ROCK BED STORAGE WITH HEAT PUMP

FINAL REPORT
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ABSTRACT

The study, "Rock Bed Storage with Heat Pump," established the feasibility of mating a heat pump to a rock bed storage to effect optimal performance at the lowest cost in single family residences.

The operating characteristics of off-the-shelf components of heat pump/rock bed storage systems were studied, and the results were used to formulate configurations of representative systems. These systems were modeled and subsequently analyzed using the TRNSYS computer program and a life cycle cost analysis program called LCCA. A detailed load model of a baseline house was formulated as part of the TRNSYS analysis.

Results of the analysis involved the development of a technique to confine the range of heat pump/rock bed storage systems to those systems which are economical for a specific location and set of economic conditions. Additionally, the results included a comparison of the detailed load model with simple UA models such as the ASHRAE bin method.

Several modifications and additions were made to the TRNSYS and LCCA computer programs during the course of the study.

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ABBREVIATIONS

ARI	Air-Conditioning and Refrigeration Institute
Btu	British thermal unit
cfm	cubic feet per minute
COP	coefficient of performance
c_p	specific heat
ΔT	differential temperature
DBR	declining block rate
DHW	domestic hot water
db	dry bulb
dp	dew point
EER	energy efficiency ratio
EPRI	Electric Power Research Institute
°F	Fahrenheit degrees
FEF	free energy fraction
ft	feet
gal	gallon
GJ	gigajoule
h	film or surface thermal conductance
H	solar insolation
hr	hour
in.	inch
k	thermal conductivity
kg	kilogram
kJ	kilojoule
kw	kilowatt
kwhr	kilowatt-hour
lb	pound
LCCA	life cycle cost analysis computer program
m	meters
min	minute
psi	pounds per square inch
R	thermal resistance
RLH	room latent heat
RSH	room sensible heat
SAHP	solar-assisted heat pump
SCFM	standard cubic feet per minute
SFR	single family residence
SHR	sensible heat ratio
SPF	seasonal performance factor
sq ft	square feet
SST	saturated suction temperature
TMY	typical meteorological year
TOD	time of day
TRNSYS	Transient Simulation Computer program
TX-valve	thermostatic expansion valve
U	overall thermal transmittance
wb	wet bulb

INTRODUCTION

Heat pump/rock bed storage systems are attracting more and more attention because of their apparent potential to provide more economy and greater flexibility than conventional stand-alone heat pump systems or solar systems. Many studies, e.g. References 16, 20 and 21, have analyzed the thermal performance of heat pump/rock bed storage systems, with and without solar assistance, but none has analyzed the systems adequately from an economics standpoint. One set of conditions (location, heat load, weather, and utility rate) may favor a parallel solar-assisted heat pump (SAHP) arrangement; while another set may favor a series arrangement or a system without collectors. This project studies the problems of finding the optimum heat pump/rock bed storage system(s) for use in heating single family residences.

The first task in the project was gathering data and analyzing the characteristics of the various components of heat pump/rock bed storage systems. Based on the analyses, several system configurations were formulated as TRNSYS models.

A typical baseline house was selected for an accurate and complete building load model. The house was formulated in detail as a TRNSYS model, and the TRNSYS load subroutine was modified to account for latent cooling load.

Since one of the goals was to determine the most cost effective system, particular attention was given to the economic analysis. A new subroutine was written for TRNSYS to account properly for the cost of the energy consumed by a system.

Initially, several sizes of stand-alone heat pumps were analyzed for performance and economics in Albuquerque, New Mexico, and Madison, Wisconsin. This was followed by a look at the economic and performance advantages of combining rock bed storage and solar collectors with the heat pump.

The reader is cautioned about the mixture of dimensional units in this report. The modernized metric system utilizing the International System of Units is used in the TRNSYS computer analysis; English units are used in the heat loss calculations and other ASHRAE-related procedures.

METHOD OF ANALYSIS

Figure 1 illustrates the method that was used to establish the feasibility of mating a heat pump to a rock bed storage to effect optimal performance at the lowest cost in single family residences. This was accomplished through:

- analytical characterization of each subsystem;
- control algorithm for the entire system;
- definition of the various modes of operation;
- analysis of the thermodynamic and energy transport phenomena; and
- optimization of initial and life cycle costs.

FIGURE 1. METHOD OF ANALYSIS

The performance analysis consisted of TRNSYS computer simulations of candidate system models and those efforts which supported it. These included construction of a detailed building load model and preparation of representative weather data.

The economic analysis used the new TRNSYS utility cost subroutine (TYPE 35) and Zia Associates' life cycle cost analysis computer program (LCCA). TYPE 35 determines the amount of energy required to operate the system and computes the cost of that energy using the appropriate utility rates and rate structure. This energy cost data is subsequently input to LCCA along with other economic data to obtain life cycle costs. LCCA also calculates the payback period and rate of return for the system being evaluated.

Load Model

The two-story house shown in Figure 2 was chosen for the load model. It is Capp Homes' Virginian house plan. This house, which hereinafter is referred to as the baseline house, is typical of many of the homes being built in the United States today, evidenced by the fact that it is currently Capp Homes' biggest seller.

Both levels of the baseline house comprise approximately 1650 sq ft of living area. The first floor features a living room with a fireplace, a kitchen and dining area, a half-bath and a den. The second floor has four bedroom, two bathrooms and a walk-in closet.

For this study it was assumed the house would be built facing north, and solar collectors, if used, would be mounted on the rear roof. It would also be built with a full basement in which a rock storage bin could be located. Additionally, the house would be painted or stained a light color, and it would be occupied by a married couple with two children.

The baseline house is estimated to have a heat loss of approximately 13,800 Btu per degree day which corresponds to a $\overline{UA}$ of 537 Btu/hr⁰F. This heat loss is based on the calculations presented in Appendix A. When the heat gains due to people, lights and appliances, and solar are taken into account, the effective $\overline{UA}$ is reduced to 278 Btu/hr⁰F (528 kJ/hr⁰C).

The various heat gains and losses are discussed in the following sections describing the load model of the baseline house. TRNSYS subroutines 17, 18 and 19 together model the conditioned space using the ASHRAE transfer function method, and TYPE 23 models the domestic hot water (DHW) system. Table 1 is a representative list of the parameter values used in the models.

FIGURE 2. THE BASELINE HOUSE -- CAPP HOMES' VIRGINIAN

TABLE 1. LOAD MODEL PARAMETER DATA*

TYPE 17 (WALLS)				TYPE 18 (ROOF)				TYPE 19 (ROOM AND BASEMENT)			
PARAMETER NO.	NOMENCLATURE		VALUE	NOMENCLATURE		VALUE	NOMENCLATURE		VALUE		
	Symbol	Description		Symbol	Description		Symbol	Description			
1	Mode	Walls Combined into Single Unit?	2	N _{roof}	Roof-top Collectors? No	-1	Mode	Temperature Control	2		
2	n _s	Exterior Surface Absorptance	0.5	N _{ins}	Amt. Ceiling Insulation - 9 in.	3	V	Room Volume (m ³)	377.4		
3	e	Surface Emittance	0.8	a _s	Roof Absorptance	0.8	Rate	No. of Air Changes/Hr - Fixed Amt.	0.5		
4	N _b	No. of "b" Coefficients	4	e	Roof Emittance	0.8	A	Floor Area (m ²)	75.8		
5	N _d	No. of "d" Coefficients	3	A _s	South Roof Area (m ²)	41.6	IC	Construction Weight - Light	1		
6	A _s	South Wall Area (m ²)	50.54	A _e	East Roof Area (m ²)	4.84	CAPAC	Room Capacitance (kJ/°C)	30,000		
7	A _e	East Wall Area (m ²)	36.42	A _n	North Roof Area (m ²)	41.6	UA	Building UA (kJ/hr·°C)	494.6		
8	A _n	North Wall Area (m ²)	50.54	A _w	West Roof Area (m ²)	4.84	IBASE	Basement? Yes	1		
9	A _w	West Wall Area (m ²)	36.42	A _c	Ceiling Area (m ²)	79.0	DEPTH	Basement Height (m)	2.438		
10	T	Window Transmittance	0.64	Q	Attic Infiltration Rate (m ³ /hr)	100.0	PERIM	House Perimeter (m)	35.4		
11	N _g	No. of Window Glazings	2	U _{BE}	Collector Back & Edge Coef. (kJ/hr·m ² ·°C)	0	T _{grd}	Ground Water Temperature (°C)	11.5		
12	F _{sw}	Window Fraction of South Wall	0.165	S _s	South Roof Slope (deg)	18.4	Q _{gen}	Internal Heat Generation (kJ/hr)	0		
13	F _{ew}	East	0.109	S _N	North Roof Slope (deg)	18.4	PEPL	No. of Occupants	4		
14	F _{nw}	North	0.144				C _{min}	Minimum Capacitance of Load HX (kJ/°C)	1383		
15	F _{ww}	West	0.054				EFF	Load HX Effectiveness	1		
16	F _{ss}	Shaded Fraction of South Windows	0.0				CP	Specific Heat of Load HX Fluid (kJ/kg·°C)	1.012		
17	F _{es}	East	0.0				IFC	F _c =f(K _T)? Yes	1		
18	F _{ns}	North	0.0				M _{int gen}	Internal Moisture Generation Rate (kg/hr)	0.7		
19	F _{ws}	West	0.0				CAP _m	Room Moisture Capacitance	4416		
20	b _o	Transfer Function Coef. (Btu/hr·ft ² ·°F)	0.00509				T _s	HX Coil Surface Temperature (°C)	10		
21	b ₁		0.02644								
22	b ₂		0.00838								
23	b ₃		0.00010								
24	d ₁	(dimensionless)	-0.59602								
25	d ₂		0.08757								
26	d ₃		-0.00002								

*Values shown are for a 1.5 ton stand-alone heat pump system located in Albuquerque, New Mexico.

TYPE 17 (Walls). Mode 2 of this subroutine was used to model the exterior walls and fenestration areas. It was possible to use this option because all of the walls are assumed to be of the same construction and could be treated as a single unit in TRNSYS. This simplified the modeling task.

The walls were modeled using the transfer function coefficients for a frame wall with 3-in. insulation (Item No. 36 in Table 27 of Chapter 25, 1977 ASHRAE Handbook of Fundamentals). Thermal and physical properties of the various layers of the wall construction are shown in Figure 3. Since the walls are light colored on the outside, solar absorptance and emittance values of 0.5 and 0.30, respectively, were assumed. The effective solar transmittance through the fenestration areas of the exterior walls is 0.64. This assumes the windows and sliding glass door are doubled-glazed, framed with wood sash and unshaded.

FIGURE 3. EXTERIOR WALL CONSTRUCTION OF BASELINE HOUSE

TYPE 18 (Roof). The roof model used transfer function coefficients that are written into the program. The appropriate set of coefficients is specified by ceiling insulation thickness, which for this analysis was 9 in. This amount of insulation has an effective R-value of about 30.

A roof slope of 18.4° (4:12 pitch) was assumed for systems that do not use solar collectors. For the solar systems a collector tilt corresponding to latitude plus 10 degrees was used to achieve good heating performance. This resulted in a collector slope of 45° (1:1 pitch) in Albuquerque and 53° (4:3 pitch) in Madison. The non-collector covered areas of the roof were assumed to have a solar absorptance of 0.8 and emittance of 0.8.

An attic infiltration rate of $100 \text{ m}^3/\text{hr}$ was assumed. This corresponds to 0.069 cfm/ft^2 of ceiling area.

TYPE 19 (Room and Basement). The interior space of the house and the basement were modeled using Mode 2 of this subroutine to take advantage of the detailed and realistic simulation of the load-supply interaction. This option, known as temperature level control, is ideally suited for load models where the performance of a heating and cooling source component is strongly dependent on the return temperature from the load or the recent on-off history of the source. Such is the case with a heat pump/rock bed storage system because temperature profiles in storage are very dependent on the temperature and flow rate of the air passing through the rock bed. The room temperature calculated using temperature level control is a function of all the heat gains and losses from the room as well as heat transferred to or from the room by the heating and cooling system.

The transfer function coefficients in TYPE 19 are specified by the weight of construction. For this study, the coefficients were modified to account for the fact that a portion of the energy transferred into the conditioned space is lost to the surroundings and therefore does not contribute to the space cooling load. The portion that is lost to the surroundings is a function of the thermal conductance between the space air and the surroundings. The construction weight of the baseline house was considered light, which according to ASHRAE is "a frame exterior wall and 2-in. concrete floor slab with a mass of approximately 30 lb of material per sq ft of floor area." The total mass of the baseline house, including a portion of the basement, was assumed to be 80,000 lb. This alone provides a thermal capacity of $16,000 \text{ Btu/}^{\circ}\text{F}$ ($30,000 \text{ kJ/}^{\circ}\text{C}$). Any storage that may be part of a heating and cooling system will add to this amount.

The total space heating or cooling load calculated in TYPE 19 consists of the sensible loads due to conduction and solar heat gain from TYPE 17 (exterior walls and fenestration areas) and TYPE 18 (roof) plus:

- the sensible heat transferred through the basement;
- the sensible heat exchanged by infiltration;
- the sensible heat gain from lights, appliances and the occupants; and
- the latent heat gain from internal moisture generation.

The loads due to infiltration and to lights and appliances are time dependent. The other loads are not, except as they may be influenced by time dependent factors such as sunlight and temperature.

The infiltration rate profile shown in Figure 4 was assumed. It is based on the house being well-sealed, caulked and well-insulated. It ranges between 0.5 and 1.0 air changes per hour depending on the time of day. The peak amounts occur during the normal waking hours as a result of movement in and out of the house, cooking, clothes drying, and other activities that require outdoor air.

Figure 5 shows the electric light and appliance profile that was used. It is based on the electric demand curves for all-electric, residential customers presented in Ref. 1. The total daily consumption is 27 kwhr. This results in annual nonheating load of approximately 3.5×10^7 kJ.

TYPE 23 (DHW). The analysis used the DHW usage profile that was put together by the Rand Corporation for a typical SFR (Ref. 26). The profile is shown in Figure 6 as it applies to the baseline house where the total daily DHW consumption is assumed to be 100 gal at a delivery temperature of 120 °F.⁷ This results in an annual DHW heating load of approximately 2.6×10^7 kJ.

FIGURE 4. DAILY INFILTRATION RATE PROFILE

FIGURE 5. DAILY ELECTRIC LIGHT AND APPLIANCE LOAD PROFILE

FIGURE 6. DAILY DOMESTIC HOT WATER USAGE PROFILE

Weather Data

The Typical Meteorological Year (TMY) weather and solar data compiled by Hall, et.al. at Sandia Laboratories was used in the analysis (Ref. 21). This data which currently exists for 26 U. S. cities including Albuquerque and Madison was generated using statistical techniques.

A TMY is comprised of specific calendar months selected from the entire recorded span (typically from January 1953 through December 1975) for a given location as "the most representative, or typical, for that location and month." For example, the January data selected for TMY is the one which is most nearly like the composite of all 23 Januarys. This procedure is followed for each month of the year to arrive at the TMY. Table 2 lists the particular months comprising the TMY for Albuquerque and Madison:

TABLE 2. WEATHER AND SOLAR DATA COMPRISING TYPICAL METEOROLOGICAL YEAR

LOCATION	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
ALBUQUERQUE	1958	1953	1965	1966	1964	1969	1957	1954	1967	1965	1959	1954
MADISON	1965	1960	1972	1964	1953	1957	1973	1963	1958	1974	1965	1954

Computer Programs

Three computer programs were used in this study. They are SOLMOD -- a Zia Associates' program written to reformat SOLMET weather data; TRNSYS -- a solar energy system simulation program developed by the University of Wisconsin Solar Energy Laboratory; and LCCA -- a Zia Associates' life cycle cost analysis program.

SOLMOD. Program SOLMOD was used to prepare the TMY weather data for use with TRNSYS. The TMY data was obtained on magnetic tape from the National Climatic Center in the SOLMET format (Ref. 21), and as such was incompatible with TRNSYS because

- 1) SOLMET data is blocked with 24 records per block which precludes hour-by-hour readings of the information, and
- 2) SOLMET solar data is recorded as 9999 during night hours.

SOLMOD corrects these problems by

- 1) reading selected data from the SOLMET tape and writing it in proper format on another tape, and
- 2) converting the 9999 solar data to 0000.

SOLMOD also prints the data in a compact form. A listing of SOLMOD is presented in Appendix B.

TRNSYS. Version 9.2 of TRNSYS was used. This program models the individual components of a solar system in mathematical terms and connects them together by information flow representing ducts, pipes or wires. This modular simulation technique reduces the complexity of system simulation by reducing a large problem into small problems which can be solved independently. TRNSYS solves for the transient performance of the various components in response to meteorological data which are time-varying forcing functions.

Several changes were made to TRNSYS to adapt it to the needs of this study and to the CDC 6400 computer. The changes consisted of modifications to subroutine types 3,17,18,19 and 20 and two new subroutines -- TYPE 34 and TYPE 35. Brief descriptions of the changes are presented in the following paragraph. Listings of the modified subroutines are shown in Appendix C; the new subroutines are listed in Appendix D.

TYPE 3 (Pump and Fan) -- modified to account for the electrical energy consumed and the heat added to the flow stream by the pump or fan. The TRNSYS Manual documentation was changed to reflect these modifications.

TYPE 17 (Walls) -- added 0.001 to the variable TIME in line 47. This had to be done before the integer of TIME was computed because of the way CDC 6400 computers compute integers from real variables.

TYPE 18 (Room) -- required the same modification as TYPE 17, except in this case 0.001 was added in line 25.

TYPE 19 (Room and Basement) -- added the capability to calculate latent cooling load. The University of Wisconsin Solar Energy Laboratory supplied a listing of the changes that had to be made. To implement these changes required changing input Q_{PEPL} (time dependent heat gain due to people) to input Q_{infil} (time dependent air infiltration).

TYPE 20 (Heat Pump) -- modified as follows:

- changed the output from energy rates to mass flow rate and temperature representing the flow stream across the inside coil. This information was needed as input data to TYPE 19 (the space load). As a result of this change, the temperature and flow rate entering the coil are inputs 7 and 8, and the temperature and flow rate leaving the coil are outputs 17 and 18.
- increased variable NSTK from four to six in line 20 to prevent the heat pump from sticking before the thermostat controlling it had stuck.
- set variable INF (9) equal to one so the heat pump would be called every time step.
- replaced the call to subroutine DATA with appropriate heat pump performance equations. See Appendix G for a listing of these equations for the high efficiency heat pump.

TYPE 34 (Relative Humidity) -- calculates the relative humidity of air given the dry bulb and dew point temperatures. TYPE 34 was necessary because SOLMET weather tapes do not contain relative humidity data, only dry bulb and dew point temperatures.

TYPE 35 (Utility Cost) -- computes the utility cost of supplying the energy needs of the load. This is accomplished by first integrating and summing the electrical energy consumed by the various components (similar to a watthour meter), and then at specified intervals computing the utility bill using the appropriate utility rates. TYPE 35 is necessary to handle the complex rate structures used by most utilities. Complete documentation of TYPE 35 suitable for the TRNSYS Manual is included in Appendix D.

A one-year TRNSYS simulation of a stand-alone heat pump system approximately 700 to 800 central processor seconds on a CDC 6400 computer.

LCCA. This program was used to analyze the economics of the candidate systems and to compare their relative cost effectiveness. LCCA uses the annual energy cost generated by TRNSYS TYPE 35 (utility cost) and other economic data (discussed in the next section) to calculate the year-by-year costs, present worth of the costs and life cycle costs of the system. LCCA compares the systems on the basis of rate of return on investment and payback period.

LCCA incorporates a life cycle cost methodology developed by the Solar Energy Research Institute (Ref. 15). This cost methodology consists of finding the present worth, using the discount rate of the yearly system costs and summing these present worths over the period of analysis. The sum, which is called the life cycle cost, is converted into a levelized annual stream by multiplying by the capital recovery factor. This levelized annual stream can then be converted into a cost per energy value by dividing by the energy delivered to or absorbed from the space.

A detailed description of LCCA is presented in Appendix E along with a listing of the program and a sample printout. LCCA requires very little computer time. A typical run takes only 2 to 3 seconds on a CDC 6400 system.

Economic Assumptions

Since economics are very important in determining the optimum configuration of a heat pump/rock bed storage system, considerable effort was spent developing tools and techniques that could be used to compare the economic worth of candidate systems. In addition to the economic computer programs (TRNSYS TYPE 35 and LCCA) discussed in the previous section, economic data was compiled for the analysis. These data are discussed in the following paragraphs.

Equipment Costs. Heat pump costs vary widely as indicated by the Carrier data presented in Table 3. The four units shown are all commercially available and would be expected to have a common application, since each has a capacity of approximately 3.0 tons. However, their costs (list prices) are quite different, ranging from a low of \$2772 for the builder's model to \$4019 for a three-piece unit with a two-speed compressor. This cost differential represents a difference of more than 45 percent.

The costs of the components that go into making up a quality 3-ton, split-system, air-source heat pump (similar to the 38RQ or 38HQ units listed in Table 3) are shown in Table 4. Heat exchange coils (indoor and outdoor) and controls are the most expensive items in the system.

