this inconsistency was not investigated in detail, but it should be noted that this was during the time when the TECH test data collection was starting, and data were obtained only six days during that month.

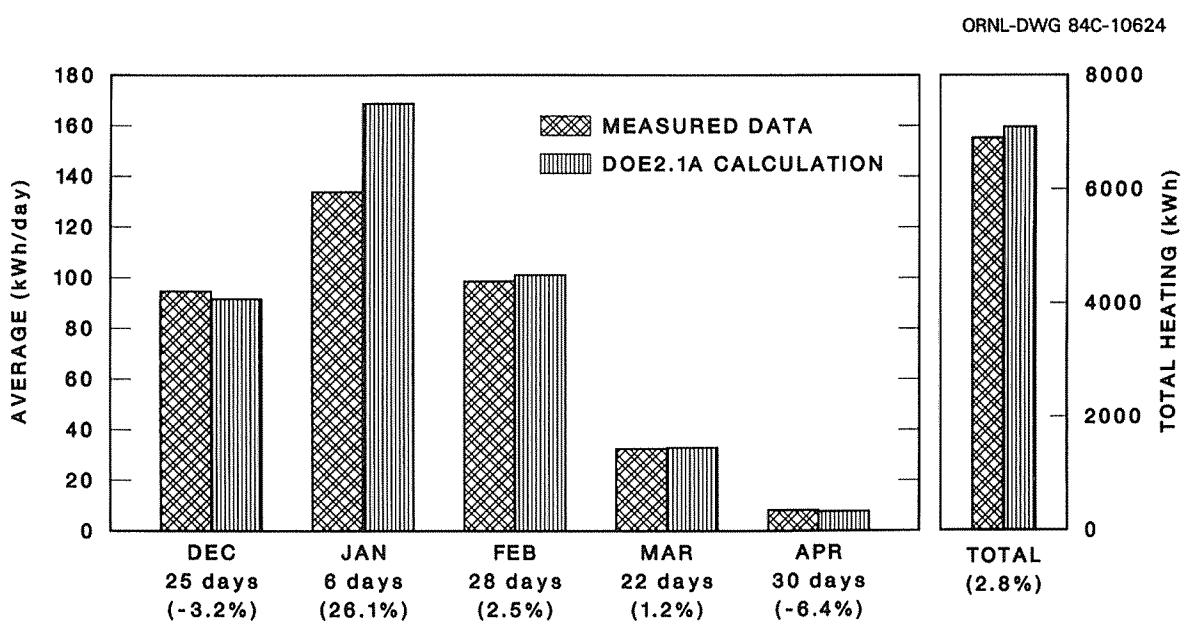


Fig. 4.1. Comparison of DOE-2.1A calculated to measured data for the TECH control house during the 1977–1978 winter heating season (electric resistance heating).

Considering this, the ability of DOE-2.1A to predict heating energy use during this season is very good.

The program was next used to predict the heat pump energy use during the summer of 1978. Results of this simulation and the measured data for this season are presented in Fig. 4.2. The number of days indicated on the plot are again to show the number of days when the data were obtained. The difference between the total calculated and the measured data for this season is about 3%. Agreement for the average HVAC system energy use rate for each month is generally good. The greatest difference (10%) was during June, which is within the accuracy required for this study. No trends were observed in these comparisons that would lead to suspect in the DOE-2.1A simulation procedure.

In the cooling season simulation, only the total cooling capacity and the EER of the heat pump at ARI-rated conditions and the indoor air circulation rate were specified. It was assumed that the DOE-2.1A program's default performance factors for this unit operating at nonrated conditions, given in Appendix A, are applicable. It might have been possible to improve the agreement between the predicted and measured values by specifying the heat pump performance factors in the analysis. But, in view of the good agreement obtained using the default factors, this effort did not appear to be warranted.

The comparison for the winter of 1978–1979 was done as a point of interest for the medium-efficiency heat pump operating during this season, since the scope of this study was limited to air conditioning. The comparison for this season is presented in Fig. 4.3, where it is shown that the cumulated predicted energy use is about 11% below the measured value. Also, the predicted values of the average monthly energy use rates are consistently lower than the measured values. Again for this simulation, only the heating capacity and the COP of the heat pump at the ARI-rated conditions and the indoor air

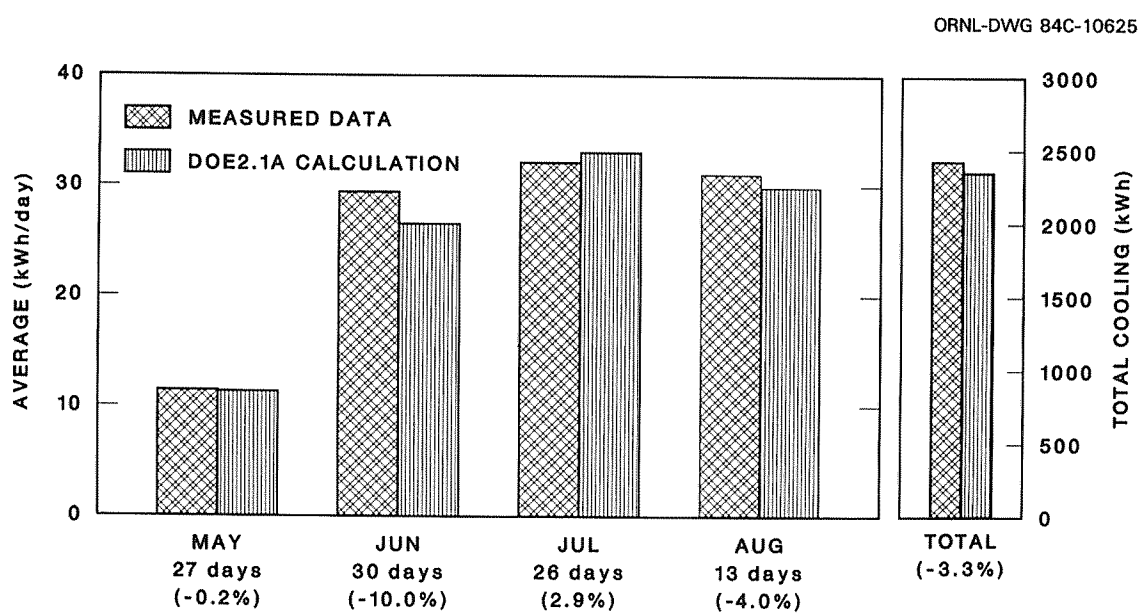


Fig. 4.2. Comparison of DOE-2.1A calculated to measured data for the TECH control house during the 1978 summer cooling season (medium-efficiency heat pump).

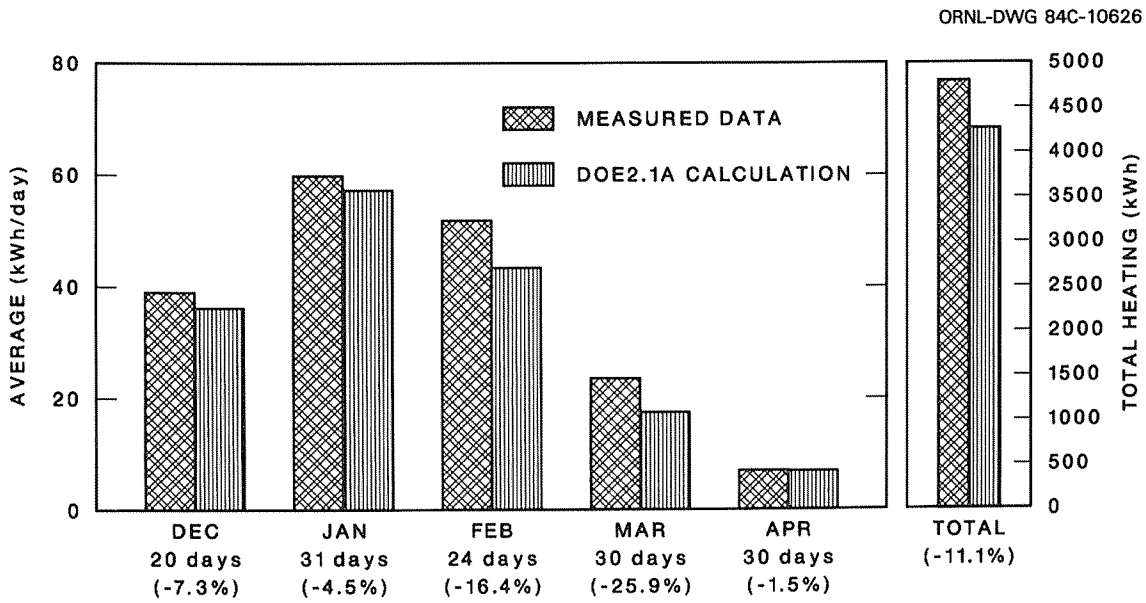


Fig. 4.3. Comparison of DOE-2.1A calculated to measured data for the TECH control house during the 1978–1979 winter heating season (medium-efficiency heat pump).

circulation rate were specified. The program's default performance factors, given in Appendix A, were assumed to be applicable here, also. In the default mode, the DOE-2.1A program assumed no degradation of the heat pump performance due to defrosting requirements. No doubt ignoring the energy requirements for heat pump defrosting explains, at least in part, the reason for the predicted energy use values being lower than the measured values. The scope of this study did not allow for further investigation of this factor.

4.2.2 Data Comparison of Little Rock, Arkansas, Test Houses

The four Little Rock, Arkansas, test houses were included in this confirmation study (1) to provide further confidence in the simulation model and (2) to determine if the unknown life styles of these occupied houses can be simulated. All of these houses had submetered data for the electrical energy consumed by their air conditioning systems during the summer of 1981. As for the TECH control house, days with missing data were excluded, and other days were also excluded because of a combination of missing data and other anomilies, such as obvious vacation periods.

Details of house construction, shading, and internal heat loads were not known with the precision of those for the TECH control house. Therefore, assumptions had to be made regarding the values for some of the parameters in the DOE-2.1A simulations. There was some iteration in comparing the predicted and measured results to estimate ventilation due to window openings and vacation periods. When the windows were closed, the rate of air infiltration into the houses was assumed to be the DOE-2.1A default relation for residences.

$$ACPH = 0.252 + 0.0251W + 0.0084\,|\Delta T|\,. \tag{4.2}$$

The weather data for the simulation were assumed to be those measured at the Little Rock, Arkansas, airport for 1981.

In the simulation of the Little Rock test houses, the total cooling capacities and the EERs for the air conditioning units (heat pump for House 2) and the indoor air circulation rates were specified. The heating capacities of the gas furnaces for Houses 1, 3, and 4 and the heating capacity and the COP of the heat pump for House 2 at the rated conditions were also specified. At nonrated conditions, it was assumed that the default performance factors, given in Appendix A, are applicable. Except for House 4, the thermostat heating and cooling set points were specified to be 70°F and 78°F, respectively. To obtain reasonable simulation of House 4 performance, these temperatures had to be set at 68°F and 70°F, respectively.

In some cases, natural ventilation by window opening was specified. This was specified only to take place at hours of the day that were consistent with a reasonable life-style and when the outdoor air enthalpy was less than the indoor air enthalpy. At these times, the air conditioning unit was assumed to be off.

Results of the simulations for House 1 are summarized in Fig. 4.4. It can be seen that the comparison of the predicted and measured values is very good here. Assuming that the windows of this house were never opened, the overall agreement is within 2.7%. Some improvement in the comparison was made by assuming that the windows were opened during the cooler days. The greatest benefit of using this assumption is in May, and the overall results agree within 1.2%. These differences show the importance of the life-style effects in the simulations. These effects had been deduced by comparison of the predicted and measured data, since there were no data on the occupant behavior.