TABLE 3. HEAT PUMP SYSTEM COST COMPARISON*
(Carrier Split-System, Air-Source Units)

DESCRIPTION	TWO PIECE UNIT		THREE PIECE UNIT	
	38CQ Series	38RQ Series	38HQ Series	38TQ Series
Type of Unit	Builder's Model	Deluxe Model	Single-speed Compressor	Two-speed Compressor
<u>Subassembly Model Numbers:</u>				
Outdoor section	38CQ039	38RQ040	38HQ960	38HQ960
Compressor section	N/A	N/A	38HQ140	38TQ140
Indoor section (fan/coil)	40FS160/28MQ042	40FS160/28MQ042	40FS160/28MQ042	40FS160/28TQ042
Auxiliary heater section (20 kw)	40FQ916041	40FQ916041	40FQ916041	40FQ916041
<u>Capacities:**</u>				
Cooling				
Btu/hr @ 80°F db/67°F wb (95°F condenser)	38,000	40,000	40,000	42,000
EER (Btu/hr per watt)	6.8	7.8	8.0	8.1
FEF (%)	49.7	56.5	57.3	57.7
Heating (High Temperature)				
Btu/hr @ 47°F db/43°F wb (70°F condenser)	41,000	44,000	45,000	46,000
COP	2.6	2.7	2.9	2.9
FEF (%)	60.9	62.8	65.1	65.1
Heating (Low Temperature)				
Btu/hr @ 17°F db/15°F wb (70°F condenser)	21,500	23,500	24,000	26,000
COP	1.7	1.9	2.1	2.0
FEF (%)	40.5	47.7	51.6	48.8
<u>Costs:</u>				
Outdoor section	\$1697	\$2046	\$ 795	\$ 795
Compressor section	----	----	1328	1945
Indoor sections: Fan	269	269	269	269
Coil	261	261	261	461
Auxiliary heater section (20 kw)	304	304	304	304
Accessories (excluding refrigeration lines)	241	241	198	245
List Price	2772	3121	3155	4019
Dealer Cost (single lot) x 0.5	1308	1472	1489	1896
Dealer Cost (carload) x 0.4	1046	1178	1191	1517

* Compiled by R. E. Jones, heat pump consultant.

** Based on performance data published in ARI Directory, July 1 - December 31, 1978.

TABLE 4. COMPONENT COST BREAKDOWN FOR A 3-TON AIR-SOURCE HEAT PUMP

COMPONENT	DESCRIPTION	COST**
Compressor	Sundstrand H20B243AB	\$ 138.50
Outdoor coil	Carrier 38HQ940	307.00
Indoor coil	Carrier 40AQ024331FE with auxiliary heater section	247.50
Reversing valve	Alco 401RB3F46	31.41
TX-valve	Alco TCLE2HW	21.48
Accumulator	Tecumseh TA-1607	19.34
Filter/drier	Alco BFK-164S	9.50
Crankcase heater	Gould CH-100	9.50
High pressure controller	Penn P20 DB-1	7.20
Low pressure controller	Penn P20 DB-4	6.89
Sight glass	Alco AMI-1554	5.41
Access valve	Shreader (C & D) T-36	1.55
External equalizer	1/4-in. x 4 ft @ 20¢/ft	.80
Cabinet	---	65.00
Equipment pad and frame	Carrier frame	26.67
Controls	---	300.00
Hard-start kit	Start and run capacitor and run relay	20.71
Refrigeration lines:		
• 7/8-in. OD x 100 ft	Copper tubing @ 80¢/ft, insulation @ 87¢/ft and 2 fittings @ \$4.00 ea.	175.00
• 1/2-in. OD x 100 ft	Copper tubing @ 40¢/ft, insulation @ 53¢/ft and 2 fittings @ \$4.00 ea.	101.00
Misc. copper fittings	---	13.20

* Compiled by R. E. Jones, heat pump consultant.

** Estimated single lot cost to a HVAC dealer.

The installed cost of a heat pump can be estimated using the equation shown in Figure 7, i.e.

$$\text{Installed Cost (1979 \$)} = 3450 \left[\frac{\text{Capacity in Tons}}{3} \right]^{0.9}$$

This equation is based on the work of Christian (Ref. 5). It has been updated from 1976 dollars using an inflation rate of 6 percent per annum. Operating and maintenance costs for the heat pump are estimated to be 5 percent of the installed cost. This allows for a maintenance program consisting of periodic filter changes and lubrication as well as for abnormal repairs.

The cost of installing a rock bed storage can be estimated using the following equation developed by Kahan (Ref. 37).

$$\text{Installed Cost (1979 \$)} = \$2000 \left[\frac{\text{Mass in Tons}}{16} \right]^{2/3}$$

Figure 8 is a plot of this equation.

A cost summary of different sizes of air-to-air heat pumps and rock bed storage is presented in Table 5. Table 5 also shows the costs that would be incurred in purchasing and installing the necessary flow control dampers and electronic controller for a heat pump/rock bed storage system. It should be noted that these costs are for new construction. System costs for retrofit projects will be 10 to 25 percent higher (Ref. 36).

TABLE 5. HEAT PUMP/ROCK BED STORAGE EQUIPMENT COSTS

COMPONENT DESCRIPTION	INSTALLED COSTS (1979 \$)
Split Systems Air-to-Air Heat Pump: 1.5 Ton	\$1850
2.0 Ton	2400
2.5 Ton	2930
3.0 Ton	3450
Rock Bed Storage: 7500 kilograms	1250
15,000 "	2050
30,000 "	3250
Flow Control Dampers: consists of a pair of louvered dampers and an actuator motor	75/set
Electronic Controller: consists of four differential temperature circuits and two temperature sensors	300

FIGURE 7. INSTALLED COST OF A STAND-ALONE AIR-TO-AIR HEAT PUMP

FIGURE 8. INSTALLED COST OF ROCK BED STORAGE

Utility Rates. Two types of electric rates are available to residential customers in Albuquerque and Madison (Tables 6 and 7, respectively). They are the declining block rate (DBR) and the time-of-day (TOD) rate. Each uses a different procedure to recover utility costs which typically are categorized as customer costs, demand costs and energy (fuel) costs.

The DBR structure charges for electric service on the basis of blocks of kilowatt-hours (kwhr) used. The first block covers customer and energy costs; the next set of blocks charges for demand and energy costs; and the last block covers mostly energy costs. As the name implies the rate per unit kwhr decreases with increasing blocks of kwhr.

The TOD rate structure charges for customer and demand costs in the first billing item and for energy costs on the basis of kwhr consumed. TOD rates change as a function of time, typically time of day, time of week, time of year (season) or a combination of all three. Lower off-peak rates are charged during periods when the utility has a reduction in load or load valley, and on-peak rates apply during periods when the utility's power system is likely to reach a peak load. The rationale behind the TOD rate structure is to offer an incentive to the customer(s) to use power during the utility's off-peak times which in effect increases the utility's load factor (efficiency of plant use).

Life Cycle Costs. Life cycle costing is often used to make alternate energy-source decisions. These are made on the basis of total life cycle cost, or life cycle cost as a levelized stream, or life cycle cost per energy unit delivered. This technique is used because alternate energy systems generally have high front-end (initial) costs and low energy costs, and therefore should not be compared solely on the basis of first cost. The key to successful life cycle costing is to select realistic economic conditions. The inflation rates (fuel and general), interest rates and discount rates are particularly important in the analysis.

Table 8 lists the baseline economic conditions assumed for this study. The assumptions include fuel inflation rates based on the "high case" scenario (7.5 percent for the first 10 years and 5.8 percent for the next 10) defined by the Electric Power Research Institute (EPRI) in Supply 77 (Ref. 38). The high case represents a median condition between two other scenarios -- a "base case" in which fuel inflates at a rate of 6.4 percent for the first 10 years and 5.7 percent for the next 10; and a "worst case" in which the inflation rates are 9.2 percent and 5.9 percent for the two 10-year periods.

A 10 percent discount rate was used to discount future recurring costs to present value. This rate, which determines the annual fixed charge rate applied to amortization of the initial capital costs, was specified as an upper limit by DOE (Ref. 38). DOE's lower limit of 6 percent was also used in the sensitivity analysis discussed in the Results section of this report.

TABLE 6. ALBUQUERQUE'S RESIDENTIAL ELECTRIC UTILITY RATES
(Public Service Company of New Mexico)

DESCRIPTION	MONTHLY RATES
<u>Declining Block Rate:</u> (Underground)	
Customer Charge (single phase)	\$2.60
Consumption:	
First 200 kwhr	3.2¢ per kwhr
Next 600 kwhr	2.7¢ per kwhr
All additional kwhr: Nov - May	1.8¢ per kwhr
June- Oct	2.0¢ per kwhr
Fuel Cost Adjustment (including cost of service adjustment)	2.598¢ per kwhr
<u>Time-of-Day Rate:</u> *	
Fixed Charges:	
System charges (single phase)	\$ 4.00
Facilities charges (10-20 kw)	12.00
Meter charge	5.00
Consumption:	\$21.00
Winter (Dec-Feb): on-peak -- 8 a.m. - 10 p.m. weekdays only	4.6¢ per kwhr
off-peak -- 10 p.m. - 8 a.m. plus weekends	1.9¢ per kwhr
Summer (June-Aug): on-peak -- 11 a.m. - 7 p.m. weekdays only	6.7¢ per kwhr
off-peak -- 7 p.m. - 11 a.m. plus weekends	2.0¢ per kwhr
Other months: on-peak -- 8 a.m. - 10 p.m. weekdays only	3.9¢ per kwhr
off-peak -- 10 p.m. - 8 a.m. plus weekends	2.2¢ per kwhr
Fuel Cost Adjustment and Cost of Service Adjustment	None**
* A proposed rate under consideration by the New Mexico Public Service Commission.	
** Folded into consumption rates at 1977 rates.	

TABLE 7. MADISON'S RESIDENTIAL ELECTRIC UTILITY RATES
(Madison Gas and Electric Company)

DESCRIPTION	MONTHLY RATES	
	Winter (Oct 15-June 15)	Summer (June 15-Oct 15)
<u>Declining Block Rate:</u>		
Customer Charge	\$ 2.00	\$ 2.00
Consumption:		
First 100 kwhr	3.60¢ per kwhr	4.20¢ per kwhr
Next 400 kwhr	3.50¢ per kwhr	4.20¢ per kwhr
Next 500 kwhr	3.20¢ per kwhr	4.20¢ per kwhr
All additional kwhr	2.75¢ per kwhr	4.20¢ per kwhr
Fuel Cost Adjustment	0.31¢ per kwhr	0.31¢ per kwhr
<u>Time-of-Day Rate:*</u>		
Customer Charge (meter charge)	\$ 5.00	\$ 5.00
Consumption:		
On-peak - 7 a.m. - 7 p.m. weekdays only	5.86¢ per kwhr	8.97¢ per kwhr
Off-peak - 7 p.m. - 7 a.m. plus weekends	1.43¢ per kwhr	1.43¢ per kwhr
Fuel Cost Adjustment	0.24¢ per kwhr	0.24¢ per kwhr
<p>* A rate offered by Wisconsin Electric Power Company to residential customers on the outskirts of Madison. Currently, this rate is mandatory for homes with a consumption greater than 37,000 kwhr/year; it will soon be mandatory for homes which have a consumption in excess of 20,000 kwhr/year. Note: The baseline house for this study with its annual consumption of more than 25,000 kwhr would fall in this category.</p>		

TABLE 8. LIFE CYCLE COST ANALYSIS DATA

INPUT DESCRIPTION	SOURCE OF DATA	INPUT VALUES	
		ALBUQUERQUE	MADISON
Term of analysis	Estimate of system life	20 years	20 years
Term of loan	Typical house mortgage	20 years	20 years
Interest rate on loan	Typical house mortgage	10.0%	10.0%
Number of fuel inflation periods	EPRI Supply 77 (Ref. 38)	2	2
Length of first fuel inflation period	EPRI Supply 77 (Ref. 38)	10 years	10 years
Length of second fuel inflation period	EPRI Supply 77 (Ref. 38)	10 years	10 years
Fuel inflation rate, first period	EPRI Supply 77 (Ref. 38)	7.5%	7.5%
Fuel inflation rate, second period	EPRI Supply 77 (Ref. 38)	5.8%	5.8%
Discount rate	Alternative investments available to typical homeowner	10.0%	10.0%
General inflation rate	EPRI Supply 77 (Ref. 38)	5.0%	5.0%
Property tax appraisal rate	Local assessor's office	100%	100%
Property tax assessment rate	Local assessor's office	33 1/3%	100%
Property tax mill levy	Local assessor's office	63.090	24.153
Sales tax rate (state plus local)	State tax office	4.25%	4.00%
Owner income tax rate	State tax office and federal tax table	20.6%	24.7%
Heat pump cost (installed)	Unitary Air-to-Air Heat Pumps (Ref. 5)	\$1850 (1.5 ton)	\$1850 (1.5 ton)
Down payment on system	Typical house mortgage	10.0%	10.0%
Heat pump annual maintenance costs	Unitary Air-to-Air Heat Pumps (Ref. 5)	5.0% of installed cost	5.0% of installed cost
Tax credit (solar systems only)	Energy Tax Act of 1978 (Ref. 30)	*	*
* 30% of the first \$2000 plus 20% of the next \$8000.			

COMPONENT DEFINITION

A study of the operating characteristics of the principal components of a heat pump/rock bed storage was made to determine how the performance of each is affected by incorporation into the coupled system. The components studied included the heat pump, rock bed storage and flow control dampers.

Heat Pump

System configuration has a significant effect on the operation of a heat pump when coupled to rock bed storage. Whether storage is located on the ambient side of the heat pump or on the load side is an important consideration. Load-side storage, for example, can subject the indoor coil to temperatures as high as 95°F during the heating mode and as low as 50°F during cooling when there is flow through storage. Conventional off-the-shelf heat pumps can not operate reliably under these conditions. They are usually designed to operate with a relatively fixed air temperature entering the indoor coil (typically 68°F in the heating mode and 77°F in the cooling mode). Operation at temperatures other than these will cause excessive condenser pressures which will damage the compressor.

A generic, split-system heat pump made up of off-the-shelf components was designed specifically for this study to handle the extreme environments imposed by load-side storage. This heat pump, which hereinafter is referred to as the high efficiency unit, is comprised of the following 1.5 ton components:

Compressor - Sunstrand model H20B193AB, heat pump rated

Outdoor coil - generic unit with a heat exchange effectiveness ($\overline{hA}$) of 1707 Btu/hr °F, an air flow rate of 1600 cfm and fan motor power consumption of 350 watts.

Indoor coil - generic unit with an $\overline{hA}$ of 1050 Btu/hr °F, a flow rate of 900 cfm and power consumption of 300 watts in the "blow-through" configuration.

Expansion device - a thermostatic-expansion (TX) valve with an assumed constant 6 °F superheat leaving the evaporator.

Controls - 90 watts including crankcase heater.

The performance characteristics of these components along with an estimate of pressure losses in the refrigerant lines and reversing valve were used to calculate the high efficiency heat pump performance data presented in Appendix F.

Figure 9 compares the performance (capacity and COP) of the high efficiency heat pump with that of a commercially available unit of comparable size. Overall the high efficiency unit performs better. The only exception occurs at an outdoor temperature of 95°F where the commercial unit's cooling capacity peaks. The peak results from the use of a capillary tube. Performance degrades on both sides of 95°F because this fixed-orifice device provides insufficient pressure drop at lower temperatures and too much pressure drop at higher temperatures.

Figure 10 shows how the high efficiency unit would perform when operating with load-side storage. Both heating and cooling performance decrease as the air temperature entering the indoor coil (storage temperature) moves further away from the design temperatures of 68°F heating and 77°F cooling.

The performance of a heat pump/rock bed storage system can also be improved through the use of variable capacity heat pumps. Two-speed compressors and unloading compressors provide the capability. Another technique would be to control the compressor speed so a constant differential temperature is maintained between the entering source temperature and the exiting refrigerant saturation temperature. This would require additional and more complex refrigeration controls.

FIGURE 9. PERFORMANCE COMPARISON BETWEEN THE HIGH EFFICIENCY HEAT PUMP AND A COMMERCIALY AVAILABLE UNIT

FIGURE 10. EFFECT OF VARIABLE INDOOR COIL TEMPERATURE ON HEAT PUMP PERFORMANCE
(Characteristic of Load-Side Heat Pump/Rock Bed Storage System)

Rock Bed Storage

Rock beds used in heat pump systems will be similar in many respects to those used in conventional air-based solar systems. The design criteria are basically the same with both systems placing a premium on storage capacity, temperature stratification and minimal flow resistance. The criteria, for example, that govern the placement and construction of storage are the same for both. These criteria are, in addition to minimizing cost,

- to locate components close to each other to minimize the length of connecting ductwork.
- to locate storage indoors if possible to protect against weathering and in an area that otherwise would be unused (e.g. in the basement or crawl space). A utility room and garage are other possible locations, but space in these areas is generally more valuable for other purposes.
- to construct storage to minimize structural loading -- from the storage weight and from pressures exerted on the walls by thermal expansion of the rocks.
- to seal and insulate storage to minimize heat loss and to keep water out, particularly if located outdoors. Water can damage the storage container and possibly the storage medium, as well as degrade the performance of the insulation.

Moisture condensation inside storage is a potential problem for rock beds used in heat pump systems. During the cooling season when the rock bed is being used for cold storage, condensation will occur when the air temperature in the rock bed is below the dew point. Calculations were made to determine the magnitude of this problem, and the results which are presented in Appendix G indicate that there will not be enough moisture deposited in storage to form droplets. However, water vapor can degrade the performance of the insulation and could, over a long period of time, damage the container and/or the storage medium. To protect against moisture damage, the following guidelines should be followed:

- construct the container of concrete;
- use waterproof insulation (vapor barriers and closed-cell foam insulation); and
- use rocks such as granite which do not react with water. Note: Appendix H describes why granite is also the best storage medium from a heat transfer standpoint.

Pressure drop in storage is another factor requiring special consideration in heat pump/rock bed storage systems. Heat pumps require high air flow rates (typically 700 to 900 cfm per ton of refrigeration), and this means a large supplemental power consumption if the resistance to air flow through the rocks is large. A pressure drop between 0.10 and 0.15 in. of water is recommended to ensure obtaining comparable pressure drops through each of storage, heat pump coils and the distribution ductwork, resulting in a total system pressure drop of not more than 0.5 in. of water. To achieve uniform distribution of air in the storage plenums, most of the storage pressure drop should occur in the rock bed itself with only a small amount occurring in the inlet and outlet plenums.

Uniformity of rock size is an important parameter in minimizing rock bed pressure drop. If the rock sizes are not uniform, the smaller rocks will fall into spaces between the larger ones, and reduce the void fraction (cross-sectional flow area) and increase the resistance to air flow. This problem becomes worse with time. The combined effects of gravity and thermal expansions and contractions cause rock to pack tighter. This alters the air flow path through storage and enhances the possibility of the air "short-circuiting" storage i.e., across the top in side-to-side storage containers or along the outer walls in top-to-bottom containers. In addition to using uniform rock sizes, packing can be minimized by partitioning the rock bed into layers, with wire mesh.

Flow Control Dampers

Well-sealed flow control dampers are an essential part of any heat pump/rock bed storage system. Damper leakage results in poor system performance, particularly during storage heating when leakage to the outdoors results in a direct loss of heat and leakage indoors increases the cooling load of the structure and adds to the discomfort of the occupants.

Well-sealed dampers have not been used in the past in residential forced-air heating systems, because the typical house furnace operates only when heat is needed and any leakage is generally to the heated space. In this case, the leakage can be considered useful heat. As a result, the accepted practice has been to install dampers such that "leaks are not audible and cannot be felt by hand". (Ref. 34).

This standard is not acceptable for heat pump/rock bed storage systems, because it can result in leakage rates as high as 15 percent of the total flow volume. Leaks of this magnitude can severely degrade the overall performance of the system, particularly if the leaks contribute to thermoisphoning through storage and heat loss to the outdoors.

SYSTEMS

Description, Analysis and Performance Results

A compilation of all known configurations of heat pump/rock bed storage systems (including SAHP systems) was made, and each was catalogued according to type. In the case of the SAHP systems this involved identifying each as being either a series, parallel or multi-source system. The important features and modes of operation of each system were also identified. From this list, one of each of the following configurations was selected for further analysis:

- stand-alone heat pump
- heat pump/rock bed storage (ambient-side)
- heat pump/rock bed storage (load-side)
- solar-assisted heat pump/rock bed storage

Complete descriptions of these systems (including a schematic and TRNSYS information flow diagram of each) are presented later in this section.

Baseline System

Before any performance analyses were performed, the baseline house was sized for a stand-alone heat pump system using the performance data shown in Table 9. This system, known as the baseline system, was sized in accordance with accepted industry procedure, i.e. just large enough to handle the peak instantaneous cooling load. Whatever heating capacity this allows is acceptable, and if additional heat is required to handle the peak heating load, the difference is made up with supplementary resistance heat. This procedure ensures the best comfort control.

The rationale behind this procedure is that a heat pump sized to handle the peak heating load will in most cases result in an oversized unit for cooling. An oversized unit will cycle on and off frequently, which is hard on the equipment, and will provide poor humidity control during the cooling cycle. A complete discussion on the effects of undersizing and oversizing heat pumps relative to heating and cooling is presented in Appendix I.

As described in Appendix J, the baseline system turned out to be a 1.5 ton unit in both Albuquerque and Madison. The amount of electrical resistance heat required to supplement this size of unit was determined from the balance point charts shown in Figure 11. The solid line in these charts represents the heating requirements of the house; the dotted lines indicate the heating capacities of the equipment. Balance points occur where the solid and dotted lines cross. The charts show that 9.6 kw of electric resistance heat is required to supplement the 1.5 ton heat pump in Albuquerque where the design outside temperature is 0°F, and 14.4 kw is required in Madison where the design temperature is -15°F.

TABLE 9. SINGER AIR-TO-AIR HEAT PUMP PERFORMANCE DATA

OPERATIONAL MODE	OUTDOOR TEMPERATURE* (°F)	MODEL H-R551018-2T (1.5 TON)		MODEL H-R551024-2S (2.0 TON)		MODEL H-R553030-2T (2.5 TON)		MODEL H-R552036-1T (3.0 TON)	
		CAPACITY (BTU/HR)	ELEC PWR (KW)	CAPACITY (BTU/HR)	ELEC PWR (KW)	CAPACITY (BTU/HR)	ELEC PWR (KW)	CAPACITY (BTU/HR)	ELEC PWR (KW)
HEATING WITH CONDENSER INLET TEMPERATURE = 68°F	-13	8,000	2.10	8,200	2.15	9,500	2.51	13,000	3.45
	-3	8,700	2.20	9,500	2.23	12,000	2.61	16,300	3.59
	7	10,000	2.30	11,500	2.31	14,800	2.75	19,600	3.70
	17**	12,000	2.40	14,000	2.45	17,500	2.90	23,000	3.85
	27	14,500	2.48	16,700	2.60	22,000	3.05	27,800	3.95
	37	17,000	2.55	20,000	2.78	26,000	3.20	33,000	4.09
	47**	19,500	2.60	23,500	2.95	30,500	3.35	39,000	4.20
	57	22,500	2.70	27,200	3.18	34,800	3.50	45,900	4.30
	67	24,500	2.75	30,000	3.40	38,500	3.61	51,000	4.40
COOLING WITH EVAPORATOR INLET TEMPERATURE = 77°F	72	16,900	2.25	21,500	2.70	29,200	3.10	35,200	3.95
	77	17,400	2.35	22,200	2.90	29,700	3.20	35,500	4.00
	82	17,900	2.45	22,700	3.00	29,500	3.27	35,400	4.20
	87	18,000	2.50	22,900	3.10	29,200	3.30	35,200	4.35
	92	18,300	2.60	23,100	3.20	28,800	3.40	35,000	4.50
	95**	18,500	2.65	23,000	3.30	28,000	3.45	34,500	4.60
	97	17,800	2.70	22,800	3.30	27,800	3.55	34,000	4.62
	102	17,000	2.75	22,200	3.35	26,700	3.64	33,000	4.70
	107	16,300	2.85	21,500	3.45	25,000	3.70	32,000	4.80
	112	15,000	2.90	20,500	3.55	23,100	3.80	30,500	4.90
	117	13,700	3.00	19,000	3.60	21,000	3.85	29,000	5.00

*Evaporator inlet temperature for heating; condenser inlet temperature for cooling.
**Certified ratings based on ARI Standard 240-75.

FIGURE 11. BALANCE POINT CHARTS FOR ALBUQUERQUE AND MADISON

TABLE 10. SUMMARY OF STUDY RESULTS IN ALBUQUERQUE

SYSTEM		HEAT PUMP		ROCK BED SIZE (KG)	UTILITY RATE STRUCTURE	CONTROL ALGORITHM	PERFORMANCE			TOTAL ENERGY CONSUMED (KWH)	1979 ENERGY COST (\$)	ESTIMATED INSTALLED COST (\$)
NO.	TYPE	TYPE	SIZE (TONS)				HEATING		COOLING SYSTEM SPF			
							HEAT PUMP COP	SYSTEM SPF				
A1	Stand-Alone Heat Pump	Singer	1.5	- - -	DBR	Figure 12	1.83	1.78	2.13	23,430	1170	1850
A2	Stand-Alone Heat Pump	Singer	2.0	- - -	DBR	Figure 12	1.98	1.95	2.21	22,920	1147	2400
A3	Stand-Alone Heat Pump	Singer	2.5	- - -	DBR	Figure 12	2.19	2.18	2.56	22,220	1116	2930
A4	Stand-Alone Heat Pump	Singer	3.0	- - -	DBR	Figure 12	2.16	2.16	2.30	22,600	1133	3450
A5	Stand-Alone Heat Pump	High Eff.	1.5	- - -	DBR	Figure 12	2.43	2.37	2.35	22,160	1114	2110
A6	Stand-Alone Heat Pump	High Eff.	1.5	- - -	TOD	Figure 12	2.43	2.37	2.35	22,160	939	2110
A8	HP/RB, Ambient-Side, Configuration A	Singer	1.5	15,000	DBR	Figure 14	1.98	1.90	1.93	23,370	1167	4350
A9	HP/RB, Ambient-Side, Configuration B	Singer	1.5	15,000	DBR	Figure 16	1.85	1.70	1.87	23,870	1190	4350
A10	HP/RB, Ambient Side, Configuration B	Singer	1.5	15,000	DBR	Figure 16	1.86	1.76	1.87	23,840	1187	4350
A11	HP/RB, Ambient-Side, Configuration B	Singer	1.5	15,000	DBR	Figure 16	1.86	1.77	1.88	23,590	1177	4350
A17	HP/RB, Load Side	High Eff.	1.5	15,000	TOD	Figure 18	2.39	2.27	2.42	22,646	913	4350
A18	HP/RB, Load Side	High Eff.	3.0	15,000	TOD	Figure 18	2.46	2.43	2.42	22,425	893	6430
A20	HP/RB, Load Side	High Eff.	1.5	15,000	TOD	Figure 18	2.39	2.24	2.42	22,701	914	4350
A23	HP/RB, Load Side	High Eff.	1.5	15,000	TOD	Figure 18	2.39	1.50	2.59	25,556	963	4350
A27	HP/RB, Load Side	High Eff.	1.5	3,750	TOD	Figure 18	2.41	2.32	2.37	22,192	923	3360
A28	HP/RB, Load Side	High Eff.	1.5	7,500	TOD	Figure 18	2.40	2.30	2.38	22,404	919	3810
A29	HP/RB, Load Side	High Eff.	1.5	30,000	TOD	Figure 18	2.39	2.17	2.44	23,132	917	5810

TABLE 11. SUMMARY OF STUDY RESULTS IN MADISON

SYSTEM		HEAT PUMP		ROCK BED SIZE (KG)	UTILITY RATE STRUCTURE	CONTROL ALGORITHM	PERFORMANCE			TOTAL ENERGY CONSUMED (KWH)	1979 ENERGY COST (\$)	ESTIMATED INSTALLED COST (\$)
NO.	TYPE	TYPE	SIZE (TONS)				HEATING		COOLING			
							HEAT PUMP COP	SYSTEM SPF	SYSTEM SPF			
M1	Stand Alone Heat Pump	Singer	1.5	- - -	DBR	Figure 12	1.70	1.48	2.16	30,710	1093	1850
M2	Stand-Alone Heat Pump	Singer	2.0	- - -	DBR	Figure 12	1.85	1.63	2.24	29,550	1057	2400
M3	Stand-Alone Heat Pump	Singer	2.5	- - -	DBR	Figure 12	2.01	1.86	2.68	28,260	1015	2930
M4	Stand-Alone Heat Pump	Singer	3.0	- - -	DBR	Figure 12	1.97	1.88	2.31	28,510	1024	3450
M5	Stand-Alone Heat Pump	High Eff.	1.5	- - -	DBR	Figure 12	2.33	1.92	2.42	27,770	1001	2110
M6	Stand-Alone Heat Pump	High Eff.	1.5	- - -	TOD	Figure 12	2.33	1.92	2.42	27,770	1112	2110
M7	Stand-Alone Heat Pump	High Eff.	3.0	- - -	DBR	Figure 12	2.28	2.20	2.41	26,780	971	3930
M8	HP/RB, Ambient-Side, Configuration B	Singer	1.5	15,000	DBR	Figure 16	1.57	1.39	2.12	31,540	1119	4350
M12	HP/RB, Ambient-Side, Configuration B	Singer	1.5	30,000	DBR	Figure 16	1.77	1.54	2.12	30,330	1082	5550
M13	HP/RB, Load-Side	High Eff.	1.5	15,000	DBR	Figure 18	2.36	1.94	2.61	28,400	1062	4350
M14	HP/RB, Load-Side	High Eff.	1.5	15,000	DBR	Figure 18	2.35	1.91	2.61	28,040	1012	4350
M15	HP/RB, Load-Side	High Eff.	1.5	15,000	DBR	Figure 18	2.35	1.93	2.57	27,860	1006	4350
M16	HP/RB, Load-Side	High Eff.	3.0	15,000	DBR	Figure 18	2.30	2.21	2.58	26,990	982	6430
M17	HP/RB, Load-Side	High Eff.	1.5	15,000	TOD	Figure 18	2.32	1.58	2.48	28,154	1057	4350
M18	HP/RB, Load-Side	High Eff.	3.0	15,000	TOD	Figure 18	2.42	2.18	2.44	27,030	977	6430
M19	HP/RB, Load-Side	High Eff.	3.0	30,000	TOD	Figure 18	2.24	2.16	2.44	27,319	969	7630
M20	HP/RB, Load Side	High Eff.	1.5	15,000	TOD	Figure 18	2.32	1.87	2.46	28,141	1060	4350
M21	HP/RB, Load-Side	High Eff.	1.5	15,000	TOD	Figure 18	2.31	1.79	2.46	28,675	1050	4350
M22	HP/RB, Load-Side	High Eff.	1.5	15,000	TOD	Figure 18	2.30	1.68	2.46	29,452	1038	4350
M23	HP/RB, Load-Side	High Eff.	1.5	15,000	TOD	Figure 18	2.29	1.43	2.53	31,947	1037	4350
M24	HP/RB, Load-Side	High Eff.	1.5	3,750	TOD	Figure 18	2.31	1.59	2.44	30,105	1067	3360
M25	HP/RB, Load-Side	High Eff.	1.5	7,500	TOD	Figure 18	2.30	1.49	2.45	31,037	1052	3810
M26	HP/RB, Load-Side	High Eff.	1.5	30,000	TOD	Figure 18	2.29	1.38	2.73	33,266	1044	5810

Stand-Alone Heat Pump

Two types of stand-alone heat pumps were analyzed -- a commercially available off-the-shelf unit and the high efficiency unit. The high efficiency unit also uses off-the-shelf components, but is designed to be able to take the high condenser temperatures inherent in load-side rock bed configurations. A schematic of the stand-alone heat pump systems is shown in Figure 12. Figure 13 shows a TRNSYS information flow diagram of a stand-alone heat pump.

The first analysis performed on the stand-alone heat pump involved evaluating the baseline system in Madison and Albuquerque. Four sizes of the off-the-shelf heat pump were simulated. The results are shown in Tables 10 and 11. It was found that the 1.5 ton unit was the smallest unit that could handle the cooling load in both Madison and Albuquerque. The 1.5 ton was also found to be the most cost-effective of the four conventional units. The 1.5 ton high efficiency heat pump was also simulated in Madison and Albuquerque under the declining block and time-of-day rate structure. These results are also shown in Tables 10 and 11.

As a check on the validity of the TRNSYS simulation of stand-alone heat pumps, heating Seasonal Performance Factors (SPF) generated by TRNSYS were compared to heating SPF's generated by the ASHRAE bin method. This method is described in Chapter 43 of the 1976 ASHRAE Systems Handbook. Table 12 presents a comparison of SPF's for Albuquerque computed using the two TRNSYS load models -- the simple UA model (TYPE 12) and the detailed load model (TYPES 17, 18 and 19 combined) -- and the ASHRAE bin method. The results are based on a UA value of 278 Btu/hr °F, which was determined to be the effective UA.

TABLE 12. COMPARISON BETWEEN TRNSYS AND BIN METHOD OF CALCULATING HEATING SEASONAL PERFORMANCE FACTOR

HEAT PUMP SIZE (TONS)	HEATING SPF FOR ALBUQUERQUE (UA = 278 Btu/hr·°F)			PERCENT DIFFERENCE	
	TRNSYS ANALYSIS		ASHRAE BIN METHOD*	TRNSYS DETAILED LOAD MODEL TO BIN	TRNSYS UA MODEL TO BIN
	DETAILED LOAD MODEL	UA MODEL			
1.5	1.78	1.96	1.93	8.4	1.6
2.0	1.95	2.10	2.08	6.7	1.0
2.5	2.18	2.37	2.35	7.8	0.9
3.0	2.16	2.37	2.35	8.8	0.9

*Calculated by Singer Research and Development Corporation

FIGURE 12. STAND-ALONE HEAT PUMP SYSTEM

Modes of Operation

- Heats the house from outside air.
- Cools the house from outside air.

Control Algorithm

Heating:

Heat pump compressor and blowers are on when thermostat 1st stage contacts are closed.

Cooling:

Heat pump compressor and blowers are on when thermostat 3rd stage contacts are closed.

FIGURE 13. TRNSYS INFORMATION FLOW DIAGRAM FOR STAND-ALONE HEAT PUMP SYSTEM

Note that the SPF's calculated with TRNSYS using the $\overline{UA}$ model are in close agreement with the SPF's calculated with the bin method. Since the $\overline{UA}$ model and the bin method are very similar in their calculation of the heating load ($Q = \overline{UA} (T_{in} - T_{amb})$), it is not surprising that they produce similar results. The small discrepancies between the two are probably attributable to differences between the TMY and the bin weather data.

The SPF's calculated with the TRNSYS detailed load model vary significantly from those calculated with the $\overline{UA}$ model or bin method. The detailed load model SPF is consistently lower than that calculated using the other two methods. A possible explanation for this discrepancy is that the $\overline{UA}$ type of load model calculates heating load and heat pump performance as a function of outside temperature, whereas the detailed load model calculates heat pump performance as a function of outside temperature and the heating load as a function of outside temperature, time-dependent infiltration, solar heat gain and appliance heat gain. The effect of including temperature-independent heat gains is to reduce or eliminate the heat load when the outside temperature is relatively high (above 50°F). This is the outside temperature range in which the heat pump operates most efficiently. In other words, the detailed load model shifts the heat load to periods when outside temperature is lower and the heat pump has lower performance.

The TRNSYS detailed load model is a better approximation of reality than the bin method. The bin method appears to result in unrealistically high heating SPF's for heat pumps.

Heat Pump/Rock Bed Storage

Two heat pump/rock bed storage configurations were analyzed. They are ambient-side and load-side. In the ambient-side configuration the outdoor or ambient-side of the heat pump is connected to the rock bed (Figures 14-17). The load-side configuration has the indoor side of the heat pump connected to storage. (Figure 18). These configurations were selected for analysis because they are simple (minimum number of dampers and ducts) and have the capability to reverse air flow through storage to take advantage of temperature stratification in the rock bed.

Ambient-Side Storage. The basic concept of ambient-side storage with a heat pump is that it can take advantage of the day-night temperature swing and improve the heat pump SPF. During the heating season, relatively warm daytime air can be circulated through storage to provide a source for nighttime heating. During the cooling season, cool night air can be circulated through storage to provide a better sink for daytime cooling.

Figure 14 shows the simplest configuration of ambient-side storage that can circulate outside air through storage and reverse flow through storage. This configuration (Configuration A) requires a larger heat pump outside unit blower to handle the additional pressure drop produced by the rock bed. The power consumption values in the heat pump data were modified to account for the larger blower.

FIGURE 14. HEAT PUMP/ROCK BED STORAGE SYSTEM
(AMBIENT-SIDE CONFIGURATION A)

Modes of Operation

- Heats or cools the house from outside air.
- Heats or cools the house from storage.
- Heats or cools storage with outside air (compressor is off in this mode).

Control Algorithm

Heating:

Normal damper positions: D_1 open / D_2 closed

D_3 open / D_4 closed

- Heat pump compressor and blowers on in heating mode when thermostat 1st stage contacts are closed.
- D_3 closed / D_4 open and outside unit blower on when $T_{amb} > (T_{R2} + 60^\circ\text{C})$ and thermostat 1st stage contacts are open.
- D_1 closed / D_2 open when $T_{R1} > (T_{amb} + 60^\circ\text{C})$ and thermostat 1st stage contacts are closed.

Cooling:

Normal damper positions: D_1 open / D_2 closed

D_3 open / D_4 closed

- Heat pump compressor and blowers on in cooling mode when thermostat 3rd stage contacts are closed.
- D_3 closed / D_4 open and outside unit blower on when $T_{R2} > (T_{amb} + 6^{\circ}\text{C})$ and thermostat 3rd stage contacts are open.
- D_1 closed / D_2 open when $T_{amb} > (T_{R1} + 6^{\circ}\text{C})$ and thermostat 3rd stage contacts are closed.

FIGURE 15. TRNSYS INFORMATION FLOW DIAGRAM FOR AMBIENT-SIDE HEAT PUMP/ROCK BED STORAGE SYSTEM (CONFIGURATION A)

FIGURE 16. HEAT PUMP/ROCK BED STORAGE SYSTEM
(AMBIENT-SIDE CONFIGURATION B)

Modes of Operation

- Heats or cools the house from outside air.
- Heats or cools the house from storage.
- Heats or cools storage with outside air (compressor is off in this mode).

Control Algorithm

Heating:

Normal damper positions: D_1 open / D_2 closed

- Heat pump compressor and blowers on in heating mode when thermostat 1st stage contacts are closed.
- Auxiliary blower on when $T_{amb} > (T_{R2} + 3^{\circ}\text{C})$ and thermostat 1st stage contacts are open.
- D_1 closed / D_2 open when $T_{R1} > (T_{amb} + T')$ and thermostat 1st stage contacts are closed.

Cooling:

Normal damper positions: D_1 open / D_2 closed

- Heat pump compressor and blowers on in cooling mode when thermostat 3rd stage contacts are closed.
- Auxiliary blower on when $T_{R2} > (T_{amb} + 3^{\circ}\text{C})$ and thermostat 3rd stage contacts are open.
- D_1 closed / D_2 open when $T_{amb} > (T_{R1} + T')$ and thermostat 3rd stage contacts are closed.

where $T' = 1^{\circ}\text{C}$ in system 9
 $= 6^{\circ}\text{C}$ in system 10
 $= 12^{\circ}\text{C}$ in systems 11 and 12

FIGURE 17. TRNSYS INFORMATION FLOW DIAGRAM FOR AMBIENT-SIDE HEAT PUMP/ROCK BED STORAGE SYSTEM (CONFIGURATION B)

The control algorithm for this system is also shown in Figure 14. The philosophy of the control algorithm in heating is that outside air can be circulated through storage when it is warmer than storage and storage can be a heat source for the heat pump when it is warmer than ambient. The cooling control algorithm circulates outside air through storage when the outside air is cooler than storage and uses the storage as a source for the heat pump when the storage is cooler than ambient.

Results for Configuration A of ambient-side storage are shown in Tables 10 and 11, System 8. The poor performance of this system is due to several factors. The single blower interfacing with the rock bed means that the rock bed cannot be charged while the heat pump is heating the house, which severely limits the amount of storage charging. The use of the heat pump blower for rock bed charging is inefficient because the blower has to force air through the outdoor coil even though the heat pump is not running.

As a result of these drawbacks, the system was redesigned as Configuration B (shown in Figure 16). Two dampers were replaced by a second blower which provides the capability to charge storage even though the heat pump is running. The second blower is also more efficient in charging storage, and the additional capital investment required for it is small since it replaces two dampers. The basic control algorithm for the redesigned system is the same as that for the original system. Several simulations were made with this system using various storage sizes and control algorithms (Tables 10 and 11, Systems 9 through 12).

The results are not very promising. In only one case was the annual energy cost less than that of the stand-alone heat pump. In several cases, the heating SPF was significantly improved over the stand-alone system, but the annual energy cost was increased primarily due to the power consumption of the charging blower. The one case where the energy cost was reduced was in Madison where the storage size was increased to 30,000 kilograms. The economics of such marginal performance increases with large capital investments required for the rock bed and associated equipment eliminates their economic viability.

Load-Side Storage. Load-side storage (Figures 18 and 19) was found to be more versatile than ambient-side. Load-side storage allows the rock bed to be heated or cooled for later heating or cooling of the residence, which allows the system to take advantage of utility rate structures. It also allows storage to be heated with resistance coils in series with the heat pump or alone. No modification of the heat pump blowers is necessary since the indoor unit is already designed to handle the pressure drop of a residence's duct system.

Declining Block Rates - Under the declining block rate the philosophy of the control algorithm was the same as that used for ambient-side storage. The rock bed was used to allow the heat pump to be run when the ambient temperature conditions are best for its operation. The implementation of this control philosophy cannot be accomplished by temperature control alone as in the case of the ambient-side storage. It was decided to control the "charging" (heating or cooling) of the rock on a time-of-day basis.

FIGURE 18. HEAT PUMP/ROCK BED STORAGE SYSTEM
(LOAD-SIDE CONFIGURATION)

Modes of Operation

- Heats or cools the house from outside air.
- Heats or cools the house from storage (compressor is off in this mode).
- Heats or cools storage with the heat pump or resistance coils or both.

Control Algorithm (Declining Block Rate)

Heating:

Normal damper positions: D_1 open / D_2 closed

D_3 open / D_4 closed

- D_3 closed / D_4 open and conditioner blower on when $T_{room} < T'$, $T_{R1} > 80^\circ\text{F}$, heat pump not on and time is between t_a and t_b .
- heat pump compressor and blower on in heating mode when thermostat 1st stage contacts are closed.
- resistance coil on when thermostat 2nd stage contacts are closed.
- D_1 closed / D_2 open and heat pump compressor and blower on when time is between 12:00 pm and t_c pm and the thermostat 1st stage contacts are open and $T_{R2} < 75^\circ\text{F}$.

Figure 18 (cont.)

Cooling:

- D_3 closed / D_4 open and conditioner blower on when $T_{room} > T'$ and $T_{R1} < 74^{\circ}$ and heat pump not on.
- heat pump compressor and blower on in cooling mode when thermostat 3rd stage contacts are closed.
- D_1 closed / D_2 open and heat pump compressor and blower on in cooling mode when time is between the hours of 12:00 am and 6:00 am, $T_{R2} > 65^{\circ}$ and the thermostat 3rd stage contacts are open.

$T' = 71^{\circ}\text{F}$ in system 13

$= 70^{\circ}\text{F}$ in systems 14, 15 and 16

where t_a = does not apply in systems 13 and 14;
 $= 12:00$ am in systems 15 and 16.

t_b = does not apply in systems 13 and 14;
 $= 8:00$ am in systems 15 and 16.

t_c = 6:00 pm in system 13;
 $= 8:00$ pm in systems 14, 15, and 16.