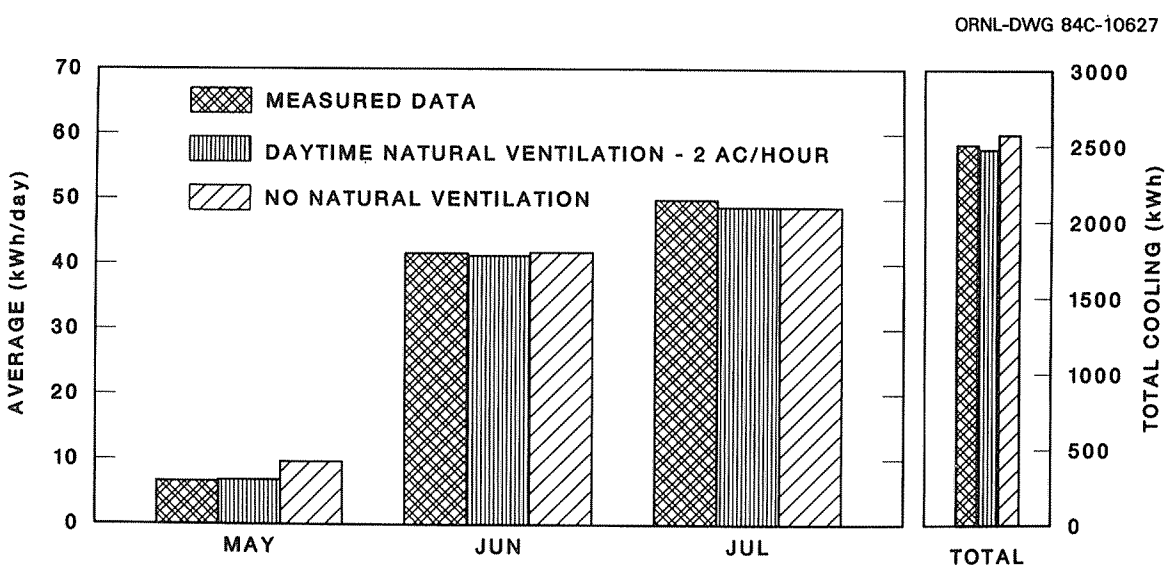


Fig. 4.4. Comparison of DOE-2.1A calculated to measured data for Little Rock House No. 1 during the 1981 summer cooling season.

The importance of life-style is further illustrated in the Little Rock House 2 comparisons, shown in Fig. 4.5. Neglecting any window opening, the predicted seasonal energy use is about 16% higher than the measured value. The greatest disagreement occured during the months demanding less air conditioning, such as September, where the values differed by 79%. (The May comparison includes some heating energy and, thus, is of limited value here.) One possible explanation for this is natural ventilation through window openings during the cooler weather periods. Assuming that the windows were opened during the cooler hours of the day reduced the seasonal error to less than 12%, and slightly better agreement was obtained for September (63%). Further agreement between the predicted and measured energy use was obtained by assuming that the shading coefficient[13] of the windows is 0.55 instead of 0.86 (by the use of drapes and blinds) in addition to the window openings. In this case, the predicted seasonal energy use is 6% lower than the measured value, and the agreement for September is about 36%. While the exact life-style scenario cannot be pinpointed by this exercise, it can be seen that the DOE-2.1A program can adjust the predicted energy use rates for these effects to the precison required in this study.

The influence of the life-style was also seen in the comparison of the data for House 3, shown in Fig. 4.6. Here, without any natural ventilation due to opening of windows or extra shading of the windows, the DOE-2.1A-predicted seasonal energy use was about 61% greater than the measured value. Including these effects resulted in much better agreement, as shown in Fig. 4.6. Assuming natural ventilation during the daylight

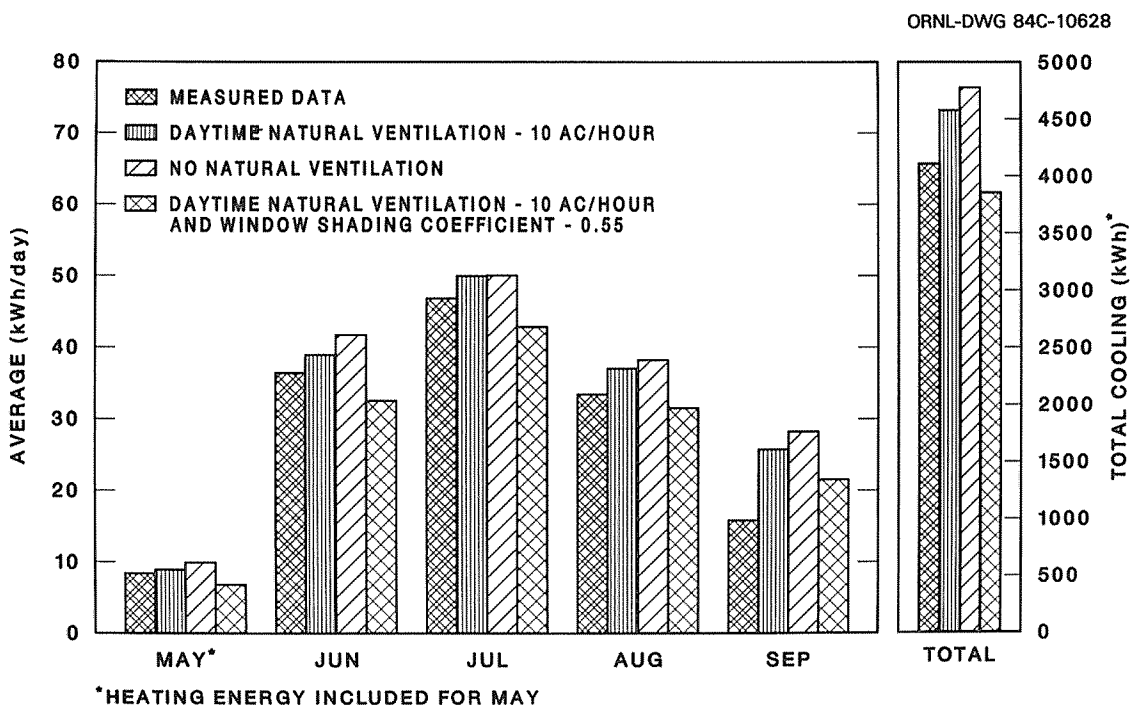


Fig. 4.5. Comparison of DOE-2.1A calculated to measured data for Little Rock House No. 2 during the 1981 summer cooling season.

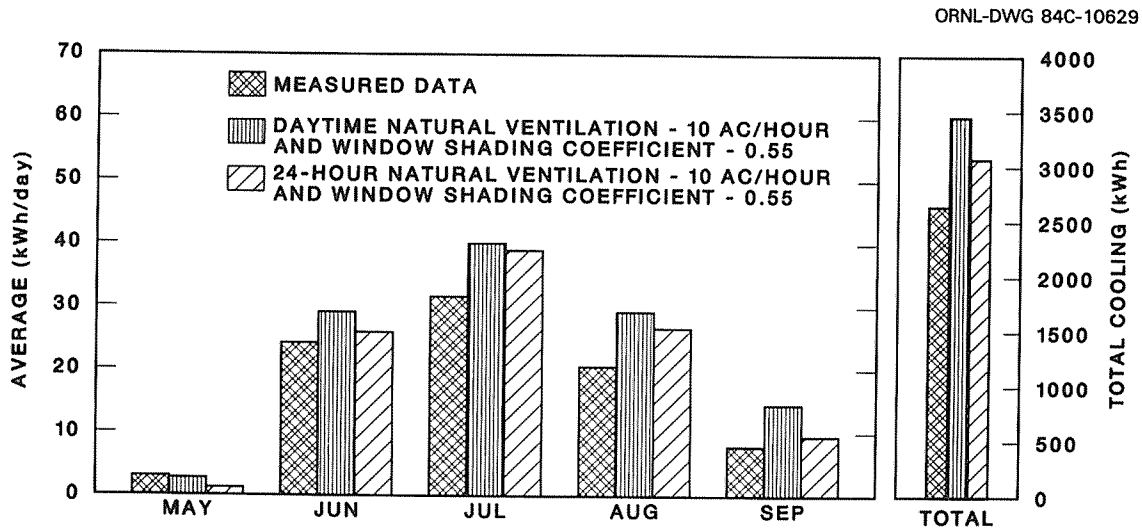


Fig. 4.6. Comparison of DOE-2.1A calculated to measured data for Little Rock House No. 3 during the 1981 summer cooling season.

hours and extra window shading reduced the error in the seasonal energy use from 61% to 31%. Assuming next that the windows were opened during the cooler hours anytime of the 24-h day further reduced the error to 16%. There were still significant (up to 30%) differences during the months of July and August, but in absence of actual life-style data, further investigation of the life-style effects was not warranted for this study.

For Little Rock House 4, the thermostat cooling set point temperature had to be reduced to 70°F to obtain good agreement between the predicted and measured air conditioning unit seasonal energy use rate. It can be seen in Fig. 4.7 that about a 7%

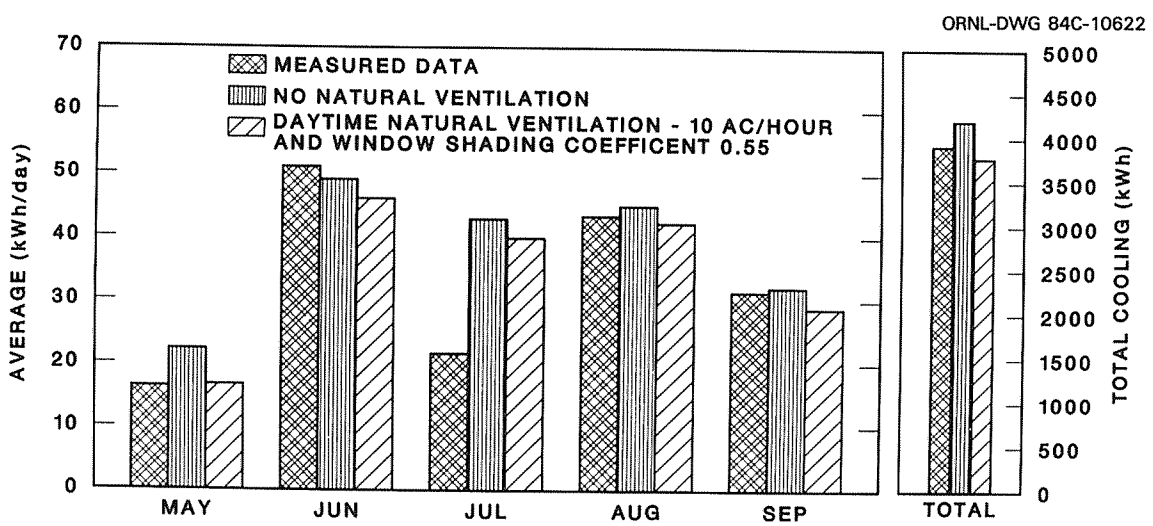


Fig. 4.7. Comparison of DOE-2.1A calculated to measured data for Little Rock House No. 4 during the 1981 summer cooling season.

agreement was obtained assuming that the windows were closed at all times and that there was limited shading of the windows. There was some improvement in the agreement of the comparison, about 3%, by assuming that the windows could be opened during the day and having additional shading of the windows. The greatest error between the predicted and measured data is for July. There was not sufficient information available to determine the cause of this. However, for this house, only four days of measured data were obtained during this month, which implies that they may be suspect.

5. ANALYSIS AND RESULTS

5.1 Methodology

The DOE-2.1A program was used to predict the air conditioning system performance for the prototypical Hastings two-story house, described in Sect. 3.3. This program was used for obtaining data on 32 cities in the continental United States, which were selected to represent distinct weather conditions in all geographical areas. It was assumed that the typical meteorological year (TMY) weather data represented the weather conditions in each of these cities. It was further assumed that the prototypical house has average internal heat loads of 1270 Btu/h due to four occupants and 1 kW (or 3412 Btu/h) due to equipment (lighting and appliances). The schedules assumed for these internal loads are given in Appendix B.

The effects of the air conditioning unit rated efficiency and oversizing were simulated. In all cases, the existing unit in the house having an EER of 6 was assumed to be replaced with a unit having an EER of 10. For both cases, seasonal energy consumptions were predicted for units properly sized (no excess capacity) for the house and for units having 25% and 50% extra capacities.

The proper size of each unit was determined by trial and error using the DOE-2.1A program. The air conditioning unit capacities and the indoor circulating flow rates were selected, and the program was run iteratively to determine the values of these variables where the indoor temperatures did not exceed 80°F for more than 1% of the cooling season. The thermostat was set at 70°F for heating and 78°F for cooling. The circulating air flow rate was specified to be 100 ft^3/min (at sea level) for each 3000 Btu/h total capacity, and the minimum temperature that the cooled air could leave the evaporator coils was assumed to be 55°F.

The thermal energy extracted from the conditioned spaces and the electrical energy consumed by the air conditioning units during the year were calculated using the DOE-2.1A program. In these simulations, the total capacities of the heating and cooling units and their efficiencies at design conditions were specified. The performance of each air conditioning unit at off-design conditions was assumed to be that predicted by the program's default performance factor relations given in Appendix A. It was assumed for this particular simulation that the windows would not be opened during the cooling season.

For each of the specified conditions, the seasonal energy efficiency ratio (SEER) was calculated from the DOE-2.1A-predicted annual energy extraction and electrical consumption values. The DOE-2.1A-predicted electrical consumption values included the crankcase heater energy.* From the energy consumption value for each of these conditions, the seasonal energy cost (based on the 1982 average residential electric costs for the state in which the city is located) was calculated. Simple payback was then determined by dividing the new air conditioning replacement cost by the first-year energy cost savings.

*The DOE-2.1A default relation for predicting performance degradation due to cycling is somewhat different from that commonly used in industry. The difference in the predicted energy savings using these two relations is about 5%, which is within the accuracy of this study. Further details are given in Appendix C.

Interest on investment, escalating energy costs, and anticipated reduced maintenance were not considered.

5.2 Cost/Benefit Evaluation Results

Results of these simulations for the replacement air conditioners are summarized in Table 5.1 for each of the 32 cities evaluated (Column 1). Column 2 identifies the percentage that the air conditioning unit is oversized, on the basis of the criteria described in Sect. 5.1. Columns 3 and 4 list the annual energy consumption required to handle the cooling load for two air conditioning unit efficiencies: an original unit with an EER = 6 and a replacement unit with an EER = 10. The seasonal cooling load for each size unit is shown in Column 5. The cooling loads for the oversized units are slightly greater since they maintain slightly lower interior temperatures during the peak load days. Columns 6 and 7 list the calculated SEER for each locality and air conditioner size for both the low- and high-efficiency units. These values vary with climatic conditions as well as with the air conditioner size. The SEER values generally decrease with oversizing because of increased cycling losses. The difference between the energy consumption of the low-efficiency (Column 3) and high-efficiency (Column 4), properly sized units is shown in Column 8. Column 9 represents the maximum energy savings that would occur if the original 50% oversized, low-efficiency air conditioning unit was replaced by a properly sized, high-efficiency unit.

Columns 11 and 12 reflect annual dollar savings for the respective energy savings in Columns 8 and 9, on the basis of estimated average electricity prices (Column 10) for the state in which the city is located.[13] The air conditioning unit ratings, shown in Column 13, are those for the properly sized units and for 25% and 50% oversized units. (The 25% and 50% oversized units have 25% and 50%, respectively, greater rated capacities than that for the properly sized unit.) Columns 14 and 15 give the estimated installed cost of new replacement air conditioning units. Column 14 shows the replacement costs for units having a rated EER = 6, and Column 15 shows the replacement costs for units having a rated EER = 10. The following equations were used for the installed unit costs:

- For EER = 6, cost = 250 + (0.025)(Btu/h rating);
- For EER = 10, cost = 400 + (0.04)(Btu/h rating).

These equations were developed from a regression of dealer costs from manufacturers for units having a variety of efficiencies. These costs were then doubled to include installation costs, keeping in mind that only the air conditioning unit needs replacing—not the ducting, wiring, etc.

The final results of this evaluation are benefit versus cost for replacement air conditioners, shown as simple payback in Columns 16 and 17. Simple payback, as used here, is defined as installed cost divided by first-year energy savings. Column 16 shows the payback of replacing a low-efficiency air conditioning unit (EER = 6) with a high-efficiency unit (EER = 10) with both the existing and replacement units being properly sized for the load (zero percent oversized). The upper number is based on charging full

Table 5.1. Air conditioning cost/benefit evaluation

Column numbers are given at the top of the table.
Numbers in brackets are ARI-rated energy efficiency ratios [EERs].

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
City	Percent oversize	Annual energy consumption (kWh)		Seasonal cooling load (10^6 Btu)	SEER		Energy savings (kWh)		Electricity price (¢/kWh)	Annual savings ($)		Air conditioner rating (Btu/h)	Air conditioner cost ($)		Simple payback (years)	
		[6]	[10]		[6]	[10]	[6] − [10]	50%[6] − 0%[10]		[6] − [10]	50%[6] − 0%[10]		[6]	[10]	0%[6] − 0%[10]	50%[6] − 0%[10]
Albuquerque, N. Mex.	0	5,060	3,148	26.471	5.23	8.41	1,912			189.48		26,000	900	1,440		
	25	5,387	3,344	26.587	4.94	7.95	2,043	2,556	9.91	202.46	253.30	32,500	1,063	1,700	7.60/2.85	5.68/0.85
	50	5,704	3,534	26.634	4.67	7.54	2,170			215.05		39,000	1,225	1,960		
Atlanta, Ga.	0	4,766	2,954	26.885	5.64	9.10	1,812			94.59		23,000	825	1,320		
	25	5,122	3,167	27.306	5.33	8.62	1,955	2,485	5.22	102.05	129.72	28,750	969	1,550	13.95/5.23	10.18/1.60
	50	5,439	3,358	27.498	5.06	8.19	2,081			108.63		34,500	1,113	1,780		
Bakersfield, Calif.	0	7,159	4,385	37.115	5.18	8.46	2,774			190.02		28,000	950	1,520		
	25	7,621	4,662	37.312	4.90	8.00	2,959	3,661	6.85	202.69	250.78	35,000	1,125	1,800	8.00/3.00	6.06/0.88
	50	8,046	4,917	37.370	4.64	7.60	3,129			214.34		42,000	1,300	2,080		
Bangor, Maine	0	650	535	1.757	2.70	3.28	115			9.04		8,000	450	720		
	25	673	549	1.801	2.68	3.28	124	157	7.86	9.75	12.34	10,000	500	800	79.6/29.9	58.3/13.78
	50	692	561	1.825	2.64	3.25	131			10.30		12,000	550	880		
Birmingham, Ala.	0	4,411	2,735	24.673	5.59	9.02	1,676			85.48		17,000	675	1,080		
	25	4,685	2,899	24.892	5.31	8.59	1,786	2,191	5.10	91.09	111.74	21,250	781	1,250	12.6/4.7	9.7/1.72
	50	4,926	3,044	24.949	5.06	8.20	1,882			95.98		25,500	888	1,420		
Boise, Idaho	0	1,678	1,136	7.560	4.51	6.65	542			16.37		15,000	625	1,000		
	25	1,773	1,193	7.689	4.34	6.45	580	716	3.02	17.52	21.62	18,750	719	1,150	61.1/22.9	46.3/8.65
	50	1,852	1,241	7.715	4.17	6.22	611			18.45		22,500	813	1,300		
Boston, Mass.	0	1,573	1,074	7.102	4.51	6.61	499			48.60		19,000	725	1,160		
	25	1,667	1,130	7.184	4.31	6.36	537	681	9.74	52.30	66.33	23,750	844	1,350	23.87/8.95	17.49/2.97
	50	1,755	1,183	7.229	4.12	6.11	572			55.71		28,500	963	1,540		
Chicago, Ill.	0	2,528	1,635	12.711	5.03	7.77	893			58.05		22,000	800	1,280		
	25	2,695	1,736	12.870	4.78	7.41	959	1,217	6.50	62.34	79.11	27,500	938	1,500	22.05/8.27	16.18/2.59
	50	2,852	1,830	12.969	4.55	7.09	1,022			66.34		33,000	1,075	1,720		
Corpus Christi, Tex.	0	11,070	6,687	65.030	5.87	9.72	4,383			333.55		28,000	950	1,520		
	25	11,796	7,123	65.511	5.55	9.20	4,673	5,801	7.61	355.62	441.46	35,000	1,125	1,800	4.56/1.71	3.44/0.50
	50	12,488	7,538	65.807	5.27	8.73	4,950			376.70		42,000	1,300	2,080		
Denver, Colo.	0	1,478	1,017	6.542	4.43	6.43	461			29.73		15,000	625	1,000		
	25	1,561	1,067	6.657	4.26	6.24	494	617	6.45	31.86	39.80	18,750	719	1,150	33.64/12.61	25.13/4.70
	50	1,634	1,111	6.699	4.10	6.03	523			33.73		22,500	813	1,300		
Fort Worth, Tex.	0	8,683	5,290	47.275	5.44	8.94	3,393			258.21		33,000	1,075	1,720		
	25	9,317	5,670	47.814	5.13	8.43	3,647	4,609	7.61	277.54	350.74	41,250	1,281	2,050	6.66/2.50	4.90/0.66
	50	9,899	6,019	48.080	4.86	7.99	3,880			295.27		49,500	1,488	2,380		
Great Falls, Mont.	0	924	693	3.144	3.40	4.54	231			10.05		15,000	625	1,000		
	25	967	719	3.196	3.31	4.45	248	312	4.35	10.79	13.57	18,750	719	1,150	99.50/37.31	73.69/13.78
	50	1,005	742	3.218	3.20	4.34	263			11.44		22,500	813	1,300		
Knoxville, Tenn.	0	3,886	2,431	21.459	5.52	8.83	1,455			58.64		22,000	800	1,280		
	25	4,153	2,591	21.737	5.23	8.39	1,562	1,965	4.03	62.95	79.19	27,500	978	1,500	23.43/8.19	16.16/2.59
	50	4,396	2,737	21.864	4.97	7.99	1,659			66.86		33,000	1,075	1,720		
Little Rock, Ark.	0	6,160	3,784	33.784	5.48	8.93	2,376			178.68		28,000	950	1,520		
	25	6,589	4,041	34.048	5.17	8.43	2,548	3,207	7.52	191.61	241.17	35,000	1,125	1,800	8.51/3.19	6.30/0.91
	50	6,991	4,282	34.222	4.90	7.99	2,709			203.72		42,000	1,300	2,080		
Los Angeles, Calif. (Airport)	0	1,781	1,189	8.637	4.85	7.26	592			40.55		18,000	700	1,120		
	25	1,908	1,265	8.729	4.57	6.90	643	840	6.85	44.05	57.54	22,500	812	1,300	27.62/10.36	19.46/3.39
	50	2,029	1,337	8.783	4.33	6.57	692			47.40		27,000	925	1,480		
Medford, Oreg.	0	2,390	1,566	11.920	4.99	7.61	824			26.62		17,000	675	1,080		
	25	2,525	1,648	12.034	4.77	7.30	877	1,080	3.23	28.33	34.88	21,250	781	1,250	40.6/15.2	31.0/5.50
	50	2,646	1,720	12.069	4.56	7.02	926			29.91		25,500	888	1,420		
Miami, Fla.	0	12,090	7,268	73.237	6.06	10.08	4,822			281.60		24,000	850	1,360		
	25	13,008	7,819	74.465	5.72	9.52	5,189	6,527	5.84	303.04	381.18	30,000	1,000	1,600	4.83/1.81	3.57/0.55
	50	13,795	8,291	74.873	5.43	9.03	5,504			321.43		36,000	1,150	1,840		
Minneapolis, Minn.	0	1,488	1,022	6.687	4.49	6.54	466			24.74		18,000	700	1,120		
	25	1,581	1,077	6.828	4.32	6.34	504	638	5.31	26.76	33.88	22,500	812	1,300	45.27/16.98	33.06/5.76
	50	1,660	1,125	6.883	4.15	6.12	535			28.41		27,000	925	1,480		
Minot, N. Dak.	0	912	687	3.247	3.56	4.73	225			13.46		12,000	550	880		
	25	952	711	3.305	3.47	4.65	241	302	5.98	14.41	18.06	15,000	625	1,000	65.4/24.5	48.7/9.97
	50	989	733	3.332	3.37	4.55	256			15.31		18,000	700	1,120		
New Orleans, La.	0	8,516	5,173	50.734	5.96	9.81	3,343			208.27		25,000	875	1,400		
	25	9,100	5,524	51.315	5.64	9.29	3,576	4,453	6.23	222.78	277.42	31,250	1,031	1,650	6.72/2.52	5.05/0.76
	50	9,626	5,840	51.536	5.35	8.82	3,786			235.87		37,500	1,188	1,900		
Orlando, Fla.	0	9,895	5,982	56.966	5.76	9.52	3,913			228.52		28,000	950	1,520		
	25	10,618	6,415	57.450	5.41	8.96	4,203	5,324	5.84	245.46	310.92	35,000	1,125	1,800	6.65/2.49	4.89/0.71
	50	11,306	6,828	57.769	5.11	8.46	4,478			261.52		42,000	1,300	2,080		
Phoenix, Ariz.	0	12,760	7,721	65.350	5.12	8.46	5,039			435.87		36,000	1,150	1,840		
	25	13,535	8,187	65.537	4.84	8.01	5,348	6,562	8.65	462.60	567.61	45,000	1,335	2,200	4.22/1.58	3.24/0.42
	50	14,283	8,635	65.591	4.59	7.60	5,648			488.55		54,000	1,600	2,560		
Reno Nev.	0	1,565	1,071	6.890	4.40	6.43	494			29.10		14,000	600	960		
	25	1,652	1,124	6.988	4.23	6.22	528	659	5.89	31.10	38.82	17,500	688	1,100	33/12.4	24.7/4.47
	50	1,730	1,171	7.031	4.06	6.00	559			32.93		21,000	775	1,240		
Sacramento, Calif.	0	5,255	3,270	27.956	5.32	8.55	1,985			135.97		24,000	850	1,360		
	25	5,593	3,473	28.064	5.02	8.08	2,120	2,644	6.85	145.22	181.11	30,000	1,000	1,600	10.0/3.8	7.5/1.16
	50	5,914	3,666	28.103	4.75	7.67	2,248			153.99		36,000	1,150	1,840		
Salt Lake City, Utah	0	2,700	1,745	12.849	4.76	7.36	955			38.20		22,000	800	1,280		
	25	2,863	1,843	12.965	4.53	7.03	1,020	1,265	7.40	40.80	50.60	27,500	938	1,500	33.51/12.57	25.30/4.05
	50	3,010	1,931	12.999	4.32	6.73	1,079			43.16		33,000	1,075	1,720		
San Antonio, Tex.	0	7,484	4,553	42.209	5.64	9.27	2,931			233.05		20,000	750	1,200		
	25	7,956	4,856	42.580	5.35	8.80	3,120	3,820	7.61	237.43	290.70	25,000	875	1,400	5.1/1.9	4.1/0.69
	50	8,373	5,087	42.658	5.09	8.39	3,286			250.06		30,000	1,000	1,600		
San Francisco, Calif.	0	733	591	1.885	2.57	3.19	142			9.73		15,000	625	1,000		
	25	767	612	1.910	2.49	3.12	155	206	6.85	11.44	14.11	18,750	719	1,150	102.8/38.5	70.9/12.97
	50	797	630	1.921	2.41	3.05	167			11.44		22,500	813	1,300		
Seattle, Wash.	0	599	513	1.293	2.16	2.52	86			1.72		12,000	550	880		
	25	618	524	1.340	2.17	2.56	94	119	2.00	1.88	2.38	15,000	625	1,000	511.63/191.86	369.75/75.63
	50	632	533	1.358	2.15	2.55	99			1.98		18,000	700	1,120		
Spokane, Wash.	0	1,199	859	4.759	3.97	5.54	340			6.80		15,000	625	1,000		
	25	1,259	895	4.810	3.82	5.37	364	453	2.00	7.28	9.06	18,750	719	1,150	147/55	110/20.64
	50	1,312	927	4.832	3.68	5.21	385			7.70		22,500	813	1,300		
Syracuse, N.Y.	0	1,218	861	5.342	4.39	6.20	357			34.06		16,000	650	1,040		
	25	1,290	904	5.454	4.23	6.03	386	493	9.54	36.82	47.03	20,000	750	1,200	30.53/11.45	22.11/4.04
	50	1,354	942	5.519	4.08	5.86	412			39.30		24,000	850	1,360		
Topeka, Kans.	0	3,760	2,362	20.268	5.39	8.58	1,398			92.69		24,000	850	1,360		
	25	4,005	2,509	20.452	5.11	8.15	1,496	1,864	6.63	99.18	123.58	30,000	1,000	1,600	14.67/5.50	11.01/1.70
	50	4,226	2,641	20.529	4.86	7.77	1,585			105.09		36,000	1,150	1,840		
Washington, D.C.	0	3,171	2,014	16.787	5.29	8.34	1,157			73.82		23,000	825	1,320		
	25	3,398	2,150	17.059	5.02	7.93	1,248	1,588	6.38	79.62	101.31	28,750	969	1,550	17.88/6.71	13.03/2.04
	50	3,602	2,273	17.179	4.77	7.56	1,329			84.79		34,500	1,113	1,780		