$T'' = 76^{\circ}\text{F}$ in system 13

$= 77^{\circ}\text{F}$ in systems 14, 15 and 16.

Control Algorithm (Time-of-Day Rate)

Heating:

Normal damper positions: D_1 open / D_2 closed

D_3 open / D_4 closed

- D_3 closed / D_4 open and conditioner blower on when $T_{room} < 70^{\circ}$, $T_{R1} > 80^{\circ}$ and on-peak.
- Heat pump compressor and blower on in heating mode when thermostat 1st stage contacts are closed.
- D_1 closed / D_2 open and heat pump on in heating mode when off-peak, $T_{R2} < 95$ and thermostat 1st stage contacts are closed.

Figure 18 (cont.)

- resistance coils on when:

- thermostat 2nd stage contacts are closed in systems 17-19, 27-29.
- when thermostat 2nd stage contacts are closed or when the heat pump is on between the hours of:
 - 4am and 7am in systems 20 and 21
 - 1am and 7am in system 22
 - 7pm and 7am in system 23

Cooling:

Normal damper positions: D_1 open / D_2 closed

D_3 open / D_4 closed

- D_3 closed / D_4 open and conditioner blower on when $T_{room} > 77^{\circ}$, $T_{R1} < 74^{\circ}$ and on-peak.
- heat pump compressor and blower on in cooling mode when thermostat 3rd stage contacts are closed.
- D_1 closed / D_2 open and heat pump compressor and blower on in cooling mode when off-peak, $T_{R2} > 65^{\circ}$ and thermostat 3rd stage contacts are open.

FIGURE 19. TRNSYS INFORMATION FLOW DIAGRAM FOR LOAD-SIDE HEAT PUMP/ROCK BED STORAGE SYSTEM

The basic assumption was that on most days, the ambient temperature is warmest during the afternoon (12 p.m. to 6 p.m.) and coolest during the early morning (12 a.m. to 6 a.m.). To take advantage of this diurnal temperature swing during the winter, storage was charged during the afternoon and discharged to the load during the early morning. The reverse of this was done during the cooling season. The complete control algorithm is listed in Figure 18.

Load-side results under declining block rates are shown in Tables 10 and 11, Systems 13 through 16. The effect of using a larger heat pump in this configuration was also investigated. In all cases, heat pump performance increased only slightly, and the additional load of the distribution blower caused larger utility costs than in the equivalent stand-alone system.

Time-of-Day Rates - An important possible application of heat pumps and rock beds is in taking advantage of time-of-day utility rates. Fortunately, residential time-of-day utility rates have been proposed in Albuquerque and instituted in the Madison area, and thus, provide a good test of the effects of time-of-day rates on system and control algorithm design.

The basic philosophy of the control algorithm used under the time-of-day rate is straight forward. The heat pump should be used to charge storage whenever electrical rates are low, i.e. during the off-peak period. The stored heat or cool is then used to meet the load during the on-peak period of high electrical costs.

The load-side configuration has the capability of heating the storage with electrical resistance heat. The optimum control algorithm for this mode of operation depends on the ability of the heat pump to sufficiently charge storage and the differential between on peak and off peak electrical rates. If the goal is to just save energy, then the control algorithm should be designed to minimize electrical resistance heating. If the goal is to minimize the electric utility bill, then the control algorithm, storage size, and heat pump size would be arranged so that all heating is done off-peak by the heat pump. However, if the goal is the lowest life cycle cost, then the effect of the high capital cost associated with a heat pump large enough to do all or most of the heat pump heating has to be considered. In this analysis, simulations were first made with Madison weather data and with a 1.5 ton high efficiency heat pump and no resistance coils heating storage (Table 11, System 17). It became apparent from examining the plots of storage temperature during charging, that greater heating capacity would be desirable, as the 1.5 ton heat pump was spending most of the off-peak time heating the load instead of the storage. A simulation was then run with a 3.0 ton high efficiency heat pump (Table 11, System 18). The extra heating capacity allowed the storage to charge up and the heating SPF was increased while the utility cost was decreased. However, the high initial capital cost of this system makes it uneconomic.

It was therefore decided to try using resistance coils to make up for the lack of heating capacity. The resistance coils were doubled in capacity and were turned on for progressively longer time periods during the off-peak period (Table 11, Systems 20 through 23).

Figure 20 shows the effect of using increasing amounts of resistance heat to charge storage. In Madison, the effect of using more off-peak resistance heat is to lower the annual utility cost. It should be noted that the savings become progressively smaller as more off-peak resistance heat is used, but as no capital investment is required to leave the coils on for longer periods of time, the maximum off-peak resistance is the most cost effective.

The control algorithm for the maximum off-peak resistance heat is to turn on the resistance coils whenever the heat pump is heating storage (Figure 18).

In Albuquerque, the effect of using off-peak resistance heat to charge storage is much different than in Madison. As can be seen in Figure 20, any use of off-peak resistance heat to charge storage increases the utility cost. The optimum control algorithm for Albuquerque is shown in Figure 18. The difference between the Albuquerque and Madison optimum control strategy can be attributed in part to the differences in the time-of-day rate in each location. In Madison, the winter on-peak rate is 5.68¢/kwhr and the off-peak is 1.43¢/kwhr. The ratio between these two rates is 3.97. This means that resistance coils operating off-peak in Madison have an effective economic COP of approximately 4.0 while the stand-alone, high efficiency heat pump has an average COP during January of 2.19. Therefore, in Madison with the Wisconsin Electric Power Company's time-of-day rate, it is cheaper to run a resistance coil off-peak than a heat pump on-peak.

This is not the case in Albuquerque where the ratio between the on and off-peak rates is 2.42 and the COP of the high efficiency heat pump in January is 2.44. Therefore, in Albuquerque with the Public Service of New Mexico time-of-day rate, it is probably not cheaper to run a resistance coil off-peak than it is to run a heat pump on-peak.

Once the control algorithms were optimized in the two cities, simulations were run to examine the effect of the rock bed storage size on system performance and economics (Tables 10 and 11, Systems 24 through 29). Figure 21 shows a plot of annual utility cost as a function of storage size in Madison and Albuquerque. In both locations, increasing the storage size resulted in the annual utility cost dropping, but when the utility cost was increased beyond a certain size (approximately 15,000 kg in both cities) the utility cost began to rise slightly. The occurrence of such an optimum was unexpected in a plot of annual utility cost. The apparent reason that such an optimum exists is that as the storage exceeds a certain value, the storage does not get "uncharged" very much during each on-peak period and the heat pump has to heat up an already hot storage or cool an already cool storage. This causes the COP of the heat pump in both heating and cooling to drop slightly.

FIGURE 20. EFFECT OF USING ELECTRICAL RESISTANCE COILS TO HEAT STORAGE DURING OFF-PEAK PERIODS WITH A LOAD-SIDE HEAT PUMP/ROCK BED STORAGE SYSTEM

FIGURE 21. EFFECT OF STORAGE SIZE UNDER TIME-OF-DAY RATES
FOR A LOAD-SIDE HEAT PUMP/ROCK BED STORAGE SYSTEM

Solar-Assisted Heat Pump/Rock Bed Storage

The solar-assisted heat pump (SAHP) system that was analyzed (Figure 22) was configured by Singer Research and Development Laboratory to provide a maximum number of operational modes. A TRNSYS model of the system was received from Singer and an information flow diagram was drawn of that model (Figure 23). During checkout runs of the model several problems in the control components were noted. It became apparent that a large amount of time would be necessary to de-bug and analyze such a complex system. Further TRNSYS work was directed toward simpler heat pump/rock bed systems that could be thoroughly analyzed in the time frame of the study.

An estimate of the cost effectiveness of this SAHP system was made assuming 15,000 kg storage, 100 ft² of collector and a 1.5 ton high efficiency heat pump. The initial cost would be approximately \$6500. The maximum possible system SPF would be 3.3 based on the maximum heat pump COP. This results in an estimated 1979 utility cost in Madison of \$880 under the declining block rate and \$860 under the time-of-day rate in Albuquerque. These are the most economic rate structures in the respective cities. If these points are plotted on Figure 24, they fall above the rate of return line with tax credit in both cities. Therefore, even with a conservative estimate of system cost and an optimistic estimate of system performance, this SAHP system would not be cost effective in Madison or Albuquerque.

FIGURE 22. SOLAR-ASSISTED HEAT PUMP/ROCK BED STORAGE SYSTEM

Modes of Operation

- Heat pump heats the house by absorbing heat
 - directly from the collectors
 - from storage only
 - from the collectors and storage
 - from outdoor air only
 - from outdoor air and storage
 - from outdoor air and the collectors
 - from outdoor air, the collectors and storage.

Figure 22 (cont.)

- Heat pump cools the house by rejecting heat

- to storage only
- to outside air only
- to outside air and storage.

Control Algorithm

Heating:

Bypass damper D_1 : modulates open when $T_5 > 22.2^{\circ}\text{C}$
modulates closed when $T_5 \leq 21.1^{\circ}\text{C}$

Collector dampers D_2 and D_3 : D_2 closed and D_3 open when $T_2 > T_3$
 D_2 open and D_3 closed when $T_3 \geq T_2$

Outside air dampers D_4 , D_5 and D_6 : D_4 open and D_5/D_6 closed when $T_4 > T_{os}$
 D_4 closed and D_5/D_6 open when $T_{os} \geq T_4$

Storage dampers D_7 and D_8 : D_7 open and D_8 closed when thermostat 1st stage contacts are open.

D_7 closed and D_8 open when $T_2 > (T_1 + 1^{\circ}\text{C})$ and thermostat 1st stage contacts are open.

Outside unit blower on when $T_2 > (T_1 + 3^{\circ}\text{C})$

Outside unit blower off when $T_1 \geq T_2$

Figure 22 (cont.)

Control Algorithm

Cooling:

Bypass damper D_1 : modulates open when $T_5 < 21.1^\circ\text{C}$
modulates closed when $T_5 \geq 21.1^\circ\text{C}$

Collector dampers D_2 and D_3 : D_2 open and D_3 closed at all times.

Outside air dampers D_4 , D_5 and D_6 : D_4 closed and D_5/D_6 open when $T_4 \geq T_{os}$
 D_4 open and D_5/D_6 closed when $T_4 < T_{os}$

Storage dampers D_7 and D_8 : D_7 open and D_8 closed when thermostat 3rd stage
stage contacts open or when 3rd stage contacts
are closed and $T_3 > T_1$.

D_7 closed and D_8 open when thermostat 3rd stage
contacts are closed and $T_1 > (T_3 + 2^\circ\text{C})$.

Outside unit blower on when $T_1 > T_{os}$.

Outside unit blower off when $T_{os} \geq T_1$.

FIGURE 23. TRNSYS INFORMATION FLOW DIAGRAM FOR SOLAR-ASSISTED HEAT PUMP/ROCK BED STORAGE SYSTEM

FIGURE 24. EFFECT OF TAX CREDITS ON HEAT PUMP/ROCK BED STORAGE ECONOMICS

ECONOMICS

Comparative Results

The basic tradeoff in the economics of energy conserving systems is between capital investment cost and energy cost. The various systems studied were economically compared by plotting their estimated installed cost (capital cost) versus first year energy cost (Figures 24-26.) In addition, for a given set of economic conditions lines were plotted, using LCCA, which represent a constant rate of return for various combinations of energy and capital costs when compared to the 1.5 ton stand-alone heat pump. Systems which fall on the line or in the region below the line are cost effective when compared to the 1.5 ton stand-alone heat pump. The line represents a boundary between economic and uneconomic systems. These plots allow the evaluation of the cost effectiveness of several systems by observing whether they are in the cost effective region or observing how close they are to the cost effective region if they are not.

The sensitivity of economic parameters can be shown by plotting constant rate of return lines for various values of an economic parameter. Figure 24 shows the effect of giving a homeowner who installs a heat pump system a tax credit equal to that now being given to solar systems. (30% of the first \$2000 plus 20% of the next \$8000). The effect of such a tax credit in both Albuquerque and Madison is to move several systems from the un-cost effective region to the cost-effective region. The establishment of such a tax credit would have a beneficial effect on the economics of all the systems studied. This is not surprising since the government would, in effect, be paying for a portion of the initial cost of the system.

The sensitivity of the economic analysis to the required rate of return (discount rate) is shown in Figure 25. Two rates were plotted -- 6% and 10%. The 10% line is the case used in the other plots (Figures 24 and 26). Note that the effect of such a change in discount rate is small. None of the un-cost effective systems was shifted into the cost effective region. The effect of discount rate change appears to be smaller in Albuquerque than in Madison.

Sensitivity to fuel inflation is shown in Figure 26. The three lines represent EPRI's three scenarios concerning fuel inflation rates -- base case, high case and worst case. The other plots (Figures 24 and 25) are based on the high case scenario. The effect of changes in fuel inflation rate appear to be small relative to the effect of tax credits.

In general, it appears that the economic analysis is insensitive to any reasonable changes in fuel inflation rates or discount rate. The effect of tax credits is larger but only a few systems would be changed to cost-effective that otherwise would not be. None of the changes in the economic parameter caused a change in the relative cost effectiveness of one system to another. The most cost effective systems continued

FIGURE 25. EFFECT OF DISCOUNT RATES ON HEAT PUMP/ROCK BED STORAGE SYSTEM ECONOMICS

FIGURE 26. EFFECT OF FUEL INFLATION ON HEAT PUMP/ROCK BED STORAGE SYSTEM ECONOMICS

to be, in Albuquerque, the high-efficiency, 1.5 ton, stand-alone heat pump with the time-of-day rate and, in Madison, the high efficiency, 1.5 ton stand-alone heat pump with the declining block rate. In both cases the high efficiency, 1.5 ton, stand-alone heat pump is the most effective because of its relatively high performance and low initial cost. It should be remembered that the cost and performance data for the high efficiency heat pumps used in this study were based on theoretical models but the data is consistent with that of high performance heat pumps now being marketed by heat pump manufacturers.

Under the base case economic parameters (no tax credits, 10% rate of return, 7.5% and 9.5% fuel inflation rates) no other systems besides the stand-alone 1.5 ton high efficiency heat pumps were economic in Madison. Under the base case economic parameters in Albuquerque there were four systems which were economic. There were systems A5, A6, A27 and A28. Systems A5 and A6 are the stand-alone 1.5 ton high efficiency heat pump systems under the declining block rate and time of day rate structures. Systems A27 and A28 were the heat pump/rock bed systems in the load-side configuration under the time-of-day rate (Table 10). These were the only heat pump/rock bed systems that were cost effective but they were not as cost effective as the stand-alone 1.5 ton heat pumps under the time-of-day rate. Systems A27 and A28 are cost effective primarily because the rock storages are small and therefore the initial capital costs are low.

CONCLUSIONS

Conclusions drawn from this study of heat pump/rock bed storage systems in Albuquerque, New Mexico and Madison, Wisconsin are valid for most areas of the country. Exceptions might be areas such as the Northeast which rely on oil-fired generating plants and as a result have much higher electric utility rates. Albuquerque and Madison are served mainly by coal-fired plants. The most significant conclusions are:

- A stand-alone heat pump is the most economical system.
- The most effective heat pump is sized for maximum cooling load.
- Rock bed storage can be used to improve a heat pump's seasonal heating performance under declining block rates. The benefits derived in cooling SPF are not as great.
- Capital investment in heating and cooling equipment over the minimum amount is difficult to justify at current fuel costs and anticipated fuel inflation rates.
- Two ways to reduce life cycle cost with a heat pump/rock bed storage system are to take advantage of utility rate structures and day/night temperature swings.
- Load-side storage is the preferred configuration for a heat pump/rock bed storage system.
- The performance of an oversized heat pump can be improved by adding load-side storage to minimize the problems associated with oversizing (excessive cycling and insufficient dehumidification).
- General recommendations concerning system selection are impossible because electric utility rates vary widely throughout the country.
- The ASHRAE bin method predicts higher heating SPF's than TRNSYS with a detailed load model.

The study showed that TRNSYS and LCCA are useful analytical tools and that the new utility cost subroutine is a valuable addition to TRNSYS. The study also showed that the methodology used allows a valid life cycle cost comparison between candidate systems.

RECOMMENDATIONS

Based on the findings of this study, it is recommended that the following tasks be completed:

- Expand the capabilities of TRNSYS by:
 - adding latent cooling to the rock bed model (TYPE 10) and the heat pump model (TYPE 20).
 - rewriting the utility cost subroutine (TYPE 35) to allow more than one component of this type to be used in a simulation.
 - developing a heat pump model that accurately represents component performance.
- Use Zia's methodology to study:
 - heat pump/rock bed storage systems in areas of the country, such as the Northeast, which have high electric utility rates.
 - electrical resistance/storage systems in areas where time-of-day rates with large differences between on- and off-peak are available.
- Study methods to improve heat pump performance through the use of refrigeration controls and electronic controls.
- Resolve the differences in performance predictions that occur between the ASHRAE bin method and TRNSYS.
- Define a standard house load model and economic model (guidelines and assumptions) for use in DOE-funded studies so a comparison between study results can be made.

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APPENDIX A

BASELINE HOUSE HEAT LOSS CALCULATIONS

Design heat loss calculations were performed to determine the heating requirements of the baseline house. Table A-1 lists the heat transfer characteristics (U-values) used in the analysis; Table A-2 outlines the calculations based on a differential temperature of 80°F; and Table A-3 summarizes the results which show that the house is a 13,800 Btu/Degree Day size structure.

TABLE A-1. U-VALUES USED FOR BASELINE HOUSE HEAT LOSS CALCULATIONS

Item	Composite Material	U-Value (Btu/ft ² hr °F)
Ceiling	Pitched roof with R-30 insulation	0.04
Floor (2nd story overhang)	0.5-in. insulated sheathing (R=1.32); 6-in. fiberglass batt (R=19); and 0.75-in. plywood (R=1.18)	0.05
Walls	0.5-in. siding (R=0.79); 0.5-in. insulated sheathing (R=1.32); 3.5-in. fiberglass batt (R=11); and 0.75-in. sheetrock (R=0.15)	0.075*
Windows:		
Above ground	Double pane with wood frame (well-sealed)	0.55
Basement	Single pane with aluminum frame	1.13
Doors:		
Front	Insulated steel	0.28
Garage	1.75-in. solid core with storm door	0.28
Basement: (below grade)		
Walls (unfinished)	8-in. concrete	0.06
Floor	4-in. concrete	0.027
Infiltration:		
Single pane windows		0.62
Double pane windows		0.40
Doors		1.80
* ASHRAE Handbook of Fundamentals lists an effective U-value of 0.081 Btu/ft ² hr °F for a similar type of construction.		