installed price for the replacement unit. The lower number is based on the difference between the installed cost of a high-efficiency unit minus the cost of purchasing a replacement low-efficiency unit, assuming the original in-place unit needed replacing (because it was not functioning or had high maintenance costs).

Column 17 shows the payback of replacing an existing low-efficiency (EER = 6) unit that is 50% oversized with a high-efficiency (EER = 10) unit of proper size (zero percent oversized). The upper number of Column 17 is based on charging the full installed price for the replacement unit. The lower number is based on the difference between installing a new, properly sized, high-efficiency unit and the cost of installing a replacement 50% oversized, low-efficiency unit, again assuming that the original in-place unit needs replacing.

It should be mentioned that existing units often are oversized. Initial installations tend to be oversized because of the historical process for load estimating; that is, using the summation of numerous instantaneous peak loads (conduction, infiltration, internal, solar) that, in reality, do not occur simultaneously. The contractor then often selects the next larger sized unit and sometimes even a larger unit to provide "insurance" against call back on an unusually hot day. Added to this trend for oversizing can be a reduced load resulting from other conservation measures of the RCS program and/or other recent trends in conservation of energy in a house.

5.3 Sample Results for Little Rock, Arkansas

The use of the results in Table 5.1 is illustrated in Fig. 5.1 for one selected city, Little Rock, Arkansas. The energy use (before and after air conditioner replacement), energy and cost savings potential, retrofit costs, and payback are illustrated in this figure.

The annual energy consumption (Columns 3 and 4 of Table 5.1) for properly sized units and 50% oversized units for the original (EER = 6) and replacement (EER = 10) systems are shown at the four corners of the quadrilateral. The corresponding dollar values, in brackets, are the estimated installed costs of replacement air conditioning units (Columns 14 and 15 of Table 5.1). For example, the upper left corner of Fig. 5.1 shows that the annual energy consumption for air conditioning the prototype house in Little Rock (properly sized and EER = 10) is 3784 kWh. The estimated cost of installing this replacement unit is $1,520. Along the connecting lines between the corners are indicated the kilowatt-hours and corresponding dollar savings by moving from any one point (percentage oversized and EER) to any other upgraded condition. If an original 50% oversized air conditioning unit having an EER = 6 was replaced with a properly sized unit having an EER = 10, 3207 kWh would be saved during an average cooling season (Column 9 of Table 5.1), resulting in a seasonal dollar savings of $241 (Column 12 of Table 5.1).

Finally, the simple payback (in years) is shown on the left vertical line of Fig. 5.1 (upgrading the EER, assuming that the original unit was properly sized) and on the diagonal line (upgrading the EER, as well as reducing the size from 50% oversized to properly sized). These results are for two means of estimating air conditioner replacement costs: (1) using the full cost of installing the new unit and (2) using the difference

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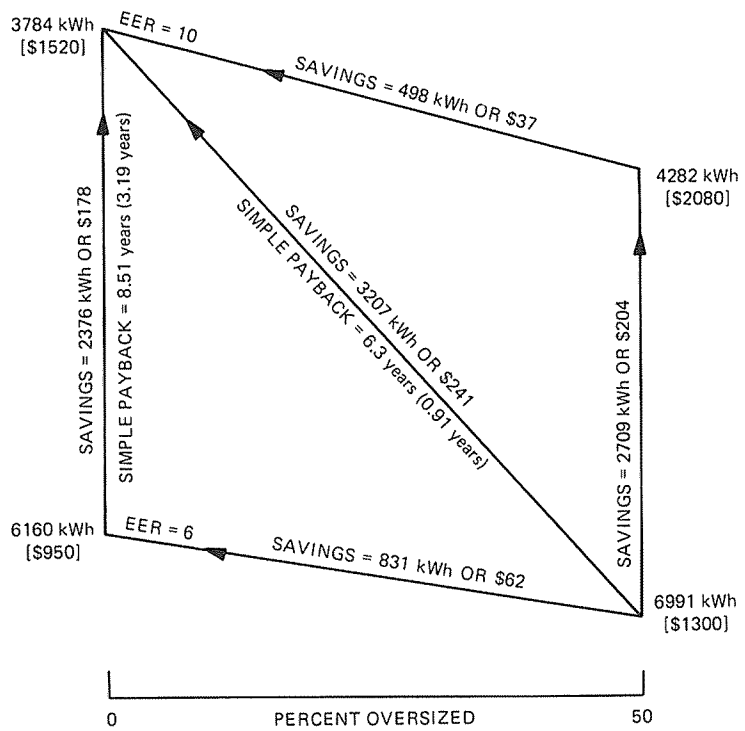


Fig. 5.1. Annual cooling energy and costs (7.52¢/kWh) for prototype two-story house in Little Rock, Arkansas, with central air conditioning (two sizes and two EERs). Notes: The values in the corners represent the following: The upper number is the seasonal energy use (in kilowatt-hours); the lower number in brackets is the installed capital cost of the replacement unit. For the simple payback designation, the first number represents the full installed cost of a replacement unit; the second number represents the difference between the full replacement cost and replacing the original unit without upgrading. The savings designation is the difference in annual energy use; the dollar value represents the cost savings for annual energy use.

between the full replacement unit cost and the cost of replacing the original unit without upgrading (shown in parentheses). Both numbers for each case are from Columns 16 and 17 of Table 5.1.

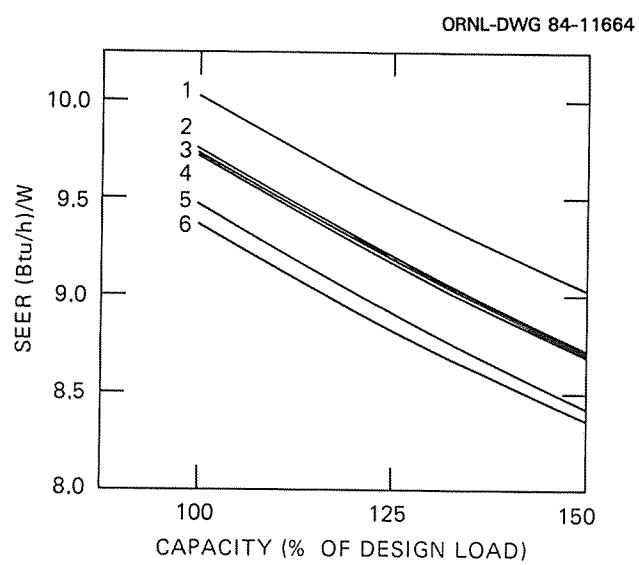
As indicated in Fig. 5.1, the homeowner can realize the quickest payback by selecting a high-efficiency replacement unit that is properly sized for the residence design load. In many cases, the existing unit is oversized because of caution used in size selection plus reduced loads after implementation of other energy conservation measures. For a 50% oversized situation, the example here indicates a simple payback of 6.3 years, on the basis of full retrofit cost, or 0.91 year if the original unit has a short expected life and needs replacement. The replacement cost would, in the latter case, be the difference between the cost of a high-efficiency, properly sized unit and the cost of a low-efficiency, oversized unit. It is noted that the additional cost for a higher efficiency unit is partially offset by purchasing a smaller capacity unit that has a higher SEER value.

5.4 Sensitivity Analysis

Various factors such as house construction, solar effects, and internal loads influence the air conditioning load, as shown in Sect. 4.2. The influence of these factors on the replacement air conditioner cost/benefit results (Sect. 5.2) was investigated for the prototypical house located in Little Rock, Arkansas.* In this analysis, an air conditioning system with an EER of 10 and sized properly for the peak load was selected for cases having different internal loads and window shading. The SEER for the system was then calculated from the results predicted by the DOE-2.1A program for each of these cases. These calculations were then repeated for the same cases, assuming 25% and 50% oversized air conditioning systems. The results of these calculations are summarized in Fig. 5.2.

The base case (Curve 5) is for the same prototypical house used for the cost/benefit analysis. This is the Hastings two-story house occupied by four people, having an average

*For the sensitivity analysis, 1981 Little Rock, Arkansas, weather data was used instead of the TMY weather data for the same city. The use of data from this source should not have any effect on the overall conclusions of this analysis, however.