TABLE A-2. BASELINE HOUSE HEAT LOSS CALCULATIONS
(Based on Design Outside Temperature of -10°F)

1st FLOOR		LIVING ROOM		LENGTH 24'0" WIDTH 12'6" HEIGHT 8'0" "				1st FLOOR		DINING-KITCHEN ROOM		LENGTH 21'6" WIDTH 10'0" HEIGHT 8'0"			
Windows and Doors -- Crackage and Area						UAT (Btu/Ft ² Hr)	Heat Loss (Btu/Hr)	Windows and Doors -- Crackage and Area						UAT (Btu/Ft ² Hr)	Heat Loss (Btu/Hr)
No.	Width of Pane	Height of Pane	No. of Lights	Lineal Ft of Crack	Area (Ft ²)			No.	Width of Pane	Height of Pane	No. of Lights	Lineal Ft of Crack	Area (Ft ²)		
1	48 1/2"	50"	1	16.0	16.0	44.0	704	1	32"	16"	2	13.3	7.1	44.0	312
2	20"	24"	2	18.0	13.3	44.0	585	Dr	2'8"	6'8"		18.7	17.8	22.4	399
4	32"	24"	2	64.0	42.7	44.0	1879	Patio	6'0"	6'9"	2	32.0	40.0	44.0	1760
Infiltration		Windows		49.0		32.2	1573	Infiltration		Windows		6.7		32.2	216
		Doors					Doors			25.4		144.0	3658		
Exposed Walls					392.0			Exposed Walls					252.0		
Net Exposed Walls					320.0	6.0	1920	Net Exposed Walls					187.1	6.0	1123
Floor (lineal feet of exposed slab)								Floor (lineal feet of exposed slab)							
Ceiling								Ceiling							
TOTAL							6666	TOTAL							7468

1st FLOOR		DEN & CLOSETS ROOM		LENGTH 14'0" WIDTH 9'0" HEIGHT 8'0"				1st FLOOR		BATH, FOYER, HALLWAY ROOM		LENGTH 14'0" WIDTH 12'6" HEIGHT 8'0"			
Windows and Doors -- Crackage and Area						UAT (Btu/Ft ² Hr)	Heat Loss (Btu/Hr)	Windows and Doors -- Crackage and Area						UAT (Btu/Ft ² Hr)	Heat Loss (Btu/Hr)
No.	Width of Pane	Height of Pane	No. of Lights	Lineal Ft of Crack	Area (Ft ²)			No.	Width of Pane	Height of Pane	No. of Lights	Lineal Ft of Crack	Area (Ft ²)		
1	32"	24"	2	16.0	10.7	44.0	471	1	28"	16"	2	12.3	6.2	44.0	273
Infiltration		Windows		8.0		32.0	258	Infiltration		Windows		6.2		32.0	200
		Doors					Doors			9.7		144.0	1397		
Exposed Walls					184.0			Exposed Walls					100.0		
Net Exposed Walls					173.3	6.0	1040	Net Exposed Walls					73.8	6.0	443
Floor (lineal feet of exposed slab)								Floor (lineal feet of exposed slab)							
Ceiling								Ceiling							
TOTAL							1769	TOTAL							2761

TABLE A-2. BASELINE HOUSE HEAT LOSS CALCULATIONS (cont.)
(Based on Design Outside Temperature of -10°F)

1st FLOOR								2nd FLOOR							
BEDROOM 1		ROOM		LENGTH 15'6" WIDTH 12'6" HEIGHT 8'0"				BEDROOM 2		ROOM		LENGTH 12'6" WIDTH 9'6" HEIGHT 8'0"			
Windows and Doors -- Crackage and Area						UAT (Btu/Ft ² Hr)	Heat Loss (Btu/Hr)	Windows and Doors -- Crackage and Area						UAT (Btu/Ft ² Hr)	Heat Loss (Btu/Hr)
No.	Width of Pane	Height of Pane	No. of Lights	Lineal Ft of Crack	Area (Ft ²)			No.	Width of Pane	Height of Pane	No. of Lights	Lineal Ft of Crack	Area (Ft ²)		
2	32"	24"	2	32.0	21.3	44.0	937	2	32"	24"	2	32.0	21.3	44.0	937
Infiltration		Windows		16.0		32.0	512	Infiltration		Windows		16.0		32.0	512
		Doors					Doors								
Exposed Walls					224.0			Exposed Walls					176.0		
Net Exposed Walls					202.7	6.0	1216	Net Exposed Walls					154.7	6.0	928
Floor (lineal feet of exposed slab)					12.5	4.0	50	Floor (lineal feet of exposed slab)							
Ceiling					193.8	3.2	620	Ceiling					118.8	3.2	380
TOTAL							3335	TOTAL							2757
2nd FLOOR								2nd FLOOR							
BEDROOM 3		ROOM		LENGTH 11'0" WIDTH 9'6" HEIGHT 8'0"				BEDROOM 4		ROOM		LENGTH 15'6" WIDTH 10'2" HEIGHT 8'0"			
Windows and Doors -- Crackage and Area						UAT (Btu/Ft ² Hr)	Heat Loss (Btu/Hr)	Windows and Doors -- Crackage and Area						UAT (Btu/Ft ² Hr)	Heat Loss (Btu/Hr)
No.	Width of Pane	Height of Pane	No. of Lights	Lineal Ft of Crack	Area (Ft ²)			No.	Width of Pane	Height of Pane	No. of Lights	Lineal Ft of Crack	Area (Ft ²)		
2	32"	24"	2	32.0	21.3	44.0	937	2	32"	24"	2	32.0	21.3	44.0	937
Infiltration		Windows		16.0		32.0	512	Infiltration		Windows		16.0		32.0	512
		Doors					Doors								
Exposed Walls					164.0			Exposed Walls					205.3		
Net Exposed Walls					142.7	6.0	856	Net Exposed Walls					184.0	6.0	1104
Floor (lineal feet of exposed slab)								Floor (lineal feet of exposed slab)					10.2	4.0	41
Ceiling					104.5	3.2	334	Ceiling					157.6	3.2	504
TOTAL							2639	TOTAL							3098

TABLE A-2. BASELINE HOUSE HEAT LOSS CALCULATIONS (cont.)
(Based on Design Outside Temperature of -10°F)

2nd FLOOR		BATHS		ROOM		LENGTH 15'6" WIDTH 11'4" HEIGHT 8'0"	
Windows and Doors -- Crackage and Area						UAT (Btu/Ft ² /hr)	Heat Loss (Btu/hr)
No.	Width of Pane	Height of Pane	No. of Lights	Lineal Ft of Crack	Area (Ft ²)		
1	32"	24"	2	16.0	10.7	44.0	471
Infiltration		Windows		8.0		32.0	256
		Doors					
Exposed Walls						90.7	
Net Exposed Walls						80.0	480
Floor (lineal feet of exposed slab)						11.3	45
Ceiling						175.7	562
TOTAL						1814	

2nd FLOOR		CLOSETS/HALLWAY		ROOM		LENGTH 10'6" WIDTH 9'6" HEIGHT 8'0"	
Windows and Doors -- Crackage and Area						UAT (Btu/Ft ² /hr)	Heat Loss (Btu/hr)
No.	Width of Pane	Height of Pane	No. of Lights	Lineal Ft of Crack	Area (Ft ²)		
Infiltration		Windows					
		Doors					
Exposed Walls						84.0	
Net Exposed Walls						84.0	504
Floor (lineal feet of exposed slab)							
Ceiling						59.8	319
TOTAL						823	

-- FLOOR		BASEMENT		ROOM		LENGTH 34'0" WIDTH 24'0" HEIGHT 8'0"	
Windows and Doors -- Crackage and Area						UAT * (Btu/Ft ² /hr)	Heat Loss (Btu/hr)
No.	Width of Pane	Height of Pane	No. of Lights	Lineal Ft of Crack	Area (Ft ²)		
5	3	2	1	50	30	90.4	2712
Infiltration		Windows		25		49.6	1240
		Doors					
Exposed Walls						928	
Net Exposed Walls						898	4310
Floor (lineal feet of exposed slab)						816.0	1632
Ceiling							
TOTAL						9894	

FLOOR		ROOM		LENGTH		WIDTH		HEIGHT	
Windows and Doors -- Crackage and Area						UAT (Btu/Ft ² /hr)	Heat Loss (Btu/hr)		
No.	Width of Pane	Height of Pane	No. of Lights	Lineal Ft of Crack	Area (Ft ²)				
Infiltration		Windows							
		Doors							
Exposed Walls									
Net Exposed Walls									
Floor (lineal feet of exposed slab)									
Ceiling									
TOTAL									

TABLE A-3. HEAT LOSS BREAKDOWN FOR BASELINE HOUSE
(Based on Differential Temperature of 80°F)

Room	Heat Loss (Btu/hr)						
	Windows	Doors	Walls	Floor	Ceiling	Infiltration	Total
1st Story:							
Living Room	3168	---	1920	---	---	1578	6666
Dining-Kitchen	312	2159	1123	---	---	3874	7468
Den/Closets	471	---	1040	---	---	258	1769
Bath/Foyer/Hallway	273	448	443	---	---	1597	2761
Subtotals	4224	2607	4526	---	---	7307	18,664
2nd Story:							
Bedroom 1	937	---	1216	50	620	512	3335
Bedroom 2	937	---	928	---	380	512	2757
Bedroom 3	937	---	856	---	334	512	2639
Bedroom 4	937	---	1104	41	504	512	3098
Baths	471	---	480	45	562	256	1814
Closets/Hallway	---	---	504	---	319	---	823
Subtotals	4219	---	5088	136	2719	2304	14,466
Basement	2712	---	4310	1632	---	1240	9894
Totals	11,155	2706	13,924	1768	2719	10,851	43,024
Percent	25.9	6.1	32.4	4.1	6.3	25.2	

$$\text{Btu/Degree Day Size} = \frac{43,024 (24)}{65 - (-10)} = 13,800$$

NOTE: Degree Day by definition is based on an inside temperature of 65°F; the heat loss calculations are based on 70°F.

APPENDIX B

SOLMOD COMPUTER PROGRAM

```

1      PROGRAM SOLMOD(INPUT,OUTPUT,TAPE,TAPE1=TAPE,TAPE4=INPUT,TAPE2,
      1  TAPE3=OUTPUT)
      DIMENSION ISTAT(24),IYR(24),IMON(24),IDAY(24),IHR(24),IMIN(24),IGS
      5  TYR(24),IDBT(24),IDPT(24),IWV(24)
      DO 60 K=1,365
      READ (1,100) (ISTAT(J),IYR(J),IMON(J),IDAY(J),IHR(J),IMIN(J),IGSTY
      1R(J),IDBT(J),IDPT(J),IWV(J),J=1,24)
      100 FORMAT (15,12,12,12,12,38X,15,45X,14,14,3X,14,14X)
      DO 60 I=1,24
      10  IF(IGSTYR(I).EQ.99999) IGSTYR(I)=00000
      GO CONTINUE
      C..PUTS DATA OUT ON PRINTER, 3 HOURS PER LINE
      DO 70 I1=1,3
      I2=I1+3
      15  I3=I1+16
      WRITE(6,300) ISTAT(I1),IYR(I1),IMON(I1),IDAY(I1),IHR(I1),IMIN(I1),
      1IGSTYR(I1),IDBT(I1),IDPT(I1),IWV(I1),ISTAT(I2),IYR(I2),IMON(I2),ID
      2AY(I2),IHR(I2),IMIN(I2),IGSTYR(I2),IDBT(I2),IDPT(I2),IWV(I2),ISTAT
      3(I3),IYR(I3),IMON(I3),IDAY(I3),IHR(I3),IMIN(I3),IGSTYR(I3),IDBT(I3
      20  4),IDPT(I3),IWV(I3)
      300 FORMAT(3(1X,15,1X,12,1X,12,1X,12,1X,12,1X,12,1X,15,1X,14,1X,14,1X,
      114,1X))
      70  CONTINUE
      C..PUTS DATA OUT ON TAPE
      DO 75 I=1,24
      25  WRITE(2,301) ISTAT(I),IYR(I),IMON(I),IDAY(I),IHR(I),IMIN(I),IGSTYR
      1(I),IDBT(I),IDPT(I),IWV(I)
      301 FORMAT (1X,15,1X,12,1X,12,1X,12,1X,12,1X,12,1X,15,1X,14,1X,14,1X,
      114,1X)
      75  CONTINUE
      WRITE(6,400)
      400 FORMAT(1H )
      60  CONTINUE
      END

```


APPENDIX C

MODIFIED TRNSYS SUBROUTINES

Type 3: Pump and Fan

Type 17: Wall

Type 18: Roof

Type 19: Room and Basement

Type 20: Heat Pump

TYPE 3: PUMP AND FAN

General Description

A pump or fan is modeled by specifying the maximum flow rate which the pump or fan is capable of delivering. The output flow rate is determined by an input control function in the range of 0 to 1. For solar energy systems, there is no continuous modulation, and the control function is either 0 or 1.

In some cases, the amount of energy consumed by the pump or fan and the amount of heat added to the flow stream is significant. This is especially true of fans used in air systems. This unit accounts for this by allowing the user to specify the amount of energy consumed and added to the flow stream with parameters 3 and 4. In the case of a fan whose motor is in the air stream, $\dot{Q}_E$ and $\dot{Q}_A$ are assumed to be the same, i.e. all the energy consumed by the motor is eventually put in the flow stream in the form of heat. In cases where the motor is outside the flow stream, $\dot{Q}_A$ will be less than $\dot{Q}_E$.

Nomenclature

- $\dot{m}_{\max}$ - maximum flow rate
- $\dot{m}_O$ - outlet flow rate
- γ - control function ($0 \leq \gamma \leq 1$)
- T_i - inlet temperature
- T_O - outlet temperature

$\dot{Q}_E$ - rate of electrical energy consumption when the pump is running

$\dot{Q}_A$ - rate of heat addition to flow when the pump is running

C_p - specific heat of fluid

$\dot{Q}_{EO}$ - rate of electric energy consumption

$\dot{Q}_{AO}$ - rate of heat addition

Mathematical Description

$$\dot{m}_o = \dot{m}_{max}$$

$$T_o = T_i + \dot{Q}_A / (\dot{m}_o C_p)$$

$$\dot{Q}_{EO} = \gamma \dot{Q}_E$$

$$\dot{Q}_{AO} = \gamma \dot{Q}_A$$

TRNSYS Component Configuration

PARAMETER NO.

1

2

3

4

INPUT NO.

1

2

3

OUTPUT NO.

1

2

3

4

DESCRIPTION

$\dot{m}_{max}$ - maximum flow rate

C_p - specific heat of fluid

$\dot{Q}_E$ - electrical energy used to drive pump

$\dot{Q}_A$ - heat added to flow by pump when running

DESCRIPTION

T_i - inlet temperature

$\dot{m}_i$ - inlet mass flow rate

γ - control function

DESCRIPTION

T_o - outlet temperature

$\dot{m}_o$ - outlet mass flow rate

$\dot{Q}_{EO}$ - rate of electrical energy consumption

$\dot{Q}_{AO}$ - rate of heat addition

Information Flow Diagram

INPUTS 3
OUTPUTS 4
PARAMETERS 4
DERIVATIVES 0

TYPE 3 (continued)

1	C	SUBROUTINE TYPE3(TIME,XIN,OUT,T,DTDT,PAR,INFO)	TR000100
		DIMENSION XIN(10),PAR(15),OUT(20),INFO(9)	TR000200
	C	THIS ROUTINE MODELS AN ON/OFF PUMP GOVERNED BY A CONTROL FUNCTION	TR000300
5	C	BETWEEN 0 AND 1. THE PUMP PRODUCES ITS MAXIMUM OUTPUT WHEN THE	TR000400
	C	INPUT CONTROL FUNCTION HAS THE VALUE 1.	TR000500
	C...	THIS ROUTINE HAS BEEN MODIFIED TO ACCOUNT FOR HEAT ADDED TO THE	10-27-78
	C...	STREAM BY THE PUMP OR FAN-BLOWER AND ALSO TO KEEP TRACK OF THE	10-27-78
10	C...	ENERGY CONSUMED. PARAMETER 3 IS THE RATE OF ENERGY CONSUMPTION AND	10-27-78
	C...	PARAMETER 4 IS THE HEAT ADDED WHEN RUNNING. OUTPUT 3 IS THE RATE	10-27-78
	C...	OF ENERGY CONSUMPTION AND OUTPUT 4 IS THE RATE OF HEAT ADDITION.	10-27-78
	C		TR000700
		IF (INFO(7).GE.0) GO TO 100	TR000800
		INFO(6)=4	10-27-78
15		INFO(9)=1	10-27-78
		CALL TYPECK(1,INFO,3,4,0)	10-27-78
	100	CONTINUE	TR001200
		FLMAX=PAR(1)	TR001300
		CP=PAR(2)	10-27-78
20		QELEC=PAR(3)	10-27-78
		QADD=PAR(4)	10-27-78
		TIN=XIN(1)	TR001400
		CF=XIN(3)	TR001500
		FLOW=CF*FLMAX	TR001600
25		IF (FLOW.GT.0) OUT(1)=TIN+(QADD/(CP*FLOW))	10-27-78
		OUT(2)=FLOW	TR001800
		OUT(3)=CF*QELEC	10-27-78
		OUT(4)=CF*QADD	10-27-78
30		RETURN	TR001900
		END	TR002000

TYPE 17: WALL

1	SUBROUTINE TYPE17(TIME,XIN,OUT,T,DTOT,PAR,INFO)	TR000100
C	TRANSFER FUNCTION WALL OR FLAT ROOF MODEL	TR000200
	DIMENSION XIN(10),OUT(20),PAR(35),INFO(9),UW(3)	TR000300
	INTEGER B,D,TSA,Q,ALPHA,AREA,TAU1,TAU2,FW,FS,EPS,AS,AE,AN,AW,	TR000400
5	FSW,FEW,FNW,FWW,FSS,FES,FNS,FWS	TR000500
	DATA ALPHA/2/,EPS/3/,AREA/6/,TAU1/7/,FW/9/,FS/10/,AS/6/,	TR000600
	AE/7/,AN/8/,AW/9/,TAU2/10/,FSW/12/,FEW/13/,FNW/14/,FWW/15/,	TR000700
	FSS/16/,FES/17/,FNS/18/,FWS/19/,TSA/3/,Q/11/	TR000800
	DATA UW/21:7,12.3,8.2/,QT/0./,QW/0./,QSHG/0./,QO/0./	TR000900
10	C IF (INFO(7) .GE. 0) GO TO 30	TR001000
	C FIRST CALL OF SIMULATION	TR001100
	INFO(6)=3	TR001200
	INFO(9) = 1	TR001300
15	N=PAR(1)	TR001400
	IF (N .NE. 1 .AND. N .NE. 2) GO TO 400	TR001500
	IF (N .EQ. 2) GO TO 12	TR001600
	C CHECK MODE 1 PARAMETERS	TR001700
	NPAR=11 + PAR(4) + PAR(5)	TR001800
20	CALL TYPECK(1,INFO,4,NPAR,0)	TR001900
	NB = PAR(4)	TR002000
	ND = PAR(5)	TR002100
	NG = PAR(8)	TR002200
	IF (NG.LT.1 .OR. NG.GT.3) GO TO 400	TR002300
25	B = 11	TR002400
	DO 5 I = 1,NB	TR002500
	OUT(TSA + I) = 20.0	TR002600
5	PAR(B + I) = PAR(B + I)+20.44	TR002700
	DO 6 I = 1,ND	TR002800
30	OUT(Q + I) = 0.0	TR002900
	GO TO 170	TR003000
	C CHECK MODE 2 PARAMETERS	TR003100
12	NPAR=19 + PAR(4) + PAR(5)	TR003200
	CALL TYPECK(1,INFO,7,NPAR,0)	TR003300
35	NB = PAR(4)	TR003400
	ND = PAR(5)	TR003500
	NG = PAR(11)	TR003600
	IF (NG.LT.1 .OR. NG.GT.3) GO TO 400	TR003700
	B = 19	TR003800
40	DO 13 I = 1,NB	TR003900
	OUT(TSA + I) = 20.0	TR004000
13	PAR(B + I) = PAR(B + I)+20.44	TR004100
	DO 14 I = 1,ND	TR004200
14	OUT(Q + I) = 0.0	TR004300
45	GO TO 170	TR004400
	C DO CALCULATIONS ONCE PER HOUR	TR004500
	30 IF((ABS(TIME-IFIX(TIME+.001)).GT.0.001).OR.(INFO(7).GT.0)) RETURN	TR004600
	N=PAR(1)	12-22-78
	IF (N .EQ. 2) GO TO 100	TR004800
50	C MODE 1 -- TRANSFER FUNCTION WALL OR FLAT ROOF	TR004900
	TA = XIN(1)	TR005000
	HT = XIN(2)	TR005100
	W = XIN(3)	TR005200
	TR = XIN(4)	TR005300
55	NB=PAR(4)	TR005400
	ND=PAR(5)	TR005500
	NG = PAR(8)	TR005600
		TR005700

TYPE 17 (continued)

	NT = PAR(11)	TR005800
	B = 11	TR005900
60	D = B + NB	TR006000
	HO=53.15 + 13.72*W	TR006100
	OUT(TSA + 1) = TA + PAR(ALPHA)*HT/HO	TR006200
	IF (NT .LT. 0) OUT(TSA + 1) = OUT(TSA + 1) + PAR(EPS)*2.041092E-07	TR006300
	(9.2844E-06(TA + 273.16)**6 - (TA + 273.16)**4)/HO	TR006400
65	S1 = 0.0	TR006500
	S2 = 0.0	TR006600
	CN = 0.0	TR006700
	DO 45 I=1,NB	TR006800
	S1 = S1 + PAR(B + I)*OUT(TSA + I)	TR006900
70	45 CN = CN + PAR(B + I)	TR007000
	DO 50 I=1,ND	TR007100
50	S2 = S2 + PAR(D + I)*OUT(Q + I)	TR007200
	Q0=S1 - S2 - CN*TR	TR007300
	QWALL=Q0*PAR(AREA)*(1.0 - PAR(FW))	TR007400
75	QSHG=HT*PAR(TAU1)*PAR(AREA)*PAR(FW)*(1.0 - PAR(FS))	TR007500
	QWINDC=UW(NG)*PAR(AREA)*PAR(FW)*(TA - TR)	TR007600
	QW=QWALL + QWINDC	TR007700
	QT=QW + QSHG	TR007800
	GO TO 170	TR007900
80	C MODE 2 -- TRANSFER FUNCTION HOUSE (4 WALLS)	TR008000
100	TA = XIN(1)	TR008100
	HTS = XIN(2)	TR008200
	HTE = XIN(3)	TR008300
	HTN = XIN(4)	TR008400
85	HTW = XIN(5)	TR008500
	W = XIN(6)	TR008600
	TR = XIN(7)	TR008700
	NB=PAR(4)	TR008800
	ND=PAR(5)	TR008900
90	NG = PAR(11)	TR009000
	B = 19	TR009100
	D = B + NB	TR009200
	ATOTAL=PAR(AS)*(1.0 - PAR(FSW)) + PAR(AE)*(1.0 - PAR(FEW)) +	TR009300
	PAR(AN)*(1.0 - PAR(FNW)) + PAR(AW)*(1.0 - PAR(FWW))	TR009400
95	FS2 = PAR(AS)*(1.0 - PAR(FSW))/ATOTAL	TR009500
	FE2 = PAR(AE)*(1.0 - PAR(FEW))/ATOTAL	TR009600
	FN2 = PAR(AN)*(1.0 - PAR(FNW))/ATOTAL	TR009700
	FW2 = PAR(AW)*(1.0 - PAR(FWW))/ATOTAL	TR009800
	HO=53.15 + 13.72*W	TR009900
100	TSAS = TA + PAR(ALPHA)*HTS/HO	TR010000
	TSAE = TA + PAR(ALPHA)*HTE/HO	TR010100
	TSAN = TA + PAR(ALPHA)*HTN/HO	TR010200
	TSAW = TA + PAR(ALPHA)*HTW/HO	TR010300
	OUT(TSA + 1) = FS2*TSAS + FE2*TSAE + FN2*TSAN + FW2*TSAW	TR010400
105	S1 = 0.0	TR010500
	S2 = 0.0	TR010600
	CN = 0.0	TR010700
	DO 145 I=1,NB	TR010800
	S1 = S1 + PAR(B + I)*OUT(TSA + I)	TR010900
110	145 CN = CN + PAR(B + I)	TR011000
	DO 150 I=1,ND	TR011100
150	S2 = S2 + PAR(D + I)*OUT(Q + I)	TR011200
	Q0=S1 - S2 - CN*TR	TR011300
	QWALL=Q0*ATOTAL	TR011400

TYPE 17 (continued)

115	QSOLS=HTS*PAR(TAU2)*PAR(AS)*PAR(FSW)*(1.0 - PAR(FSS))	TR011500
	QSOLE=HTE*PAR(TAU2)*PAR(AE)*PAR(FEW)*(1.0 - PAR(FES))	TR011600
	QSOLN=HTN*PAR(TAU2)*PAR(AN)*PAR(FNW)*(1.0 - PAR(FNS))	TR011700
	QSOLW=HTW*PAR(TAU2)*PAR(AW)*PAR(FWW)*(1.0 - PAR(FWS))	TR011800
120	QSHG=QSOLS + QSOLE + QSOLN + QSOLW	TR011900
	QWINDC = UW(NG)*(PAR(AS)*PAR(FSW) + PAR(AE)*PAR(FEW) + PAR(AN)*	TR012000
	PAR(FNW) + PAR(AW)*PAR(FWW)) * (TA - TR)	TR012100
	QW=QWALL + QWINDC	TR012200
	QT=QW + QSHG	TR012300
125	C SET OUTPUTS AND SHIFT Q'S AND TSA'S IN OUT ARRAY	TR012400
170	OUT(1) = QT	TR012500
	OUT(2) = QW	TR012600
	OUT(3) = QSHG	TR012700
	DO 180 II=2,NB	12-22-78
	I=NB+2-II	12-22-78
130	180 OUT(TSA + I) = OUT(TSA + I - 1)	TR012900
	DO 190 II=2,ND	12-22-78
	I=ND+2-II	12-22-78
	190 OUT(Q + I) = OUT(Q + I - 1)	TR013100
	OUT(Q + I) = QO	TR013200
135	RETURN	TR013300
	C PARAMETER ERROR	TR013400
400	CALL TYPECK(4,INFO,0,0,0)	TR013500
	RETURN	TR013600
	END	TR013700

TYPE 18: ROOF

1	C.	SUBROUTINE TYPE18(TIME,XIN,OUT,T,DTDT,PAR,INFO)	TR000100
	C	THIS ROUTINE USES THE TRANSFER FUNCTION METHOD TO MODEL A	TR000200
	C	PITCHED ROOF AND ATTIC. A COLLECTOR ON THE SOUTH ROOF IS	TR000300
5	C	OPTIONAL. NROOF=1 INDICATES COLLECTOR ON SOUTH ROOF WHILE NROOF	TR000400
	C	=-1 PROVIDES FOR NO COLLECTOR. PARAMETER NC DETERMINES THE AMOUNT	TR000500
	C	OF INSULATION IN THE CEILING. 1=3[, 2=6[, 3=9[, 4=12[.	TR000600
		DIMENSION XIN(8),OUT(15),PAR(30),INFO(9)	TR000700
		DIMENSION B(4,6),D(4,5),TSA(7),Q(7)	TR000800
10		DATA Q/7*0.0/,QCEIL/0.0/,QO/0.0/	TR000900
		DATA TSA/7*20.0/	TR001000
		DATA PI/3.141592/	TR001100
		DATA B/0.00084,0.00052,0.00039,0.00030, 0.00751,0.00451,0.00336,TR001200	TR001300
		10.00278, 0.00511,0.00329,0.00279,0.00238, -0.00430,-0.00272,-0.001400	TR001400
15		2.00157,-0.00125, -0.00369,-0.00234,-0.00141,-0.00118, -0.00204TR001500	TR001600
		3,-0.00130,-0.00085,-0.00070/	TR001700
		DATA D/-1.25439,-1.28241,-1.23693,-1.23383, 0.29560,0.30779,0.31TR001800	TR001900
		1851,0.30193, -0.19590,-0.18806,-0.18634,-0.16181, 0.28489,0.29TR001800	TR001900
20		2307,0.26428,0.24726, -0.07900,-0.08484,-0.07738,-0.07007/	TR002000
		IF(INFO(7).GE.0) GO TO 3	TR002100
		INFO(6)=3	TR002200
		INFO(9) = 1	TR002300
		CALL TYPECK(1,INFO,8,13,0)	TR002400
25	3	CONTINUE	12-22-78
		IF((ABS(TIME-IFIX(TIME+.001)).GT.0.001).OR.(INFO(7).GT.0)) RETURN	TR002600
		NROOF=PAR(1)	TR002700
		NC=PAR(2)	TR002800
		ALPHA=PAR(3)	TR002900
30		EPS=PAR(4)	TR003000
		AREAS=PAR(5)	TR003100
		AREAE=PAR(6)	TR003200
		AREAN=PAR(7)	TR003300
		AREAW=PAR(8)	TR003400
		AREAC=PAR(9)	TR003500
35		VOLRT=PAR(10)	TR003600
		UBE=PAR(11)	TR003700
		ANGLES=PAR(12)*PI/180.0	TR003800
		ANGLEN=PAR(13)*PI/180.0	TR003900
40		FVNSKY=(1.0 + COS(ANGLEN))/2.0	TR004000
		FVSSKY=(1.0 + COS(ANGLES))/2.0	TR004100
		FVNGRD=(1.0 - COS(ANGLEN))/2.0	TR004200
		FVSGRD=(1.0 - COS(ANGLES))/2.0	TR004300
		TA=XIN(1)	TR004400
45		HS=XIN(2)	TR004500
		HE=XIN(3)	TR004600
		HN=XIN(4)	TR004700
		HW=XIN(5)	TR004800
		W=XIN(6)	TR004900
		TC=XIN(7)	TR005000
50		TR=XIN(8)	TR005100
		RHO=636.23/(TA*1.8 + 492.0)	TR005200
		CONDI=VOLRT*RHO*1.012	TR005300
		HO= 53.15 + 13.72*W	TR005400
55		IF (NROOF) 21,21,22	TR005500
	21	C1=HO*AREAS*32.70/(HO + 32.70)	TR005600
		GO TO 23	TR005700
22		C1= 32.70*UBE*AREAS/(UBE+32.70)	

TYPE 18 (continued)

	23	C2=HO*AREAE*29.84/(HO + 29.84)	TR005800
		C3=HO*AREAN*32.70/(HO + 32.70)	TR005900
60		C4=HO*AREAW*29.84/(HO + 29.84)	TR006000
		CEFF=C1 + C2 + C3 + C4 + CONDI	TR006100
		TAMBK=TA + 273.18	TR006200
		TSKY=0.0552*(TAMBK**1.5)	TR006300
		TSURR=TAMBK + 10.0	TR006400
65		SIGMA=2.041092E-07	TR006500
		DELTAN=EPS*SIGMA*(FVNSKY*(TSKY**4 - TAMBK**4) + FVNGRD*(TSURR**4 -	TR006600
		1TAMBK**4))	TR006700
		DELTAS=EPS*SIGMA*(FVSSKY*(TSKY**4 - TAMBK**4) + FVSGRD*(TSURR**4 -	TR006800
		1TAMBK**4))	TR006900
70		IF (NROOF) 26,26,27	TR007000
	26	TSAS=TA + HS*ALPHA/HO + DELTAS/HO	TR007100
		GO TO 29	TR007200
	27	TSAS=TC	TR007300
	29	TSAE=TA + HE*ALPHA/HO	TR007400
75		TSAN=TA + HN*ALPHA/HO + DELTAN/HO	TR007500
		TSAW=TA + HW*ALPHA/HO	TR007600
		CN=0.0	TR007700
		DO 25 I=1,8	TR007800
	25	CN= CN + 20.44*B(NC,I)	TR007900
80		IF (INFO(7) .LT. 0) GO TO 70	TR008000
	41	DO 42 I=2,8	TR008100
	42	TSA(I)=OUT(I)	TR008200
		DO 43 I=1,5	TR008300
	43	Q(I)=OUT(I + 7)	TR008400
85	44	TSA(I)=(C1*TSAS + C2*TSAE + C3*TSAN + C4*TSAW + CONDI*TA)/CEFF	TR008500
		S1=0.0	TR008600
		S2=0.0	TR008700
		DO 45 I=1,8	TR008800
	45	S1=S1 + 20.44*B(NC,I)*TSA(I)	TR008900
90		DO 50 I=1,5	TR009000
	50	S2=S2 + D(NC,I)*Q(I)	TR009100
		Q0=S1 - S2 - CN*TR	TR009200
		QCEIL=Q0*AREAC	TR009300
	70	OUT(I)=QCEIL	TR009400
95		DO 80 I=1,8	TR009500
	80	OUT(I + 1)=TSA(I)	TR009600
		OUT(8)=Q0	TR009700
		DO 90 I=1,5	TR009800
	90	OUT(I + 8)=Q(I)	TR009900
100	91	RETURN	TR010000
		END	TR010100

TYPE 19: ROOM AND BASEMENT

TYPE 19 Modifications

The TRNSYS TYPE 19 component was modified to include a latent load calculation in energy rate control (MODE 1) and a dehumidification calculation in temperature level control (MODE 2). These modifications have no effect on the model in heating season calculations; they affect only air conditioning. The rate of moisture gain is calculated as follows:

$$\dot{M}_{H_2O} = \dot{M}_{int} + (W_{amb} - W_{room}) \dot{V}_{inf} \rho_{air}$$

where:

$\dot{M}_{H_2O}$ is the space moisture gain, Kg/hr

$\dot{M}_{int}$ is the rate of internal moisture generation, Kg/hr

W_{amb} is the ambient specific humidity, $\frac{\text{Kg } H_2O}{\text{Kg dry air}}$

W_{room} is room or space specific humidity, $\frac{\text{Kg } H_2O}{\text{Kg dry air}}$

$\dot{V}_{inf}$ is the volume infiltration rate, m^3/hr

ρ_{air} is the density of dry air in the room, $\frac{\text{Kg dry air}}{m^3}$

In energy rate control, the W_{room} is assumed constant at 0.0102 and the latent load is:

$$\dot{Q}_{lat} = \max \left\{ \begin{array}{l} \dot{M}_{H_2O} \Delta h_{fv} \\ 0 \end{array} \right.$$

where:

Δh_{fv} is the heat of vaporization of water

In temperature level control, the room humidity and temperature are calculated timestep by timestep.

$$CAP_m \frac{dW_{room}}{d\theta} = \dot{M}_{int\ gen} + (W_{amb} - W_{room})\dot{V}_{inf\ air} - \frac{\dot{Q}_{ac, lat}}{\Delta h_{fv}}$$

$$CAP_t \frac{dtr}{d\theta} = \Sigma(\text{sens. loads}) - \dot{Q}_{ac, sens}$$

where:

θ = time

CAP_m = room moisture capacitance (expressed as equivalent Kg dry air)

$\Sigma(\text{sens loads})$ = total sensible cooling loads (+vc) or heating loads (-vc)

$\dot{Q}_{ac, lat}$ = latent heat removed at air conditioning coil.

The heat removal rate at the cooling coil, $\dot{Q}_{ac}$, is input. The coil surface temperature, t_s , is specified, $\dot{Q}_{ac, lat}$ and $\dot{Q}_{ac, sens}$ are calculated based on a "straight line" coil condition curve, as follows:

If $W_{room} > W_s$

$$\dot{Q}_{ac, lat} = \dot{Q}_{ac} \frac{(W_{room} - W_s)\Delta h_{fv}}{(W_{room} - W_s)\Delta h_{fv} + (t_{room} - t_s)Cp}$$

$$\dot{Q}_{ac, sens} = \dot{Q}_{ac} \frac{Cp(t_{room} - t_s)}{(W_{room} - W_s)\Delta h_{fv} + (t_{room} - t_s)Cp}$$

If $W_{room} \leq W_s$

$$\dot{Q}_{ac,lat} = 0$$

$$\dot{Q}_{ac,sens} = \dot{Q}_{ac}$$

Parameters, Inputs, Outputs

Note: Unit dependent, energy flows in KJ, time in hr, length in m, temperature in °C, etc. (standard TRNSYS units)

MODE 1 Parameters 1-17: as in current TRNSYS

18: $\dot{M}_{int\ gen}$, Kg/hr

Inputs 1-5: as in current TRNSYS

6: ambient relative humidity, %

7: $\dot{Q}_{cond}$ - roof, KJ/hr

Outputs 1-3: as in current TRNSYS

4: Q_{latent} (latent load), KJ/hr

5: Total AC load, KJ/hr

MODE 2 Parameters 1-17: as in current TRNSYS

18: $\dot{M}_{int\ gen}$, Kg/hr

19: CAP_m , Kg dry air

20: t_s , °C

Inputs 1,2: as in current TRNSYS

3: $\dot{Q}_{ac}$, KJ/hr

4: t_{amb} , °C

5: $\dot{Q}_{cond}$, KJ/hr

6: $\dot{Q}_{shg}$, KJ/hr

7: $\dot{Q}_{lights}$, KJ/hr

} See TRNSYS
Manual

- 8: infiltration rate
- 9: ambient relative humidity, %
- 10: $\dot{Q}_{\text{cond}}$ - roof, KJ/hr

Outputs 1-6: as in current TRNSYS

- 7: $W_{\text{room}}, \frac{\text{Kg H}_2\text{O}}{\text{Kg dry air}}$
- 8: $\dot{Q}_{\text{ac, sens}}, \text{KJ/hr}$
- 9: $\dot{Q}_{\text{ac, lat}}, \text{KJ/hr}$

- Derivatives
- 1 $t_{\text{room}}, ^\circ\text{C}$
 - 2 $W_{\text{room}}, \frac{\text{Kg H}_2\text{O}}{\text{Kg dry air}}$

TYPE 19 (continued)

1	C.		TR000100
		SUBROUTINE TYPE19(TIME,XIN,OUT,T,DTDT,PAR,INFO)	TR000200
	C...	THIS IS JEFF BLINNS (UNIVERSITY OF WISCONSIN) TYPE 19 WHICH ACCOUNTS	
	C...	FOR LATENT COOLING LOAD. THE CHANGES WERE SENT TO ZIA IN DEC. 1978.	
5	C...	INPUT 10 HAS BEEN SET UP AS THE INPUT FOR QO FROM TYPE 18. THERE	
	C...	IS NO CAPABILITY FOR ADDITIONAL NON-DISTRIBUTED HEAT INPUTS.	
	C...	INPUT QPEPL HAS BEEN CHANGED TO INFILTRATION RATE.	
	C...		
10	C	THIS ROUTINE MODELS THE INTERIOR OF A SPACE TO BE HEATED	TR000300
	C	OR COOLED. MODE 1 IS COMPATIBLE WITH ENERGY RATE CONTROL	TR000400
	C	AND MODE 2 IS COMPATIBLE WITH TEMPERATURE LEVEL CONTROL.	TR000500
	C	ASHRAE'S TRANSFER FUNCTION METHOD OF DISTRIBUTING	TR000600
	C	TIME DEPENDANT HEAT GAINS IS USED.	TR000700
		REAL KT	TR000800
15		DIMENSION XIN(10),DTDT(2),T(2),OUT(20),PAR(20),INFO(9),QST(4)	
		DIMENSION DIST(4,3,4),W(3,4),Q(4,4),QT(4,4),FC(4)	TR001000
		COMMON /SIM/TIME0,TFINAL,DELT	TR001100
		P1(Z)=-7.90290*(Z-1.0)	
		P2(Z)=5.02808*ALOG10(Z)	
20		P3(Z)=-1.3816E-07*(10.**((11.344*(1.-1./Z))-1.))	
		P4(Z)=8.1326E-03*(10.**((-3.49149*(Z-1.))-1.))	
		P5(Z)=-9.09718*(Z-1.)	
		P6(Z)=-3.56654*ALOG10(Z)	
		P7(Z)=0.876793*(1.-1./Z)	
25		DATA IUNIT/0/	TR001200
		DATA Q/16=0.0/	TR001300
		DATA QT/16=0.0/	TR001400
		DATA W/1.000,1.000,1.000,-1.8260,-2.1092,-2.2908,1.0697,1.46	TR001500
		106,1.7252,-0.2005,-0.3331,-0.4277/	TR001600
30		DATA DIST/0.2727,0.6982,0.3178,0.3251,0.2217,0.7108,0.2605,0.2574,	TR001700
		10.2155,0.7055,0.2430,0.2503,-0.3400,-1.2017,-0.4507,-0.4267,-0.	TR001800
		23354,-1.4456,-0.4662,-0.4038,-0.3712,-1.5668,-0.5085,-0.4446,	TR001900
		30.1169,0.6817,0.2089,0.1524,0.1448,0.9639,0.2819,0.1830,0.1790,1.1	TR002000
		4378,0.3547,0.2255,-0.0064,-0.1150,-0.0328,-0.0076,-0.0128,-0.2108,	TR002100
35		5-0.0579,-0.0183,-0.0166,-0.2698,-0.0825,-0.0245/	TR002200
		IF (INFO(7).GE.0) GO TO 403	TR002300
		INFO(9) = 1	TR002400
		P8=-2.2193	
		IF (IUNIT.NE.0) CALL TYPECK(6,INFO,0,0,0)	TR002500
40		IUNIT=INFO(1)	TR002600
		N=INFO(3)	TR002700
		MODE=PAR(1)	TR002800
		IF (MODE.LT.1.OR.MODE.GT.2) CALL TYPECK(4,INFO,0,0,0)	TR002900
		VOL=PAR(2)	TR003000
45		RATE = PAR(3)	TR003100
		AREA=PAR(4)	TR003200
		IC = PAR(5)	TR003300
		CAPAC = PAR(6)	TR003400
		SUMUA = PAR(7)	TR003500
50		IBASE = PAR(8)	TR003600
		DEPTH = PAR(9)	TR003700
		PERIM = PAR(10)	TR003800
		TGRD=PAR(11)	TR003900
		GO TO (501,502),MODE	TR004000
55	501	TMIN=PAR(14)	TR004100
		TMAX=PAR(15)	TR004200
		TO=PAR(16)	TR004300

TYPE 19 (continued)

		GO TO 503	TR004400
	502	CMIN=PAR(14)	TR004500
60		EFF=PAR(15)	TR004600
		CPH=PAR(16)	TR004700
		CAPHUM=PAR(19)	
	C	CAPHUM=ROOM MOISTURE CAPACITANCE IN EQUIVALENT PG OF DRY AIR =	
	C	DERIVATIVE OF TOTAL ROOM MOISTURE WRT ROOM ABSOLUTE HUMIDITY	
65	503	IFC=PAR(17)	TR004800
		WDOOT=PAR(18)	
		GO TO (401,402),MODE	
	401	CONTINUE	TR004900
			TR005000
	C.	TYPECK CALLS FOR MODE=1	TR005100
70		TRN=PAR(18)	TR005200
		TROOM=TRN	
		OUT(2)=TRN	TR005300
		CALL TYPECK(1,INFO,N,18,0)	
		INFO(8)=5	
75		GO TO 403	TR005600
	402	CONTINUE	TR005700
	C.	TYPECK CALLS FOR MODE=2	TR005800
		CALL TYPECK(1,INFO,N,20,2)	
		INFO(8)=9	
80	403	CONTINUE	TR006100
		DO 44 J=1,4	TR006200
	44	FC(J)=1.0	TR006300
		GO TO (1,2),MODE	TR006400
	1	CONTINUE	TR006500
85	C.	READ INPUTS	TR006600
		M=0	TR006700
		K=1	TR006800
		IF (N .EQ. 7) K=0	
		GO TO 5	TR007000
90	2	M=3	TR007200
		K=1	
		IF (N .EQ. 10) K=0	
		THI=XIN(1)	TR007400
		FLWH=XIN(2)	TR007500
95		QAC=XIN(3)	
		TROOM=T(1)	
	5	TAMB=XIN(M+1)	TR007700
		QST(2)=XIN(M+2) +XIN(M+7)	
		QST(1)=XIN(M+3)	TR007900
100		QST(3)=XIN(M+4) + PAR(12)	TR008000
		QST(4)= PAR(13)*237.4	
		RH=XIN(M+6)/100.	
		QSTSUM=0.	TR008200
		DO 45 J=1,4	TR008300
105	45	QSTSUM=QSTSUM+QST(J)	TR008400
		QL=0.0	TR008500
		REM=ABS(TIME-IFIX(TIME+.001))	12-22-78
		IF (REM.GT.0.001.OR.INFO(7).GT.0) GO TO 405	TR009200
		KT=SUMUA/(2.0*PERIM)*0.1605	
		IF (QSTSUM.LE.0.0.OR.IFC.EQ.0) GO TO 114	TR009400
110	C.	DEFINE FC COEFFICIENTS FOR AIR CONDITIONING LOADS AS IN ASHRAE	TR009500
		FC(1)=1.0-0.019*KT	TR009600
		FC(2)=1.0-0.016*KT	TR009700
		FC(3)=1.0-0.022*KT	TR009800

TYPE 19 (continued)