LEGEND

CURVE NO.	HOUSE DESCRIPTION	SHADING COEFFICIENT	AVERAGE INTERNAL EQUIPMENT LOAD, kW	% DIFFERENCE FROM BASE CURVE 5
1	HASTINGS RANCH	0.6	1	6
2	HASTINGS TWO-STORY	0.6	2	3
3	LARGE RANCH	0.6	1	3
4	HASTINGS TWO-STORY	0.9	1	3
5 (BASE)	HASTINGS TWO-STORY	0.6	1	0
6	HASTINGS TWO-STORY	0.6	0	−1

Fig. 5.2. Sensitivity of varying house sizes, internal loads, and window shadings on air conditioner seasonal energy efficiency ratio (SEER).

internal equipment load of 1 kW, and a window shading coefficient of 0.6.[14] With only one variable changed at a time, Curve 6 is for the same case except that the equipment internal load is reduced to zero, which lowers the SEER about 1%. On the other hand, increasing this average load to 2 kW increases the SEER about 3%, as shown in Curve 2. Retaining the 1-kW average internal equipment load but changing the window shading coefficient from 0.6 to 0.9 also increases the SEER about 3%, as shown in Curve 4. The slopes of all of these curves with oversized equipment remain about the same. This is partially an artifact of using the DOE-2.1A default performance factors (Appendix A) in this study.

Similar curves were generated for two other houses to study the effects of house construction. Both of these houses had the same internal loads and window shading coefficients as those for the base case. Curve 1 is for the Hastings one-story ranch house,[11] which is smaller than the base case house. The SEER is predicted to be about 6% higher than that for the base case. For a larger house, Curve 3 was generated for a large ranch house having a full conditioned basement that is exposed on the back of the house. This is one of the houses in the Electric Power Research Institute data base located in Birmingham, Alabama[15] (Little Rock weather data were used here); the SEER increased over the base case value by about 3%.

It was concluded from this sensitivity analysis that the influence of these parameters on the SEER is not great, and they should not have any pronounced effects on the conclusions of the replacement air conditioner cost/benefit analysis.

5.5 RCS Model Audit Application

An examination of the results presented in Table 5.1 shows that the SEER of an installed air conditioning unit is dependent on both its rated EER and the seasonal climate conditions.* Climate conditions, of course, are dependent on the geographical locations.

To apply these results in the RCS Model Audit procedure,[3] the ratios of the SEERs to the rated EERs were calculated from the data for properly sized air conditioning systems presented in Table 5.1 and plotted on a map of the contiguous 48 states. Contours of these ratios were developed from the plotted data for the 32 cities, and they are shown in Fig. 5.3. It can be seen that they vary from 1.0 in south Florida to less than 0.6 in the very northern part of the country. These correlations do not extend to the west coast region because of the high variability of the weather in that region.

In the RCS Model Audit procedure, the ratios plotted in Fig. 5.3 are regarded as multipliers to obtain the SEER from the rated EER for a properly sized air conditioning unit. Existing air conditioning units are often oversized either because of the contractor's cautious approach or by conservation measures made after installation, as discussed in Sect. 5.2. This usually results in the degradation of the SEER value. To account for this, the data in Table 5.1 shows that there is generally about 0.2% reduction in the SEER

*The SEER as used in this study includes the energy used annually by the crankcase heater. If this energy consumption is ignored, the ratio of the SEER to the rated EER is about 1.0 for a properly sized unit throughout the country. Further details are given in Appendix C.

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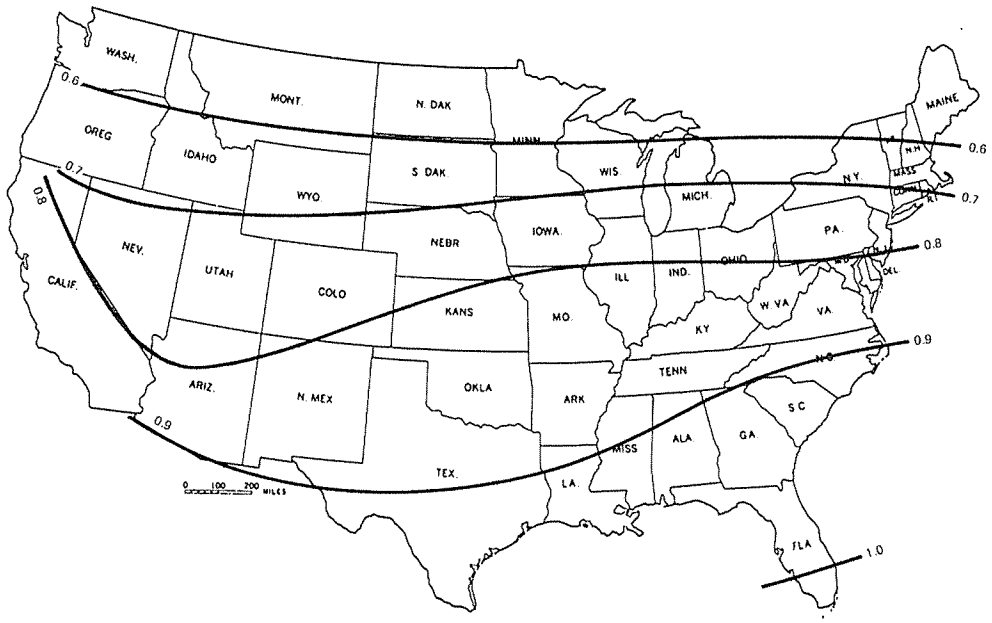

Fig. 5.3. Ratio of the SEER to the rated EER for an air conditioning unit of proper size.

value for every 1% that the unit is oversized. Therefore, in the RCS Model Audit procedure, there is a multiplier based on this relationship, in addition to the multiplier shown in Fig. 5.3 for the properly sized unit.

In selecting a replacement unit, a homeowner should specify a unit that is minimally oversized. The savings for this smaller size unit will be in the form of both lower initial capital costs and lower operating costs, because the unit will have a higher SEER value. These two factors will reduce the unit's payback time.

Also, when the payback potential of a more efficient air conditioning unit is considered, the expected life of the existing unit should be factored into the evaluation. It was shown in Table 5.1 that the differential cost of buying a more efficient unit when replacing an unit that cannot be repaired economically results in a shorter payback time. If an existing unit has only a limited expected life, the payback potential may be close to that for replacement of a nonoperating unit.

6. CONCLUSIONS

The primary purpose of this study was to evaluate the economic benefits of replacing an existing residential air conditioning unit having a relatively low efficiency rating with a new high-efficiency unit. The effect of climate and other parameters on the unit's seasonal efficiency, and therefore its operating cost, also had to be evaluated. In the course of the study, three other important results were determined: (1) the DOE-2.1A model proved to be a valuable evaluation tool for this analysis; (2) the seasonal air conditioning unit efficiency when the crankcase heater energy is included is greatly affected by weather conditions and, to a very limited degree, by house configuration and loads not related to weather; and (3) life-style can play an important role in the summer cooling load.

6.1 Benefits of Replacement Air Conditioner

It was found in this study that the simple economic payback times for replacing a central air conditioning unit vary widely—depending on the location, the excess capacity, and the state of repair of the existing unit. The payback times vary from 0.4 year to over 100 years. The trends can be illustrated with the use of Fig. 6.1. In the southern region (Zone I), simple payback time for a properly sized, high-efficiency unit replacing a properly sized, low-efficiency unit was calculated to vary from 5 years in the lower part of the region to about 10–15 years in the upper part of the region. If the existing unit needs replacement because of disrepair, these payback times are reduced to about 2 years and

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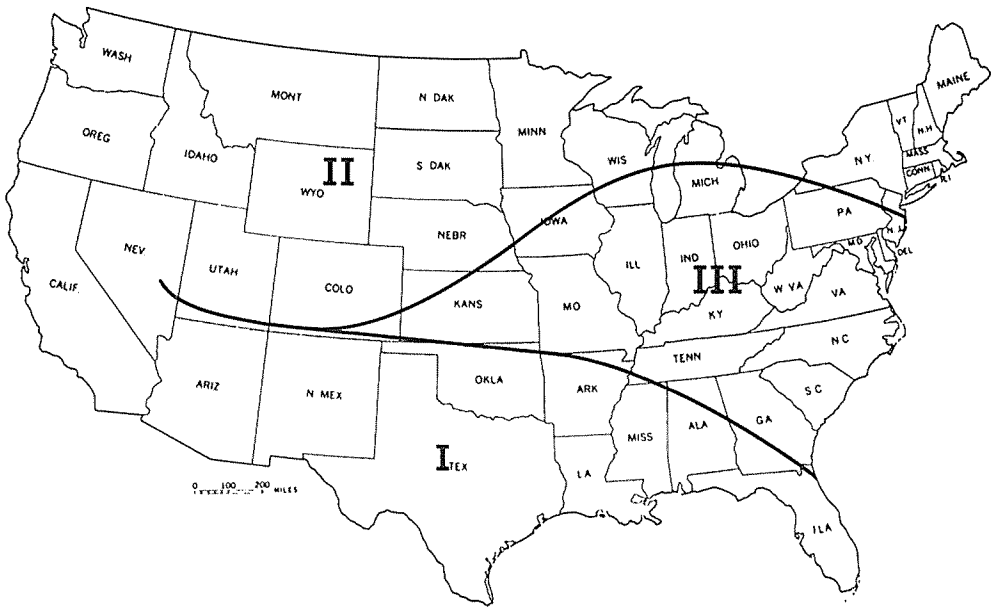

Fig. 6.1. Replacement air conditioner payback regions in the United States.

5 years, respectively. These payback times are also reduced if an existing oversized unit is replaced with a properly sized unit. For the extreme case of replacing a failed, 50% oversized, low-efficiency unit with a properly sized, high-efficiency unit, the payback time is less than 1 year throughout Zone I.

In a large northern part of the country (Zone II), the simple payback times were calculated to be greater than 10 years. Only in the extreme case, when a failed, 50% oversized, low-efficiency unit is replaced with a properly sized, high-efficiency unit do the payback times become attractive (4–8 years) in the lower parts of this region (e.g., Salt Lake City, Denver, Minneapolis, Syracuse, and Boston), they still exceed 10 years for the cities located in the extreme north (e.g., Seattle, Great Falls, Minot, and Bangor).

The payback times for the intermediate region (Zone III), which are the mid-central and the mid-Atlantic states, vary between the values calculated for the other two zones. They are always 10 years or longer for the replacement of an air conditioning unit that is in good operating condition. However, if the existing unit has failed, the payback times in this region drop to 5 to 8 years for the replacement unit. Again, if the failed, low-efficiency air conditioning unit is 50% oversized, the simple payback times are reduced to 1.5 to 3 years for a properly sized, high-efficiency replacement unit in this region.

6.2 Simulation Model Effectiveness

The DOE-2.1A model was shown to be capable of reproducing measured energy use data for residential electric resistance heating and air conditioning. This was demonstrated conclusively on the TECH house, where input data to the model and the measured data were known with confidence. There was about an 11% difference for winter heating with a heat pump, but the reason for this was not explored fully. However, default efficiencies for heat pump heating in the program do not consider the loss of efficiency due to defrosting. This accounts for at least part of the difference.

The four additional houses in Little Rock selected for the study showed that the differences between the predicted and measured data ranged from low to high. These differences are believed to be primarily due to the assumptions about the life-style in these houses. The model has the ability to make corrections for life-style, which has proven to be a significant factor in the RCS program.

6.3 Air Conditioning Unit Seasonal Efficiency Evaluation

The only efficiency rating of value in assessing residential cooling energy requirements is the seasonal energy efficiency ratio (SEER). The ARI-published SEER values are based on a dry-coil cycling test that considers only one cycling rate (on for 6 minutes and off for 24 minutes) and one outdoor temperature (82°F).[9] In this test, both the indoor and outdoor fans are cycled, but no account is made for energy used by the crankcase heater. These published SEER values provide comparative efficiencies among different air conditioning units as a guide to the purchaser.

The SEER values in this study are defined differently from the ARI-published values in that the values in this study include the energy used by the crankcase heaters. The

relations developed in this study for estimating these SEER values from the rated EER values (Sect. 5.5) are designed to be a guide for residential energy use auditors. These relations have been incorporated in the RCS *Model Audit Manual* in an optional procedure for estimating the seasonal cooling energy use.[3] These relations have been developed using the DOE-2.1A program together with its default relations to describe the air conditioning unit performance. Future studies may indicate that the relations and assumptions used to calculate the SEER values should be modified. At that time, the RCS optional procedure for estimating the SEER values should be updated using the modified relations.

6.4 Life-Style Implications

Life-style can be important in the evaluation of home energy consumption. This is especially true in the evaluation of summer cooling loads. In addition to the usual variations in internal loads, thermostat settings, etc., cooling requirements are affected by the control of the air conditioning unit (when it is allowed to run) to satisfy comfort as defined by each individual homeowner. In general, there is greater tolerance to off-design cooling requirements than to heating requirements. Often, for example, satisfactory comfort can be maintained by opening windows in mild weather that would otherwise require mechanical cooling if left to the automatic controls.

This study has shown conclusively the impact of life-style on cooling energy consumption (see Sect. 4.2.2), where differences as high as 60% can be corrected by reasonable life-style assumptions. The recommendations in the *Model Audit Manual*[3] of using historical utility data for heating and cooling load evaluations whenever possible has been further justified by this study.

7. REFERENCES

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Appendix A

DOE-2.1A PROGRAM METHODOLOGY AND ASSUMPTIONS IN THE FORM USED IN THIS STUDY

Appendix A

DOE-2.1A PROGRAM METHODOLOGY AND ASSUMPTIONS IN THE FORM USED IN THIS STUDY

When interpreting the results of this study, it is important that the reader has knowledge of the assumptions used in the DOE-2.1A simulation program. The purpose of this appendix is to present an overall description of this program and related assumptions as used in this study. An overview of the program's simulation methodology has been reported by Hirsch,[1] and further details of the program's algorithms and assumptions are given in the *DOE-2 Engineers Manual.*[2] Much of the description below follows the material presented by Hirsch.

A.1 DOE-2.1A Program

The DOE-2.1A program describes the flow of heat in a building and the associated HVAC equipment on a hourly basis. The program uses detailed data for the building geometry and construction, for HVAC equipment, and for the weather to predict the energy flow in the building. Internal heat loads in the form of people, lights, and equipment are incorporated in the energy flow analysis. One-dimensional heat flow is assumed for all the building surfaces. For external surfaces that have significant mass, the program accounts for the delay of the transfer of heat by the use of response factors. The response factor method is basically the use of a time series to describe the heat flow through a wall due to a unit temperature difference pulse through the wall surfaces. The constants in the time series are called response factors. Heat flow through the remaining exterior walls, interior walls, underground walls, and floors are assumed to be steady state.[2]

The program uses the weighting factor method—as opposed to the detailed thermal balance method—to calculate the heat or cooling loads in each zone of the building, assuming that the interior temperature in each zone is fixed. It then uses the weighting factor method again to modify the loads calculated at fixed zone temperatures and to calculate the HVAC equipment heat addition or heat extraction rates.[2]

A.2 Weighting Factor Method

The weighting factor method is basically the use of time series to describe a zone's thermal behavior due to a unit pulse of heat gain or air temperature change in the zone.[2,3] It accounts for the delays of heat flow to or from the interior air due to the absorption of energy in the surfaces and radiation exchange between the various surfaces in the zone. In DOE-2.1A, the relation (transfer function) describing the zone's response to a unit heat pulse is in the form:

$$\frac{Q(z)}{q(z)} = \frac{v_0 + v_1 z^{-1} + v_2 z^{-2}}{1 - w_1 z^{-1} - w_2 z^{-2}} , \tag{A.1}$$

where $q(z)$ and $Q(z)$ are the z-transforms of the heat pulse and the zone's thermal response, respectively, and v_0, v_1, v_2, w_1, and w_2 are the weighting factors. The weighting factors can be preselected by the user or they can be calculated by the program. Preselected values based on estimated floor weights were used in this study.

Assuming that $q(z)$ and $Q(z)$ are polynomials, the following relation can be obtained from Eq. A.1 by equating coefficients.

$$Q_\tau = v_0 q_\tau + v_1 q_{\tau-1} + v_2 q_{\tau-2} + w_1 Q_{\tau-1} + w_2 Q_{\tau-2} \quad , \tag{A.2}$$

where q_τ, $q_{\tau-1}$, and $q_{\tau-2}$ are the heat pulses and Q_τ, $Q_{\tau-1}$, and $Q_{\tau-2}$ are the thermal responses at the hours τ, $\tau-1$, and $\tau-2$. The program uses this relation to describe the zone's thermal response to heat conduction through the external walls, direct and diffuse solar radiation through the windows, building occupants and equipment, and artificial lighting. In the part of the program where these calculations are done, called LOADS, the zone air temperature remains constant at a preselected value. A heat balance is done in LOADS to determine the amount of energy that must be added or extracted by the HVAC system to maintain this fixed temperature.

Once the sensible heat addition or extraction rate in LOADS, Q_τ^L, is known, the DOE-2.1A then uses a relation analogous to Eq. A.1 to calculate the actual zone temperature and the actual HVAC system sensible heat addition or extraction rate. The part of the DOE-2.1A program where this is done is called SYSTEMS, and the transfer function used here is

$$\frac{Q^{\text{net}}(z)}{t(z)} = \frac{g_0 + g_1 z^{-1} + g_2 z^{-2} + g_3 z^{-3}}{1 + p_1 z^{-1} + p_2 z^{-2}} \quad , \tag{A.3}$$

where $t(z)$ and $Q^{\text{net}}(z)$ are the z-transforms of the air temperature pulse and the zone's thermal response, respectively, and g_0, g_1, g_2, g_3, p_1, and p_2 are the weighting factors. Using the same procedure as for Eq. A.2,

$$Q_\tau^{\text{net}} = g_0 t_\tau + g_1 t_{\tau-1} + g_2 t_{\tau-2} + g_3 t_{\tau-3} - p_1 Q_{\tau-1}^{\text{net}} - p_2 Q_{\tau-2}^{\text{net}} \quad , \tag{A.4}$$

where t_τ, $t_{\tau-1}$, $t_{\tau-2}$, and $t_{\tau-3}$ are the temperature pulses and Q_τ^{net}, $Q_{\tau-1}^{\text{net}}$, and $Q_{\tau-2}^{\text{net}}$ are the zone's thermal responses at the hours τ, $\tau-1$, $\tau-2$, and $\tau-3$.

In SYSTEMS, the temperature pulse is defined as:

$$t = T - T^L \quad , \tag{A.5}$$

where T is the computed zone temperature and T^L is the assumed fixed temperature in LOADS. The thermal response term, Q^{net}, is defined as the net heat addition or extraction from the zone due to the temperature difference pulse in the zone. At time τ, it is:

$$Q_\tau^{\text{net}} = ER_\tau - Q_T^L - (\rho c)_{\text{air}} V_{\text{inf}\tau} t_\tau - \sum (UA)_{\text{Ext}} t_\tau - \sum_{k=1}^{m} (UA)_k (t_\tau - t_\tau^k) \quad , \tag{A.6}$$

where

ER_τ = HVAC system sensible heat addition or extraction rate for the zone;

Q_τ^L = heat addition or extraction rate calculated in LOADS;

$(\rho c)_{\text{air}} V_{\text{inf}\tau} t_\tau$ = temperature correction term for infiltration air heat load;

$\sum (UA)_{\text{Ext}} t_\tau$ = temperature correction term for external wall conduction heat load;

$\sum_{k=1}^{m} (UA)_k (t_\tau - t_\tau^k)$ = temperature correction term for conduction heat load for internal walls between the given zone and the attached zone, k.

Because of the internal wall temperature correction term in this relation, it can be seen that its solution would require simultaneous solution of all these equations for the different zones. To eliminate this, it is assumed that

$$(t_{\tau-1} - t_{\tau-1}^k) \sim (t_\tau - t_\tau^k) \quad . \tag{A.7}$$

Combining Eqs. A.4 through A.7,

$$t_\tau = \frac{F - ER_\tau}{G_0} \quad , \tag{A.8}$$

where

$$F = Q_\tau^L + \sum_{n=1}^{3} G_n t_{\tau-n} + \sum_{n=0}^{2} \sum_{k=1}^{m} \left[p_n (UA)_k t_{\tau-n-1}^k \right], \text{ (Note } p_0 = 1) \quad ,$$

$$G_n = g_n + p_n \left[k_t + (\rho c)_{\text{air}} V_{\tau-n} \right], \text{ (Note } p_0 = 1, p_3 = 0) \quad ,$$

$$k_t = \sum (UA)_{\text{Ext}} + \sum_{k=1}^{m} (UA)_k \quad .$$

The actual quantity of sensible heat added or removed from the zone by the HVAC system during a given hour, ER_τ, depends on the characteristics of the equipment and its thermostat control action. In DOE-2.1A, a linear relationship is used to describe the thermostat control action of the equipment, as illustrated in Fig. A.1. It can be seen that there are five distinct control regions. At the extremes, ER is limited by the capacity of the HVAC equipment for that hour and thus has a constant value. This value is a function of the outdoor dry and the indoor wet bulb temperatures, as discussed in Sect. A.3. Within the dead band region, there is no equipment action, and the value of ER is zero. For the throttling ranges, it is assumed within these ranges that the value of ER_τ is described by the following linear relation:

$$ER_\tau = W + St_\tau \quad . \tag{A.9}$$

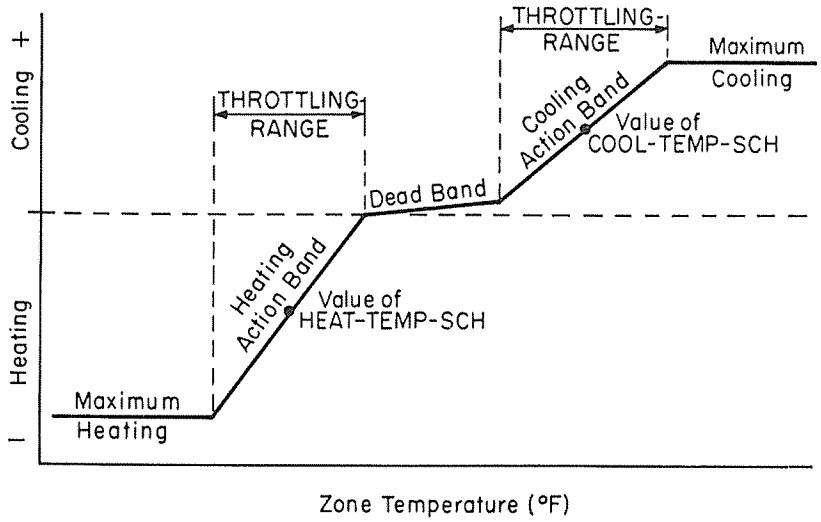

Fig. A.1. DOE-2.1A simulation of HVAC thermostat control action. Source: Lawrence Berkeley Laboratory and Los Alamos National Laboratory, *DOE-2 Engineers Manual, Version 2.1A,* LBL-11353, November 1982.

For on-off (two-position) thermostats used in this study, the throttling ranges were assumed to be 0.5°F. Assuming that the equipment capacity is dependent on the previous hour's conditions, these relations are used together with Eq. A.8 to determine the values of t_τ and ER_τ. These values are then corrected, assuming that the zone's temperature change during the hour is linear. Details of this procedure are given in the *DOE-2 Engineers Manual.*[2]

A.3 HVAC System Performance

A.3.1 Performance Characteristics

The performance of residential air conditioners and heat pumps is dependent on the indoor and outdoor dry bulb temperatures and humidities and the fraction of the time the equipment is used. Also, the performance of furnaces (oil or gas) is dependent on the fraction of time that they are used. Usually, the rated capacities of these units are published at particular temperature and humidity conditions, assuming that the unit is operating at full load. Adjustments must be made to these rated capacities at other operating conditions. The relations used in the DOE-2.1A program to adjust these capacities are described below. They contain factors—linear or quadratic functions of the temperatures, humidities, and fractional operating time—that modify the rated capacities. These functions can be either specified by the user of the program, or they can be the default relations stored in the program. Table A.1 lists the default relations for residential HVAC systems that are used in the equations given below.