115	FC(4)=1.0-0.025*KT	TR009900
114	CONTINUE	TR010000
	DO 14 J=1,4	TR010100
	DO 12 I=1,3	TR010200
	K=5-I	TR010300
120	Q(J,K)=Q(J,K-1)	TR010400
	QT(J,K)=QT(J,K-1)	TR010500
	12 CONTINUE	TR010600
	Q(J,1)=QST(J)	TR010700
	14 CONTINUE	TR010800
125	15 DO 30 J=1,4	TR010900
	S1=0.0	TR011000
	S2=0.0	TR011100
	DO 20 I=1,4	TR011200
	20 S1=S1 + DIST(J,IC,I)*FC(J)*Q(J,I)	TR011300
130	DO 25 I=2,4	TR011400
	25 S2=S2 + W(IC,I)*Q(J,I)	TR011500
	QT(J,1)=S1 - S2	TR011600
	30 CONTINUE	TR011700
	QLD=QT(1,1) + QT(2,1) + QT(3,1) + QT(4,1)	TR011800
135	405 CONTINUE	TR011900
	C RELATIVE-TO-ABSOLUTE HUMIDITY PROCESSOR	
	TABS=TAMB+273.16	
	IF(TABS)310,310,320	
140	310 Z=273.16/TABS	
	PSAT=10.** (P5(Z)+P6(Z)+P7(Z)+P8)	
	GO TO 330	
	320 Z=373.16/TABS	
	PSAT=10.** (P1(Z)+P2(Z)+P3(Z)+P4(Z))	
145	330 WAMB=0.622*RH*PSAT/(1.-RH*PSAT)	
	C LATENT LOAD CALCULATION	
	QLAT=(RATE*VOL*1.204*(WAMB-0.012)+WDOOT)*2503.	
	QINFL=(RATE*XIN(M+5))*VOL*1.2185*(TAMB-TROOM)	
	QBASE=0.	TR012100
	IF(IBASE) 35,50,40	TR012200
150	35 QBASE=PERIM*2.1633*(TAMB - TROOM)	TR012300
	GO TO 50	TR012400
	40 CONTINUE	TR012500
	QBSWL=PERIM*DEPTH*1.136*((TGRD + TAMB)/2.0 - TROOM)	TR012600
	QBSFR=AREA*1.136*(TGRD - TROOM)	TR012700
155	QBASE=QBSWL + QBSFR	TR012800
	50 CONTINUE	TR012900
	QLOAD=QL + QLD + QINFL + QBASE	TR013000
	IF(MODE.EQ.2) GO TO 205	TR013100
160	55 CONTINUE	TR013200
	IF(MODE .EQ. 2) GO TO 205	TR013300
	C. SAVE TROOM FROM PREVIOUS TIMESTEP IN OUT(15)	TR013400
	IF (INFO(7).GT.0) GO TO 56	
	OUT(15)=OUT(2)	TR013600
	56 CONTINUE	TR013700
165	TROOM=OUT(15)	TR013800
	TRN=TROOM+QLOAD*DELT/CAPAC	TR013900
	IF(TRN.LT.TMIN) GO TO 100	TR014000
	IF(TRN.LE.TMAX) GO TO 130	TR014100
	TRN=TMAX	TR014200
170	QLOAD=QLOAD+(TROOM-TMAX)*CAPAC/DELT	TR014300
	GO TO 150	TR014400

TYPE 19 (continued)

	100	TRN=TMIN	TR014500
		QLOAD=QLOAD+(TROOM-TMIN)*CAPAC/DELT	TR014600
		GO TO 150	TR014700
175	130	QLOAD=0.	TR014800
	150	OUT(1)=QLOAD	TR014900
		OUT(2)=TRN	TR015000
		OUT(3)=QBASE	TR015100
		IF(QLAT.LT.0)QLAT=0.0	
180		OUT(4)=QLAT	
		OUT(5)=QLOAD+QLAT	
		IF(QLOAD.LE.0.0)OUT(5)=QLOAD	
	C	OUT(5)=TOTAL AC LOAD IF POSITIVE AND HEATING LOAD IF NEGATIVE	TR015200
		RETURN	TR015300
185	205	CONTINUE	TR015400
		CFLOW=FLWH*CPH	TR015500
		IF(CFLOW.LT..01) GO TO 210	
		QTRAN=CMIN*EFF*(THI-T(1))	TR015700
		TOUT=THI-QTRAN/CFLOW	TR015800
190		GO TO 220	TR015900
	210	TOUT=THI	TR016000
		QTRAN=0.0	
	220	IF(INFO(7).GT.-1) GO TO 230	
		TCOIL=PAR(20)	
195		IF(TCOIL)222,222,223	
	222	ZC=273.16/(TCOIL+273.16)	
		PSC=10*(P5(ZC)+P6(ZC)+P7(ZC)+P8)	
		GO TO 224	
	223	ZC=373.16/(TCOIL+273.16)	
200		PSC=10*(P1(ZC)+P2(ZC)+P3(ZC)+P4(ZC))	
	224	WCOIL=0.622*PSC/(1.-PSC)	
		OUT(10)=WCOIL	
	230	WCOIL=OUT(10)	
		IF(T(2).LE.WCOIL)GO TO 233	
205		QLINE=1.012*(TROOM-TCOIL)+2503.*(T(2)-WCOIL)	
		GO TO 234	
	C	NO DEHUMIDIFICATION	
	233	QLINE=1.012*(TROOM-TCOIL)	
210	234	SRATIO=1.012*(TROOM-TCOIL)/QLINE	
		QACS=SRATIO*QAC	
		QACL=(1.-SRATIO)*QAC	
		QEXS=QLOAD+QTRAN-QACS	
		DTDT(1)=QEXS/CAPAC	
		DTDT(2)=(RATE*VOL*1.204*(WAMB-T(2))+WDOT-QACL/2503.)/CAPHUM	
215	300	OUT(1)=TOUT	TR016300
		OUT(2)=FLWH	TR016400
		OUT(3)=QEXS	TR016500
		OUT(4)=TROOM	TR016600
		OUT(5)=QTRAN	TR016700
220		OUT(6)=QBASE	TR016800
		OUT(7)=T(2)	
	C	T(2) IS ROOM ABSOLUTE HUMIDITY	
		OUT(8)=QACS	
		OUT(9)=QACL	
225		RETURN	TR016900
		END	TR017000

TYPE 20: HEAT PUMP

1	C.	SUBROUTINE TYPE20(TIME,XIN,OUT,T,DTDT,PAR,INFO)	TR000100
	C	THIS ROUTINE MODELS A HEAT PUMP CAPABLE OF USING AIR OR	TR000200
	C	LIQUID SINKS AND SOURCES AND OPERATING IN A VARIETY OF MODES.	TR000300
5	C	THE INTERNAL CONTROLLER WILL STICK AFTER 4 ITERATIONS AT A GIVEN	TR000400
	C	TIMESTEP IF IT HAS NOT SETTLED ON A MODE BY THAT TIME. THIS CAN BE	TR000500
	C	CHANGED BY MODIFYING THE VALUE OF NSTK IN THE DATA STATEMENT.	TR000600
	C	THE FOLLOWING MODE CODE CONVENTION IS USED:	TR000700
	C	MODE=1 - HEAT WITH SOLAR SOURCE EVAPORATOR	TR000800
10	C	MODE=2 - HEAT WITH OUTDOOR SOURCE EVAPORATOR	TR000900
	C	MODE=3 - COOL WITH SOLAR SINK CONDENSER	TR001000
	C	MODE=4 - COOL WITH OUTDOOR SINK CONDENSER	TR001100
	C	MODE=5 - DIRECT HEATING FROM SOLAR SOURCE	TR001200
	C	MODE=6 - AUXILIARY	TR001300
15	C	MODE=7 - EVERYTHING OFF	TR001400
	C	THE FOLLOWING MODE COMBINATIONS MAY OCCUR SIMULTANEOUSLY:	TR001500
	C	2-5,2-5-6,1-6,2-6,5-6	TR001600
		REAL MDOOT1,MDOOT0	TR001700
		DIMENSION XIN(10),OUT(20),PAR(20),INFO(9),Y(3)	TR001800
20		DATA NSTK/8/,IUNIT/0/	TR002000
		WAH=0.	TR002100
		QA1=0.	TR002200
		QR1=0.	TR002300
25		H1=0.	TR002400
		H2=0.	TR002500
		H3=0.	TR002600
		QDH=0.	TR002700
		QAC=0.	TR002800
		QRC=0.	TR002900
30		WAC=0.	TR003000
		QR2=0.	TR003100
		QA2=0.	TR003200
		QAUXH=0.	TR003300
		QSHRTC=0.	TR003400
35		IF(INFO(7).GE.0) GO TO 10	TR003500
		INFO(6)=19	
		INFO(9)=1	9-29-78
		OUT(19)=0.	TR003800
		IF(PAR(15).LT.0..OR.PAR(15).GT.1.) CALL TYPECK(4,INFO,0,0,0)	TR003900
40		CALL TYPECK(1,INFO,7,19,0)	
	10	IF (INFO(1) .EQ. IUNIT) GO TO 12	TR004100
		IUNIT = INFO(1)	TR004200
		CP1=PAR(1)	TR004300
		MDOOT1=PAR(2)	TR004400
45		TROOM1=PAR(3)	TR004500
		TROOMC=PAR(4)	TR004600
		TMIN1=PAR(5)	TR004700
		TMIN2=PAR(6)	TR004800
		NDA1=PAR(7)	TR004900
50		NDA2=PAR(8)	TR005000
		NDA3=PAR(9)	TR005100
		ISAVE1=PAR(10)	TR005200
		ISAVE2=PAR(11)	TR005300
		ISAVEC=PAR(12)	TR005400
55		TCOOL=PAR(13)	TR005500
		CAIR=PAR(14)	TR005600
		EFF=PAR(15)	TR005700

TYPE 20 (continued)

	TSET=PAR(16)	TR005800
	ICool=PAR(17)	TR005900
60	THEAT=PAR(18)	TR006000
	FLOWC=PAR(19)	
12	TTANK=XIN(1)	TR006100
	QLOAD=XIN(4)	TR006200
	TAMB=XIN(3)	TR006300
65	IGAM=XIN(5)	TR006400
	TINC=XIN(6)	
	MODE=IFIX(OUT(19)+.01)	TR006500
	MDOTO=MDOT1	TR006600
	C CHECK NUMBER OF ITERATIONS	TR006700
70	IF(INFO(7).LE.NSTK) GO TO 100	TR006800
	C USE PREVIOUS MODE	TR006900
	GO TO (310,400,520,520,110,600,700) ,MODE	TR007000
	C FIND NEW MODE	TR007100
100	IF(IGAM.EQ.0) GO TO 700	TR007200
75	IF (QLOAD.GE.0.) GO TO 500	TR007300
	IF(TAMB.GT.THEAT) GO TO 700	TR007400
	IF (TTANK.LE.TSET) GO TO 300	TR007500
	C DIRECT HEATING FROM STORAGE TANK	TR007600
110	MODE =5	TR007700
80	CHOT=MDOT1*CP1	TR007800
	IF (CAIR.LE.CHOT) GO TO 120	TR007900
	QMAX=CHOT*(TTANK-TROOMH)	TR008000
	GO TO 140	TR008100
120	QMAX=CAIR*(TTANK-TROOMH)	TR008200
85	140 QTRAN=EFF*QMAX	TR008300
	HRSON4=ABS(QLOAD/QTRAN)	TR008400
	IF(HRSON4.LE.1) GO TO 155	TR008500
	HRSON4=1.	TR008600
	QAUXH=-(QLOAD+QTRAN)	TR008700
90	QDH=QTRAN*HRSON4	TR008800
	TOUT=TTANK-QDH/CHOT	TR008900
	C CHECK AIR SOURCE HEAT PUMP AS AN AUXILIARY TO DIRECT HEAT.	TR009000
	C DERATE HEAT PUMP PERFORMANCE BY DECREASING TAMB.	TR009100
95	TSRC=TAMB-QDH/CAIR	TR009200
	IF(QAUXH.LE.0.0.OR.TSRC.LT.TMIN2) GO TO 800	TR009300
	CALL DATA(1SAVE2,NDATA2,3,TSRC,Y,INFO)	TR009400
	QA=Y(1)	TR009500
	QR=Y(2)	TR009600
	WA=Y(3)	TR009700
100	IF(QR.LE.0.) GO TO 800	TR009800
	H2=QAUXH/QR	TR009900
	IF(H2-1.) 159,159,159	TR010000
105	158 H2=1.	TR010100
	QAUXH=QAUXH-QR	TR010200
	GO TO 160	TR010300
	159 QAUXH=0.	TR010400
110	160 WAH=WA+H2	TR010500
	QR2=QR+H2	TR010600
	QA2=QA+H2	TR010700
	GO TO 800	TR010800
	C SYSTEM IS IN MODE 1,2,OR 6	TR010900
300	IF(TTANK.LE.TMIN1.AND.TAMB.LE.TMIN2) GO TO 600	TR011000
	IF(TTANK.LE.TAMB.AND.TAMB.GE.TMIN2) GO TO 400	TR011100
	IF(TTANK.LE.TMIN1.AND.TTANK.GT.TAMB.AND.TAMB.GE.TMIN2)GO TO 400	TR011200

TYPE 20 (continued)

115	C HEAT FROM SOLAR SOURCE EVAPORATOR	TR011300
310	MODE=1	TR011400
	CALL DATA(1SAVE1,N0DATA1,3,TTANK,Y,INFO)	TR011500
	QA=Y(1)	TR011600
	QR=Y(2)	TR011700
120	WA=Y(3)	TR011800
	IF(QR.LE.0.0) GO TO 600	TR011900
	H1=-QLOAD/QR	TR012000
	IF (H1.LE.1.) GO TO 360	TR012100
	H1=1.	TR012200
125	QAUXH=-(QLOAD+QR)	TR012300
360	WAH=WA+H1	TR012400
	QA1=QA+H1	TR012500
	QR1=QR+H1	TR012600
	TOUT=TTANK-(QA1/(MDOT1*CP1))	TR012700
130	GO TO 800	TR012800
460	CONTINUE	TR012900
	C HEAT FROM OUTDOOR SOURCE EVAPORATOR	TR013000
	MODE=2	TR013100
	CALL DATA(1SAVE2,N0DATA2,3,TAMB,Y,INFO)	TR013200
135	QA=Y(1)	TR013300
	QR=Y(2)	TR013400
	WA=Y(3)	TR013500
	IF(QR.LE.0.0) GO TO 600	TR013600
	H2=-QLOAD/QR	TR013700
140	IF (H2.LE.1.) GO TO 460	TR013800
	H2=1.	TR013900
	QAUXH=-(QLOAD+QR)	TR014000
460	WAH=WA+H2	TR014100
	QR2=QR+H2	TR014200
145	QA2=QA+H2	TR014300
	TOUT=TTANK	TR014400
	MDOT0=0.	TR014500
	GO TO 800	TR014600
	C COOLING MODE	TR014700
150	900 IF(TAMB.LE.TCOOL)GO TO 700	TR014800
	510 MODE=3+ICCOOL	TR014900
	920 CALL DATA(1SAVEC,N0DATA3,3,TAMB,Y,INFO)	TR015000
	QA=Y(1)	TR015100
	QR=Y(2)	TR015200
155	WA=Y(3)	TR015300
	IF(QA.LE.0.0) GO TO 600	TR015400
	H3=QLOAD/QA	TR015500
	IF (H3.LE.1.) GO TO 558	TR015600
	H3=1.	TR015700
160	QSHRTC=QLOAD-QA	TR015800
	GO TO 560	TR015900
	558 QSHRTC=0.0	TR016000
	560 WAC=WA+H3	TR016100
	QAC=QA+H3	TR016200
165	QRC=QR+H3	TR016300
	IF(1CCOOL.GE.1) GO TO 595	TR016400
	TOUT=TTANK+QRC/(MDOT1*CP1)	TR016500
	GO TO 800	TR016600
	595 TOUT=TTANK	TR016700
170	MDOT0=0.	TR016800
	GO TO 800	TR016900

TYPE 20 (continued)

	C AUXILIARY ONLY OPERATION	TR017000
	600 MODE=6	TR017100
	IF(QLOAD.GE.0.) GO TO 620	TR017200
175	QAUXH=-QLOAD	TR017300
	GO TO 640	TR017400
	620 QSHRTC=QLOAD	TR017500
	640 MDOTO=0.	TR017600
	TOUT=TTANK	TR017700
180	GO TO 800	TR017800
	C HEAT PUMP NOT OPERATING	TR017900
	700 MODE=7	TR018000
	705 TOUT=TTANK	TR018100
	MDOTO=0.	TR018200
185	800 OUT(1)=TOUT	TR018300
	OUT(2)=MDOTO	TR018400
	OUT(3)=WAM	TR018500
	OUT(4)=QA1	TR018600
	OUT(5)=QR1	TR018700
190	OUT(6)=H1	TR018800
	OUT(7)=H2	TR018900
	OUT(8)=H3	TR019000
	OUT(9)=QDH	TR019100
	OUT(10)=QAUXH	TR019200
195	OUT(11)=QSHRTC	TR019300
	OUT(12)=QAC	TR019400
	OUT(13)=QRC	TR019500
	OUT(14)=WAC	TR019600
	OUT(15)=QR2	TR019700
200	OUT(16)=QA2	TR019800
	OUT(17)=TINC+(QR1+QR2-QAC)/CAIR	
	OUT(18)=FLOWC*IGAM	
	OUT(19)=MODE	TR019900
	RETURN	TR020000
205	END	TR020100