Table A.1. Default values of residential HVAC system parameters used in DOE-2.1A[a]

Cooling EIR_R[b] = 0.460
Heat pump EIR_R = 0.360
Minimum T_{db}[c] for heat pump operation = 10°F
Maximum T_{db} heat pump auxiliary heater operation = 17°F
Crankcase heater power = 50 W
Maximum T_{db} for crankcase heater operation = 65°F
No defrost cycle for heat pump.
Furnace EIR_R = 1.35
B_R[d] = 0.241

$f_1 = 0.59815404 + 0.01329984\ T_{wb}{}^e - 0.00514995\ T_{db}$; Min T_{wb} = 60°F, Min T_{db} = 70°F
$f_2 = 4.26472718 - 0.04557954\ T_{wb} - 0.00221998\ T_{db}$; Min T_{wb} = 60°F, Min T_{db} = 70°F
$f_3 = 1.0$
$f_4 = \theta_F{}^f$
$f_5 = 0.49957503 - 0.00765992\ T_{wb} + 0.01066989\ T_{db}$; Min T_{wb} = 60°F, Min T_{db} = 70°F
$f_6 = 0.125 + 0.875\ \theta_c{}^g$
$f_7 = 0.34148808 + 0.00894102\ T_{db} + 0.00010787\ T_{db}^2$
$f_8 = 2.03914613 - 0.03906753\ T_{db} + 0.00045617\ T_{db}^2 - 0.00000203\ T^h$
$f_9 = 0.10969333 + 0.89030667\ \theta_H{}^i$
$f_{10} = 0.01861 + 1.094209\ \theta_H - 0.112819\ \theta_H^2$

[a]Temperatures T_{wb}, T_{db}, and T are in degrees Fahrenheit.
[b]EIR_R = rated electric input ratio.
[c]T_{db} = outdoor dry bulb temperature.
[d]B_R = rated coil bypass factor.
[e]T_{wb} = indoor wet bulb temperature.
[f]θ_F = fraction of the rated fan flow rate.
[g]θ_c = total part cooling load ratio.
[h]T = indoor dry bulb temperature.
[i]θ_H = part heating load ratio.
Source: Lawrence Berkeley Laboratory and Los Alamos National Laboratory, *DOE-2 Reference Manual, Version 2.1A,* LBL-8706 Rev. 2, May 1981.

For direct expansion air conditioning units, the relation used to modify the total rated capacity is:

$$Cap_T = Cap_{TR} \cdot f_1(T_{wb}, T_{db}) \quad , \tag{A.10}$$

where Cap_T is the total capacity, Cap_{TR} is the total rated capacity, T_{wb} is the indoor wet bulb temperature, and T_{db} is the total outdoor dry bulb temperature. Similarly for the unit's total sensible heat capacity,

$$Cap_S = Cap_{SR} \cdot f_2(T_{wb}, T_{db}) + (\rho c)_{\text{air}}\, V(1-B)(T-80) \quad , \tag{A.11}$$

where V is the circulating fan flow rate, $(\rho c)_{\text{air}}$ is the circulating air volumetric heat capacity, and T is the indoor dry bulb temperature;

$$B = B_R \cdot f_3(T_{wb}, T_{db}) \cdot f_4(\theta_F) \quad , \tag{A.12}$$

where B_R is the rated coil bypass factor and θ_F is the fraction of the rated fan flow rate.

To determine the electrical energy consumed by the direct expansion air conditioner, the DOE-2.1A program uses a term called the electric input ratio (EIR), which is defined as the ratio of the electrical energy input (Btu/h) to the total capacity (Cap_T) at the operating conditions (Btu/h). The relation used to modify the rated EIR is:

$$\mathrm{EIR} = \mathrm{EIR}_R \cdot f_5(T_{wb}, T_{db}) \cdot f_6(\theta_c) \quad , \tag{A.13}$$

where EIR_R is the rated EIR and θ_c is the total part cooling load ratio (total heat removed each hour/unit total operating capacity).

Analogous relations are used in the DOE-2.1A program to describe the performance of the heating equipment. For electric resistance heaters, the operating capacity is simply the rated capacity, and the operating EIR is 1.0. For electric heat pumps, the relations for the operating capacity and EIR are:

$$Cap_T = Cap_R \cdot f_7(T_{db}, T) \quad , \tag{A.14}$$

and

$$\mathrm{EIR} = \mathrm{EIR}_R \cdot f_8(T_{db}, T) \cdot f_9(\theta_H) \quad , \tag{A.15}$$

where θ_H is the part heating load ratio (total heat added by the unit each hour/unit total operating capacity).

For a gas or oil furnace, the operating capacity is the rated capacity. The heating input ratio (HIR) for these units, which is analogous to the EIR is determined from the rated ratio (HIR_R) by the relation:

$$\mathrm{HIR} = \mathrm{HIR}_R \cdot f_{10}(\theta_H) \quad . \tag{A.16}$$

A.3.2 Dehumidification

If the temperature of the air adjacent to the air conditioning unit coils drops below its dew point, moisture condenses on the unit's coils and the associated latent heat is removed by the unit. To account for this in the DOE-2.1A program, steady state moisture balances are written for the zones being cooled and for the air conditioning system each hour. Moisture entering the zones through air infiltration or carried into the system by any makeup air is included in these material balances.

For the zones, the moisture balance is:

$$Vw + V_{\mathrm{inf}}w = Vw_{\mathrm{exit}} + V_{\mathrm{inf}}w_0 + V\Delta w \, , \tag{A.17}$$

where V and V_{inf} are the volumetric air circulation and infiltration rates, w is the humidity ratio (pounds water/pounds dry air) of the air in the zones, w_{exit} is the humidity ratio of the air leaving the air conditioning unit, w_0 is the humidity ratio of the outside air, and Δw is the change in the humidity ratio due to moisture addition in the zones.

Not all of the circulating air leaving the zones always reaches the air conditioning unit. If part of it is vented from the air conditioning system, makeup air containing the moisture in the outside air must replace the vented air. The moisture balance for this is:

$$V w_{ent} = V_{vent} w_0 + (V - V_{vent}) w \,, \tag{A.18}$$

where V_{vent} is the air ventilation rate and w_{ent} is the humidity ratio of circulating air entering the air conditioning unit.

In the moisture balance for the air conditioning unit itself, it is assumed in the DOE-2.1A program that the air moving through the unit is composed of two streams: (1) the air stream not influenced by the unit's cooling coils and (2) the air stream that reaches equilibrium conditions at the cooling coil's surface. The fraction of the air that bypasses the coils is the bypass factor, B, and the relation for it is Eq. A.12. This factor is used in the following relation to calculate the temperature at the cooling coil surface.

$$T_{exit} = B T_{ent} + (1-B) T_{surf} \,, \tag{A.19}$$

where T_{ent} and T_{exit} are temperature of the air entering and leaving the unit as determined by the sensible heat removal calculations described in Sects. A.2 and A.3.1, and T_{surf} is the cooling coil surface temperature. The humidity ratio at the coil surface, w_{surf}, is then assumed to be that for saturated air at T_{surf} and atmospheric pressure.

Next, a check is made to determine whether the cooling coil surface is wet or dry. This is done by first assuming that the coils are dry and the humidity ratios of the air entering and leaving the air conditioning unit are identical. Combining Eqs. A.17 and A.18 using this assumption,

$$w_{ent} = w_0 + \frac{(1-P)\Delta w}{F + P} \,, \tag{A.20}$$

where F is V_{inf}/V, P is V_{vent}/V, and w_{ent} is the humidity ratio of the air entering the air conditioner. If w_{ent} is less than w_{surf}, the coils are dry and the humidity ratio in the zone is:

$$w = w_0 + \frac{\Delta w}{F + P} \,, \tag{A.21}$$

If w_{ent} is greater than w_{surf}, the coils are wet, and the humidity ratio of the air leaving the air conditioning unit is calculated by the relation:

$$w_{exit} = B\, w_{ent} + (1 - B) w_{surf} \,. \tag{A.22}$$

Then Eqs. A.17, A.18, and A.22 are combined to give the relation for calculating the zone's humidity ratio in this case to be:

$$w = \frac{(BP + F) w_0 + (1-B) w_{surf} + \Delta w}{1 + F - B(1 - P)} \,. \tag{A.23}$$

A.4 HVAC Design Calculations

The format of the DOE-2.1A program is such that all the design and performance parameters of the HVAC systems can be specified, or a minimum of these parameters can be specified. For the performance parameters, the program has default relations describing the equipment performance at off-design conditions. These parameters were discussed in Sect. A.3.

Simple relations for the equipment capacities and circulating air flow rates do not exist, however. They have to be specified, or the program calculates these capacities from the peak heating and cooling loads predicted in the LOADS part of the program. Basically, the program requires that the following sensible heat balance must be satisfied at design conditions.

$$Cap_R = (\rho c)_{\text{air}} V(T_{\text{ent}R} - T_{\text{exit}R}) \quad , \tag{A.24}$$

where the subscript R refers to the design conditions. The program requires that $T_{\text{ent}R}$ and $T_{\text{exit}R}$ be specified. If the circulating air flow rate is also specified, as was done in this study, the HVAC system capacity is determined by this relation. If the equipment sensible heat capacity is specified, the circulating air flow rate is calculated. If both the equipment capacity and the air flow rate are specified, $T_{\text{exit}R}$ is adjusted to match the other two parameters. If neither the air flow rate or the equipment capacity is specified, the program assumes that the design loads are the peak loads calculated in the LOADS portion of the program. Since the weather and indoor conditions at the time of the peak loads probably are not the design conditions, these peak loads are corrected using the performance factors discussed in Sect. A.3.

For air conditioning systems, when the LOADS peak cooling load is used to calculate the equipment total capacity or the equipment sensible heat removal capacity, the performance factors, used in the relations to calculate the capacities, are functions of the zone's wet bulb temperature. Moreover, the latent heat load is part of the total heat removal capacity of the system. The program's default procedure here is to calculate the wet bulb temperature and latent heat load from the zone's humidity ratio at the time of the LOADS peak cooling load. The humidity ratio is calculated using the procedure outlined in Sect. A.3.2, assuming that the infiltration rate and the outdoor air humidity ratio are those at the time of the peak cooling load. Often, as in this study, total heat removal capacity is specified, but the sensible heat removal capacity is not specifed. The default procedure used in the DOE-2.1A program for this case is to assume that the equipment sensible heat removal capacity is the smaller of the total capacity that was specified or the sensible heat removal capacity calculated from the LOADS peak cooling load. Details of this procedure are given in the *DOE-2 Engineers Manual.*[2]

References for Appendix A

1. J. J. Hirsch, *Simulation of HVAC Equipment in the DOE-2 Program,* LBL-14026, Lawrence Berkeley Laboratory, December 1982.

2. Lawrence Berkeley Laboratory and Los Alamos National Laboratory, *DOE-2 Engineers Manual, Verson 2.1A,* LBL-11353, November 1982.
3. J. F. Kerrisk, *Weighting Factors in the DOE-2 Computer Program,* Los Alamos National Laboratory, LA-8886-MS, June 1981.

Appendix B

ASSUMED INTERNAL LOADS FOR REPLACEMENT AIR CONDITIONER EVALUATION

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ASSUMED INTERNAL LOADS FOR REPLACEMENT AIR CONDITIONER EVALUATION

The internal loads assumed for the prototypical two-story house used in the DOE-2.1A program evaluation of replacement air conditioners are given in Table B.1. It was assumed that the house has four occupants, each generating 400 Btu/h of heat. For the schedule shown in Table B.1, this translates into an average heat rate of 1270 Btu/h. About two-thirds of this energy is assumed to be sensible heat, and the remainder is assumed to be latent heat.

It was assumed that the equipment (lighting and appliances) generate 2 kW or 6820 Btu/h of heat at full load. For the schedule shown in Table B.1, this translates into an average heat rate of 1 kW or 3410 Btu/h. It was further assumed that 75% of this heat rate is sensible heat and that the remainder is latent heat.

Table B.1. Internal load schedules assumed for replacement air conditioner cost/benefit evaluation

Occupants

Full load: 4 persons at 400 Btu/h·person = 1600 Btu/h[a]

Partial loads:

Hours[b]	Fraction of total
1–8	1.00
9–18	0.50
19–24	1.00

Equipment

Full load: 2 kW = 6826 Btu/h[c]

Partial loads:

Hours[b]	Fraction of total
1–7	0.25
8–9	0.75
10–17	0.50
18–22	0.75
22–24	0.50

[a]Portions of total occupant loads partitioned into sensible and latent heat loads determined by DOE-2.1A program default relations. Roughly two-thirds of total is sensible heat, and the remainder is latent heat.

[b]Hour 1 is from 12 p.m. to 1 a.m., etc.

[c]75% of the equipment load is assumed to be sensible heat, and the remainder is assumed to be latent heat.

Appendix C

SENSITIVITY OF THE PREDICTED SEER VALUES TO THE ASSUMED CYCLING DEGRADATION COEFFICIENTS AND CRANKCASE HEATER ENERGY USE

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SENSITIVITY OF THE PREDICTED SEER VALUES TO THE ASSUMED CYCLING DEGRADATION COEFFICIENTS AND CRANKCASE HEATER ENERGY USE

C.1 Assumed Values

The seasonal energy efficiency ratio (SEER) values listed in Table 5.1 were calculated by dividing the energy extracted (Btu/h) by the air conditioning unit by the energy consumed (W) by the unit over the entire year. In this study, the electric input ratio (EIR), defined as the ratio of the electrical energy input (Btu/h) to the unit's total capacity at operating conditions (Btu/h), was specified for the unit operating at full load at the rated condition. The DOE-2.1A program default relations for residential HVAC systems were then used to modify the rated EIR value to calculate the amount of energy consumed by the unit each hour. (These default relations are listed in Appendix A.) One of these relations modifies the EIR value to account for the hours when the unit is operating at part load. This relation is defined as

$$f_6(\theta_c) = \frac{\theta_c}{\mathrm{EER_{cyc}/EER_{ss}}} \quad , \tag{C.1}$$

where

$f_6(\theta_c)$ = EIR modification factor used in Eq. A.13 (Appendix A),
θ_c = part cooling load ratio (total heat removed each hour/total operating capacity),
$\mathrm{EER_{cyc}/EER_{ss}}$ = ratio of cycling EER to steady state EER.

The default value of $f_6(\theta_c)$ for residential systems[1] is

$$f_6(\theta_c) = 0.125 + 0.875\theta_c \quad . \tag{C.2}$$

This relation is plotted in Fig. C.1, and the corresponding $\mathrm{EER_{cyc}/EER_{ss}}$ values are plotted in Fig. C.2.

In addition to degradation of performance due to cycling, energy is consumed by the air conditioning unit's crankcase heater when the unit is not running and the ambient temperature is sufficiently low. Central air conditioning units are usually fitted with these units, which are small electrical resistance heaters, to protect the compressor from liquid accumulation. (The accumulated liquid could cause major damage to the compressor.) For the DOE-2.1A default residential HVAC system, the program assumes that this heater draws 50 W electricity during hours when the unit does not operate and the ambient air temperature is below 65°F.

C.2 Degradation Coefficients

A common figure of merit used in industry to describe a unit's cycling losses is the degradation coefficient, C_D.[2] This ratio is defined as

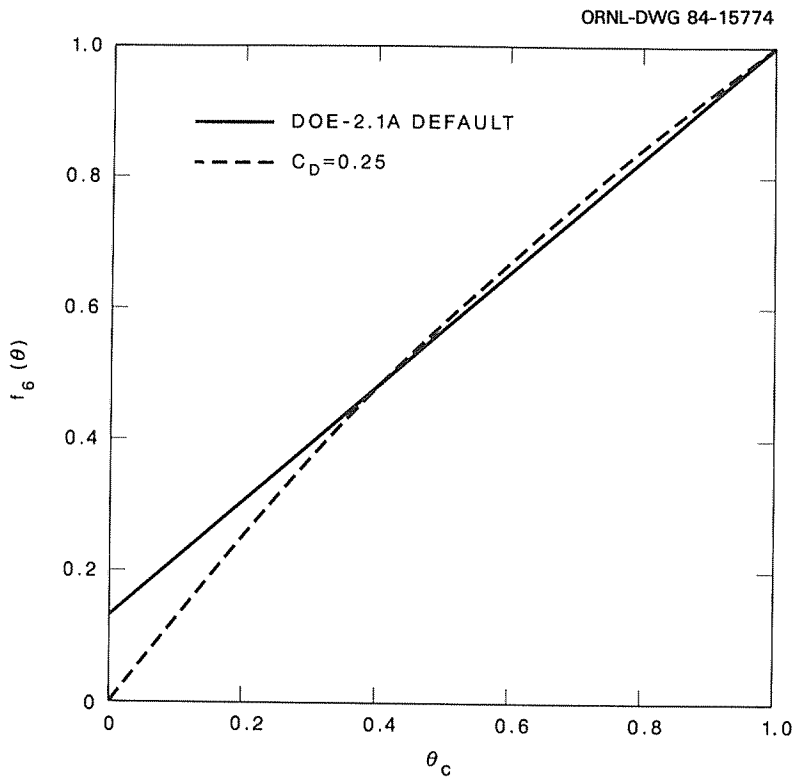

Fig. C.1. Values of EIR (electric input ratio) modification factor, $f_6(\theta_c)$. DOE-2.1A is the building energy use program, C_D is the degradation coefficient, and θ_c is the part cooling load ratio.

$$C_D = \frac{1 - \mathrm{EER}_{\mathrm{cyc,dry}}/\mathrm{EER}_{\mathrm{ss,dry}}}{1 - \theta_c} \tag{C.3}$$

where

$\mathrm{EER}_{\mathrm{cyc,dry}}/\mathrm{EER}_{\mathrm{ss,dry}}$ = ratio of cycling EER to steady state EER for dry (no latent heat removel) conditions.

Test standards have been developed to measure $\mathrm{EER}_{\mathrm{cyc,dry}}/\mathrm{EER}_{\mathrm{ss,dry}}$ and θ_c, from which C_D can be determined. In lieu of tests, a C_D value of 0.25 is considered acceptable by the industry.[2] Assuming

$$\frac{\mathrm{EER}_{\mathrm{cyc}}}{\mathrm{EER}_{\mathrm{ss}}} \sim \frac{\mathrm{EER}_{\mathrm{cyc,dry}}}{\mathrm{EER}_{\mathrm{ss,dry}}} = 1 - C_D + C_D\theta_c \ , \tag{C.4}$$

the values of this ratio for a C_D of 0.25 are plotted in Fig. C.2. The corresponding values of the EIR modification factor, $f_6(\theta_c)$ can be determined by the relation

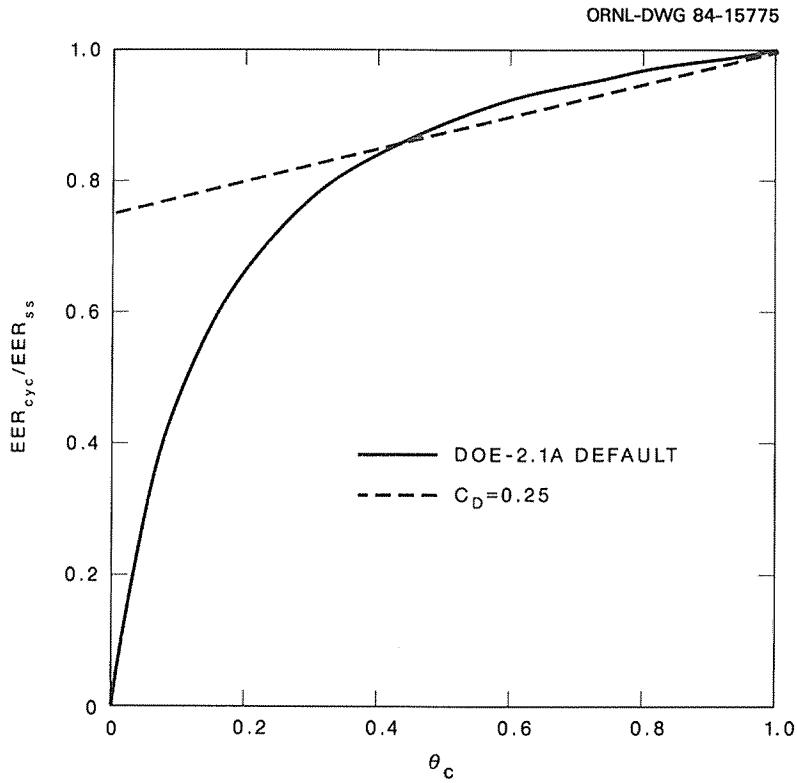

Fig. C.2. Values of ratio of cycling EER to steady state EER. EER is the energy efficiency ratio; DOE-2.1A is the building energy use program, C_D is the degradation coefficient, and θ_c is the part cooling rate ratio.

$$f_6(\theta_c) = \frac{\theta_c}{1 - C_D + C_D\theta_c} \quad . \tag{C.5}$$

Values of this factor for a C_D of 0.25 are plotted in Fig. C.1.

For the curves of EER_{cyc}/EER_{ss} presented in Fig. C.2, the curve assuming a C_D of 0.25 is typical of that for a unit having the indoor air circulation for cycling during the operation of the unit.[3] (The test procedure for determining C_D turns off the indoor fan when the compressor is turned off.[2])

The influence of the choice of the relation for $f_6(\theta_c)$ on the calculated SEER values is shown in Table C.1. The calculations were repeated for the prototypical houses in Minneapolis and Orlando, using the $f_6(\theta_c)$ relation based on the C_D value of 0.25. For an air conditioning unit with a rated EER of 10, there is a small increase (0.3 to 1.13) in the calculated SEER, depending on the location and the amount of cycling (oversizing) of the unit. The greatest increase was calculated for the 50%-oversized unit in Orlando.

C.3 Crankcase Heater

Although crankcase heaters are generally low-power devices (typically ~50 W for residential units), a significant portion of the unit's annual energy requirement can be used

Table C.1. Influence of cycling energy use relations and crankcase heater energy use on the SEER values (rated EER = 10)

(Units are Btu/h·W)

Description	Minneapolis	Orlando
With crankcase heater		
DOE-2.1A default cycling relation		
Properly sized unit[a]	6.54	9.52
50% oversized unit	6.12	8.46
Cycling degradation coefficient = 0.25		
Properly sized unit[a]	6.84	10.02
50% oversized unit	6.67	9.57
Without crankcase heater		
DOE-2.1A default cycling relation		
Properly sized unit[a]	9.58	9.75
50% oversized unit	8.55	8.62
Cycling degradation coefficient = 0.25		
Properly sized unit[a]	10.16	10.23
50% oversized unit	9.66	9.76

[a]Properly sized unit total capacity for the prototypical two-story house in Minneapolis is 18,000 Btu/h and for the same house in Orlando is 28,000 Btu/h.

by the heater. This is particularly true in the northern climates, where the air conditioning units tend to have lower capacities and to have shorter running times during the year.

The DOE-2.1A calculations were repeated for the prototypical houses in Minneapolis and Orlando, assuming that the air conditioning unit has no crankcase heater (setting the heater power to 0). Results of these calculations are given in Table C.1. The influence of the heater on the calculated SEER value is great in Minneapolis, but small in Orlando. These results reflect the relatively limited use of air conditioning in Minneapolis but indicate the use of the crankcase heater during much of the year in that city. The opposite is true in Orlando, where there is a very limited use of the crankcase heater. For properly sized units without crankcase heaters, the SEER is about the same as the rated EER in both cities. There is some degradation (0.5 to 1) in the SEER because of oversizing.

C.4 Conclusion

Given the nature of this study, the effects of the choice of the EIR modification function $f_6(\theta_c)$ on the conclusions of this study are minor. Replacement of the DOE-2.1A default curve for cycling degradation with one based on a C_D value of 0.25 reduces the predicted energy savings due to the air conditioning unit replacement about 5%. This is certainly within the accuracy of this study and does not affect the conclusions in Sect. 6.

Elimination of the crankcase heater increases the values of the SEER values, particularly in the northern portion of the country. Without crankcase heating, the ratio of the SEER to the rated EER for properly sized units is about 1.0 throughout the whole country. This ratio contrasts the large variation of this ratio in the country (shown in Fig. 5.3) when the crankcase heater energy use is included in the calculations.

The need of crankcase heating the entire year for air conditioning units throughout the country is beyond the scope of this study. It should be noted, however, that compressor replacement required because of damage due to liquid accumulation can be costly. The cost of energy to operate the crankcase heater is relatively low compared to the potential repair cost due to compressor damage. Ideally, the power to the air conditioning unit should be turned off during the heating season. However, the power to the unit must be turned on again a day or so before use to boil off the liquid accumulated in the compressor crankcase. However, this technique relys on the homeowner's control for protection to the unit.

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1. Lawrence Berkeley Laboratory and Los Alamos National Laboratory, *DOE-2 Reference Manual, Version 2.1A,* LBL-8706, Rev. 2, May 1981.
2. *1981 Standard for Unitary Air-Conditioning Equipment,* Standard 210, Air-Conditioning and Refrigeration Institute, 1981.
3. C. E. Bullock and W. R. Reedy, "Heat Pump Cyclic Performance and Its Influence on Seasonal Operation," pp. 7–1—7–19 in *Proceedings of 3rd Annual Heat Pump Technology Conference,* Oklahoma State University, Stillwater, 1978.

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