APPENDIX D

NEW TRNSYS SUBROUTINES

TYPE 34: Relative Humidity

TYPE 35: Utility Cost

TYPE 34: RELATIVE HUMIDITY

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1      C      SUBROUTINE TYPE34(TIME,XIN,OUT,T,DTDT,PAR,INFO)
C      THIS SUBROUTINE CALCULATES THE RELATIVE HUMIDITY OF
C      AIR AT ATMOSPHERIC PRESSURE GIVEN THE DRY BULB (TDB)
5      C      AND DEW POINT (TDP) TEMPERATURES OF THE AIR.
C
      DIMENSION XIN(2),OUT(3),INFO(9)
      INFO(6)=3
      INFO(9)=0
10     IF(INFO(7).GE.0) GO TO 10
      CALL TYPECK(1,INFO,2,0,0)
10     TDB=XIN(1)
      TDBF=1.8*TDB+32.
      TDP=XIN(2)
16     TOPF=1.8*TDP+32.
      IF(TOP.GE.TDB) GO TO 20
      XDB=374.11-TDB
      DBK=273.16+TDB
20     PSATDB=218.167/(EXP(2.302585*XDB*(3.2437814+5.86826E-3
1=XDB+1.1702379E-8*XDB**3)/(DBK*(1.+2.1878462E-3*XDB))))
      XDP=374.11-TDP
      DPK=273.16+TDP
      PSATDP=218.167/(EXP(2.302585*XDP*(3.2437814+5.86826E-3
1=XDP+1.1702379E-8*XDP**3)/(DPK*(1.+2.1878462E-3*XDP))))
25     RH=(PSATDP/PSATDB)*100.
      GO TO 30
20     RH=100.
30     OUT(1)=TDBF
      OUT(2)=TOPF
      OUT(3)=RH
30     RETURN
      END

```


TYPE 35: UTILITY COST

General Description

Most solar energy systems require an external source of energy to power the pumps, blowers, and control devices and to provide a backup energy source when the solar system cannot carry the load. Often this external energy is provided by public utilities in the form of electricity, natural gas, or steam. This subroutine sums and integrates the energy consumed by other subroutines, and at specific intervals, it calculates the cost of energy using utility rate structures. To be considered a proper representation of utility cost, one of the inputs should be the domestic energy use (non-system consumption).

The first parameter, MODE, selects one of three rate structures to be used. The specific rates that apply are entered as other parameters. Rates and rate structures can be obtained from the utility company that services the area under consideration. Recent electric rates for the United States are published in the National Electric Rate Book.

Only one TYPE 35 component is allowed in any simulation.

Nomenclature

CONV - conversion factor (by multiplication) to convert from Q input units to utility billing units. For example, the simulation is done in kJ and the billing unit is kwhr, $CONV = 2.778 \text{ E-04}$.

C_{cust} - the customer charge covering metering, billing, and other fixed costs.

- C_{Dn} - cost per demand unit in n^{th} demand block.
- C_{min} - minimum monthly bill.
- C_n - cost per energy unit, the n^{th} block or time period.
- D_n - demand units in the n^{th} demand block.
- D - demand as of last call to unit (MODE 3) could be used as an input to a demand controller.
- E_n - energy units in n^{th} block.
- FCA - fuel cost adjustment, cost per energy unit charged by the utility.
- γ_n - control function that turns on (value = 1) during n^{th} time-of-day period.
- Q_n - power consumed by the system or the rest of the building.

MODE 1

MODE 1 assumes the block utility rate structure and has the capability of winter/summer rate differentials. Most utilities currently use a form of the block rate to bill both gas and electricity. The most common application of the block rate is the declining block rate in which the utility charges less per energy unit for the consumption of higher amounts (blocks) of energy. Some utilities are starting to implement flat and inverted rates. Flat rates charge the same for all energy units consumed (essentially one block), and inverted rates charge more per energy unit the more that are consumed. The so-called "life line" rates for electricity where the first block of a few hundred kwhr are cheaper are inverted rates.

This subroutine provides for five energy blocks, each of which can have a summer and winter cost per unit of energy. The last block is usually "all additional," and the user should set the last block (E_n) to a very large number so that all additional energy units are charged at the last block rate. If the utility has no winter/summer rate differential, the user should make these costs (C_{wn} , C_{sn}) the same. The example on page E-10 shows the use of MODE 1 with a declining block electric rate.

<u>Parameter No.</u>	<u>Description</u>
1	MODE: Specify 1
2	CONV - Conversion factor from simulation energy units to utility billing units
3	PER - utility billing period (hours)
4	FCA - fuel cost adjustment (\$/energy unit)
5	$\dot{C}_{cust}$ - customer charge
6	$\dot{C}_{min}$ - minimum monthly bill
7	$T_{s_{start}}$ - hour of simulation that utility "summer" season starts
8	$T_{s_{end}}$ - hour of simulation that "summer" season ends
9	E_1 - utility energy units in first block
10	$\dot{C}_{w1}$ - winter cost per energy unit, first block
11	$\dot{C}_{s1}$ - summer cost per energy unit, first block
12	E_2 - utility energy units in second block
13	$\dot{C}_{w2}$ - winter cost per energy unit, second block
14	$\dot{C}_{s2}$ - summer cost per energy unit, second block
15	E_3 - utility energy units in third block
16	$\dot{C}_{w3}$ - winter cost per energy unit, third block

17	$\dot{C}_{s3}$ - summer cost per energy unit, third block
18	E_4 - utility energy units in fourth block
19	$\dot{C}_{w4}$ - winter cost per energy unit, fourth unit
20	$\dot{C}_{s4}$ - summer cost per energy unit, fourth unit
21	E_5 - utility energy units in fifth block
22	$\dot{C}_{w5}$ - winter cost per energy unit, fifth block
23	$\dot{C}_{s5}$ - summer cost per energy unit, fifth block

<u>Input No.</u>	<u>Description</u>
1	$\dot{Q}_1$ - rate of energy consumption
2	.
3	.
4	.
5	.
6	.
7	.
8	.
9	.
10	$\dot{Q}_{10}$ - rate of energy consumption

<u>Output No.</u>	<u>Description</u>
1	Q_{per} - utility energy units consumed since start of current billing period
2	Q_{tot} - total utility energy units used through last billing period
3	C_{tot} - total utility cost through most recent billing period
4	C_{cust} - utility cost due to customer charge
5	C_{fca} - utility cost due to fuel cost adjustment
6	C_1 - utility cost due to first block

- 7 C_2 - utility cost due to second block
- 8 C_3 - utility cost due to third block
- 9 C_4 - utility cost due to fourth block
- 10 C_5 - utility cost due to fifth block

Note: Outputs 2 through 10 represent accumulated costs through most recent billing period.

MODE 2

MODE 2 assumes a time-of-day rate structure in which the customer is charged different rates per energy unit depending on the time of day the energy is consumed. Off-peak or on-peak rates are a form of time-of-day rates.

This mode has the capability of handling up to five rate seasons and three daily time periods. Parameters are not required for energy rate season, but parameters must be entered for each of the three daily time periods. If a rate structure has only two time periods, the starting time of the first period should equal the ending time of the other period, and the parameters for the third period will have no effect.

MODE 2 acts as its own printer. At the end of a utility billing period, a table of the energy and the cost of the energy consumed in each of the three time periods is printed.

<u>Parameter No.</u>	<u>Description</u>
1	MODE - Specify 2
2	CONV - Conversion factor from simulation energy units to utility billing units
3	PER - utility billing period (hours)
4	FCA - fuel cost adjustment (\$/energy unit)
5	$\dot{C}_{\text{cust}}$ - customer charge
6	$\dot{C}_{\text{min}}$ - minimum monthly bill
7	DAY - day of week that simulation starts; 1=Sunday, 7=Saturday
8	SS_1 - hour of simulation that rate season starts

9	SE_1 - hour of simulation that rate season ends
10	T_{S11} - start of first daily time period is seasonal period - on-peak period (hr of day)
11	T_{E11} - end of first daily time period (hr of day)
12	$\dot{C}_{11}$ - cost per energy unit in first time period
13	WE_{11} - 1 if rate applies on weekend, 0 if rate does not apply on weekend.
14	T_{S12} - start of second daily time period (hr of day)
15	T_{E12} - end of second daily time period (hr of day)
16	$\dot{C}_{12}$ - cost per energy unit in second period
17	WE_{12} - 1 if rate applies on weekend
18	$\dot{C}_{13}$ - cost per energy unit in third time period
19	WE_{13} - 1 if rate applies on weekend

Repeat parameters 8-19 for each additional season.

<u>Input No.</u>	<u>Description</u>
1	$\dot{Q}_1$ - rate of energy consumption
:	:
10	$\dot{Q}_{10}$ - rate of energy consumption

<u>Output No.</u>	<u>Description</u>
1	γ_1 - control function = 1 during first daily time period (on-peak period)
2	γ_2 - control function = 1 during second daily time period (off-peak period)
3	DAY - day of week

MODE 3. MODE 3 assumes the Hopkinson demand energy rate structure in which there is a charge based on demand and a charge for energy consumed. Utilities quite often charge for demand and energy in blocks.

This mode has three blocks of demand and energy. As in the block rate structure, the last block size (D_n , E_n) should be set to a very large number to cover the "all additional" case.

<u>Parameter No.</u>	<u>Description</u>
1	MODE - Specify 3
2	CONV -- conversion factor from simulation energy units to utility billing units
3	PER - utility billing period (hours)
4	FCA - fuel cost adjustment (\$/energy unit)
5	$\dot{C}_{\text{cust}}$ - customer charge
6	$\dot{C}_{\text{min}}$ - minimum monthly bill
7	D_1 - demand units in first block
8	$\dot{C}_{D1}$ - cost per demand unit in first block
9	D_2 - demand units in second block
10	$\dot{C}_{D2}$ - cost per demand unit in second block
11	D_3 - demand units in third block
12	$\dot{C}_{D3}$ - cost per demand unit in third block
13	E_1 - energy units in first block
14	$\dot{C}_{E1}$ - cost per energy unit in first block

- 15 E_2 - energy units in second block
- 16 $\dot{C}_{E2}$ - cost per energy unit in second block
- 17 E_3 - energy units in third block
- 18 $\dot{C}_{E3}$ - cost per energy unit in third block

<u>Input No.</u>	<u>Description</u>
1	$\dot{Q}_1$ - rate of energy consumption
2	.
3	.
4	.
5	.
6	.
7	.
8	.
9	.
10	$\dot{Q}_{10}$ - rate of energy consumption

<u>Output No.</u>	<u>Description</u>
1	Q_{per} - utility energy units consumed since start of current billing period
2	D_{per} - peak demand required since the start of current billing period
3	T_{Dper} - simulation time at which peak demand last occurred in current billing period
4	Q_{tot} - total utility energy units

5	C_{tot} - total utility cost through most recent billing period
6	C_{cust} - utility cost due to customer charge
7	C_{fca} - utility cost due to fuel cost adjustment
8	C_{D1} - utility cost due to first demand block
9	C_{D2} - utility cost due to second demand block
10	C_{D3} - utility cost due to third demand block
11	C_{E1} - utility cost due to first energy block
12	C_{E2} - utility cost due to second energy block
13	C_{E3} - utility cost due to third energy block
14	D - demand as of last call

EXAMPLE A one-year simulation is being done of a solar space heating and cooling system located in Madison, Wisconsin. The simulation units are expressed in kJ. Madison Gas and Electric Company has the following rate for residential service:

Customer charge: \$2.00/bill

Energy charge:

<u>Winter</u>	<u>Summer</u>		
3.60¢	4.20¢	per kwhr	first 100 kwhr
3.50¢	4.20¢	per kwhr	next 400 kwhr
3.20¢	4.20¢	per kwhr	next 500 kwhr
2.75¢	4.20¢	per kwhr	all additional kwhr

Summer period: 120 days following June 16

Minimum bill; \$2.00/bill

Power cost adjustment: 0.31¢ per kwhr

Assume: 12 billing periods/year; 730 hrs per period

The required parameters are:

1	2.778E-04	730	.0031	2.00	
2.00	4008	6888	100	.036	.042
400	.035	.042	500	.032	.042
99999	.0275	.042	0	0	0

The inputs are:

Q consumed by all units powered by electricity

A type 21 printer should be set up to print out the desired outputs.

The cost information should be output at intervals identical to the billing period; in this case, 730 hours.

Information Flow Diagrams

INPUTS up to 10
 OUTPUTS 10
 PARAMETERS 23
 DERIVATIVES 0

Parameters

1) MODE: 1	9) E_1	17) $\dot{C}_{S3}$
2) CONV	10) $\dot{C}_{W1}$	18) E_4
3) PER	11) $\dot{C}_{S1}$	19) $\dot{C}_{W4}$
4) FCA	12) E_2	20) $\dot{C}_{S4}$
5) $\dot{C}_{cust}$	13) $\dot{C}_{W2}$	21) E_5
6) $\dot{C}_{min}$	14) $\dot{C}_{S2}$	22) $\dot{C}_{W5}$
7) T_s start	15) E_3	23) $\dot{C}_{S5}$
8) T_s end	16) $\dot{C}_{W3}$	

INPUTS up to 10
 OUTPUTS 12
 PARAMETERS 13
 DERIVATIVES 0

Parameters

- 1) MODE: 2
- 2) CONV
- 3) PER
- 4) FCA
- 5) $\dot{C}_{cust}$
- 6) $\dot{C}_{min}$
- 7) DAY_{1S}

- 8) SE_1
- 9) SE_1
- 10) T_{S11}
- 11) T_{E11}
- 12) $\dot{C}_{11}$
- 13) WE_{11}
- 14) T_{S12}

- 15) T_{E12}
- 16) $\dot{C}_{12}$
- 17) WE_{12}
- 18) $\dot{C}_{13}$
- 19) WE_{13}

Note: Repeat parameters
 8-19 for each
 additional season

INPUTS up to 10
 OUTPUTS 14
 PARAMETERS 18
 DERIVATIVES 0

Parameters

1) MODE: 3	10) $\dot{C}_{D2}$
2) CONV	11) D_3
3) PER	12) $\dot{C}_{D3}$
4) FCA	13) E_1
5) $\dot{C}_{cust}$	14) $\dot{C}_{E1}$
6) $\dot{C}_{min}$	15) E_2
7) D_1	16) $\dot{C}_{E2}$
8) $\dot{C}_{D1}$	17) E_3
9) D_2	18) $\dot{C}_{E3}$

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```

1      SUBROUTINE TYPE35(TIME,XIN,OUT,T,DTOT,PAR,INFO)
      DIMENSION B(5),          CBLK(6),COST(5),CD(3),E(3),CE(3)
      DIMENSION PAR(50),XIN(10),OUT(20),INFO(9),C(10,5)
      DIMENSION QBUF(3,10),QPRD(3),DMD(3),QBLK(3),SS(10,3),SE(10,3)
5      DIMENSION TS(10,3),TE(10,3),WE(10,3)
      COMMON /SIM/ TIME0,TFINAL,DELT
      COMMON /LUNITS/ LUR,LW,WIDTH
      DATA BLANK/1H /,DASH/1H-/
10     C...THIS COMPONENT CALCULATES THE COST OF THE SUPPLEMENTAL ENERGY
      C...REQUIRED BY A SYSTEM USING TYPICAL UTILITY RATE STRUCTURES. THE
      C...FIRST PARAMETER, MODE, DETERMINES WHICH RATE STRUCTURE WILL BE USED.
      C... IF MODE=1 BLOCK RATE STRUCTURE IS ASSUMED
      C... IF MODE=2 TIME OF DAY RATE STRUCTURE IS ASSUMED
15     C... IF MODE=3 DEMAND/ENERGY RATE STRUCTURE IS ASSUMED
      C...THIS TRNSYS SUBROUTINE WAS WRITTEN BY ZIA ASSOCIATES, BOULDER, CO.
      C...THIS WORK HAS BEEN SUPPORTED BY THE SOLAR HEATING AND COOLING
      C...RESEARCH AND DEVELOPMENT BRANCH, OFFICE OF CONSERVATION AND SOLAR
      C...APPLICATIONS, U.S. DEPARTMENT OF ENERGY.
      C...
20     IF(INFO(7).GE.0) GO TO 80
      C...FIRST CALL, INITIALIZE VARIABLES USED BY ALL MODES
      INFO(9)=1
      N=INFO(3)
      NP=INFO(4)
25     MODE=PAR(1)
      CONV=PAR(2)
      PER=PAR(3)
      CFCA=PAR(4)
      CCUST=PAR(5)
30     CMIN=PAR(6)
      QPER=0
      QTOT=0
      CFCATL=0
      CCUSTL=0
35     DO 5 I=1,5
5      CBLK(I)=0
      DO 7 I=1,3
7      QPRD(I)=0
      IF(MODE.NE.1) GO TO 10
      CALL TYPECK(1,INFO,N,23,0)
40     C...INITIALIZE MODE 1
      INFO(6)=10
      TSSTRT=PAR(7)
      TSEND=PAR(8)
45     B(1)=PAR(9)
      C(1,1)=PAR(10)
      C(2,1)=PAR(11)
      B(2)=PAR(12)
      C(1,2)=PAR(13)
50     C(2,2)=PAR(14)
      B(3)=PAR(15)
      C(1,3)=PAR(16)
      C(2,3)=PAR(17)
      B(4)=PAR(18)
55     C(1,4)=PAR(19)
      C(2,4)=PAR(20)
      C(2,4)=PAR(20)

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        B(5)=PAR(21)
        C(1,5)=PAR(22)
60      C(2,5)=PAR(23)
        GOTO 80
      10 IF(MODE.NE.2) GO TO 20
        CALL TYPECK(1,INFO,N,NP,0)
C...INITIALIZE MODE 2
65      NS=(NP-6)/12
        DAY=PAR(7)
        DO 15 J=1,NS
          I=(J-1)*12
          SS(J)=PAR(I+8)
          SE(J)=PAR(I+9)
70      TS(J,1)=PAR(I+10)
          TE(J,1)=PAR(I+11)
          C(J,1)=PAR(I+12)
          WE(J,1)=PAR(I+13)
75      TS(J,2)=PAR(I+14)
          TE(J,2)=PAR(I+15)
          C(J,2)=PAR(I+16)
          WE(J,2)=PAR(I+17)
          C(J,3)=PAR(I+18)
80      WE(J,3)=PAR(I+19)
      15 CONTINUE
        GOTO 80
      20 IF (MODE.NE.3) CALLTYPECK(4,INFO,0,0,0)
        CALL TYPECK(1,INFO,N,18,0)
85      C...INITIALIZE MODE 3
          INFO(6)=14
          DMD(1)=PAR(7)
          CD(1)=PAR(8)
          DMD(2)=PAR(9)
90      CD(2)=PAR(10)
          DMD(3)=PAR(11)
          CD(3)=PAR(12)
          E(1)=PAR(13)
          CE(1)=PAR(14)
95      E(2)=PAR(15)
          CE(2)=PAR(16)
          E(3)=PAR(17)
          CE(3)=PAR(18)
      80 CONTINUE
100     IF(MODE.EQ.2) GOTO 200
          IF(MODE.EQ.3) GOTO 300
C...BLOCK RATE STRUCTURE.....
          IF(INFO(7).EQ.0) GOTO 100
C... PUT INPUT VALUES INTO BUFFER
105     85 DO 90 I=1,N
          90 QBUF(I)=XIN(I)
          RETURN
C...FIRST CALL IN THE TIME STEP, INTERGRATE INPUTS
110     100 DO 110 I=1,N
          Q=QBUF(I)*DELT*CONV
          QPER=QPER+Q
          QTOT=QTOT+Q
          110 CONTINUE
          OUT(1)=QPER

```



```

115         TIMER=TIME-TIMEO+.001
           IF(AMOD(TIMER,PER).GT.DELT.OR.TIMER.LT.DELT.OR.INFO(7).EQ.-1)
             + GO TO 85
C...CALCULATES UTILITY BILL FOR BILLING PERIOD ASSUMING BLOCK RATE
           K=1
120         IF(TIME.GE.TSSTRT.AND.TIME.LT.TSEND) K=2
           CPER=CCUST+CFCAL*QPER
           CFCATL=CFCATL+CFCAL*QPER
           CCUSTL=CCUSTL+CCUST
           DO 150 I=1,5
125         IF(QPER.LE.0) GO TO 155
           IF(QPER.GT.B(I)) GO TO 130
           COST(I)=QPER*C(K,I)
           GO TO 135
130         COST(I)=B(I)*C(K,I)
135         CPER=CPER+COST(I)
           CBLK(I)=CBLK(I)+COST(I)
           QPER=QPER-B(I)
150         CONTINUE
155         IF(CPER.GE.CMIN) GO TO 160
135         CPER=CMIN
160         CTOT=CTOT+CPER
           OUT(2)=QTOT
           OUT(3)=CTOT
           OUT(4)=CCUSTL
140         OUT(5)=CFCATL
           OUT(6)=CBLK(1)
           OUT(7)=CBLK(2)
           OUT(8)=CBLK(3)
           OUT(9)=CBLK(4)
145         OUT(10)=CBLK(5)
           QPER=0.0
           CPER=0.0
           GO TO 85
C...TIME OF DAY RATE STRUCTURE.....
150         200 CONTINUE
           IF(INFO(7).EQ.0) GO TO 250
C...DETERMINE SEASON
           205 DO 206 J=1,NS
           206 IF(TIME.GT.SS(J).AND.TIME.LT.SE(J)) IS=J
155         C...PUT INPUT VALUES INTO BUFFER
           OUT(1)=0.0
           OUT(2)=0.0
           TIMEDY=AMOD(TIME,24.0)
           TIMEWK=AMOD((TIME+DAY*24.0),168.0)
160         IDAYWK=INT(TIMEWK/24.0) +1
           OUT(3)=IDAYWK
C...L IS THE TIME PERIOD OF THE DAY
           DO 230 J=1,3
           IF(TS(IS,J).GT.TE(IS,J)) GO TO 207
165         IF(TIMEDY.GT.TS(IS,J).AND.TIMEDY.LT.TE(IS,J)) GO TO 240
           GO TO 230
           207 IF(TIMEDY.LT.TS(IS,J).OR .TIMEDY.GE.TE(IS,J)) GO TO 240
           230 CONTINUE
           240 L=J
170         IF(WE(IS,L).EQ.0.AND.IDAYWK.EQ.1) L=L+1
           IF(WE(IS,L).EQ.0.AND.IDAYWK.EQ.7) L=L+1

```

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      IF(WE(IS,L).EQ.0.AND.IDAYWK.EQ.1) L=L+1
      IF(WE(IS,L).EQ.0.AND.IDAYWK.EQ.7) L=L+1
      IF(L.GT.3) CALL TYPECK(4,INFO,0,0,0)
175      DO 220 I=1,N
          QBUF(L,I)=XIN(I)
          IF(L.EQ.1) OUT(1)=1.0
          IF(L.EQ.2) OUT(2)=1.0
          RETURN
180      C...FIRST CALL IN TIMESTEP, INTERGRATE INPUTS
          250 DO 260 I=1,N
              QPRD(L)=QPRD(L)+(QBUF(L,I)*DELT*CONV)
              QTOT=QTOT+(QBUF(L,I)*DELT*CONV)
          260 CONTINUE
          TIMER=TIME-TIME0+.001
          IF(AMOD(TIMER,PER).GT.DELT.OR.TIMER.LT.DELT.OR.INFO(7).EQ.-1)
              + GO TO 205
          C...CALCULATES UTILITY BILL FOR BILLING PERIOD ASSUMING TIME OF DAY RATE
          CPER=CCUST+CFCA*(QPRD(1)+QPRD(2)+QPRD(3))
190      CCUSTL=CCUSTL+CCUST
          DO 270 I=1,3
              COST(I)=QPRD(I)*C(IS,I)
              CPER=CPER+COST(I)
              CBLK(I)=CBLK(I)+COST(I)
195      QBLK(I)=QBLK(I)+QPRD(I)
              QPER=QPER+QPRD(I)
          270 CONTINUE
              IF(CPER.GE.CMIN) GOTO 280
              CPER=CMIN
200      CMINTL=CMINTL+CMIN
          280 CTOT=CTOT+CPER
          C...PRINTS UTILITY BILL
              WRITE(LUW,291)(DASH,I=1,132)
          291 FORMAT(132A1//)
              WRITE(LUW,292) TIME
205      292 FORMAT(25X,12HUTILITY COST/18X,26HTIME OF DAY RATE STRUCTURE//5X,
                  131HPERIOD ENDING SIMULATION TIME :,F10.2)
              WRITE(LUW,293) QPRD(1),COST(1),QPRD(2),COST(2),QPRD(3),COST(3),
                  1 QPER,CPER
210      293 FORMAT(/ 22X,21HENERGY UNITS CONSUMED,10X,4HCOS,//11X,14HFIRST PER
                  11OD :,F10.2,15X,F10.2, /10X,15HSECOND PERIOD :,F10.2,15X,F10.2,/11
                  2X,14HTHIRD PERIOD :,F10.2,15X,F10.2//18X,7HTOTAL :,F10.2,15X,F10.2
                  3)
              WRITE(LUW,295)
215      295 FORMAT(/5X,16HSIMULATION TOTAL)
              WRITE(LUW,293) QBLK(1),CBLK(1),QBLK(2),CBLK(2),QBLK(3),CBLK(3),
                  1 QTOT,CTOT
              WRITE(LUW,291)(DASH,I=1,132)
              QPER=0
220      CPER=0
              DO 285 I=1,3
          285 QPRD(I)=0
              GO TO 205
          C...DEMAND/ENERGY RATE STRUCTURE, MODE=3.....
225      300 CONTINUE
              IF(INFO(7).EQ.0) GOTO 330
          C...PUT INPUT VALUES INTO BUFFER
          305 D=0

```

```

230      DO 310 I=1,N
          QBUF(I)=XIN(I)*CONV
          D=D+QBUF(I)
310      CONTINUE
          OUT(14)=D
          RETURN
235      C...FIRST CALL IN TIME STEP, INTERGRATE INPUTS
          330 DO 340 I=1,N
              Q=QBUF(I)*DELT
              QTOT=QTOT+Q
              QPER=QPER+Q
240          340 CONTINUE
              IF(D.LE.QPER) GOTO 345
              DPER=D
              TPER=TIME
          345 OUT(1)=QPER
245          OUT(2)=QPER
              OUT(3)=TPER
              TIMER=TIME-TIME0+.001
              IF(AMOD(TIMER,PER).GT.DELT.OR.TIMER.LT.DELT.OR.(INFO(7).EQ.-1)
                + GO TO 305
250      C...CALCULATES UTILITY BILL FOR BILLING PERIOD
          CPER=CCUST+CFCATL*QPER
          CFCATL=CFCATL+CFCATL*QPER
          CCUSTL=CCUSTL+CCUST
          DO 350 I=1,3
255          IF(QPER.LE.0) GO TO 365
              IF(QPER.GT.E(I)) GOTO 360
              COST(I)=QPER*CE(I)
              GO TO 365
          360 COST(I)=E(I)*CE(I)
260          CBLK(I)=CBLK(I)+COST(I)
          365 ID=I+3
              IF(DPER.LE.0) GO TO 355
              IF(DPER.GT.DMD(I)) GO TO 370
              COST(ID)=CD(I)*DPER
265          GO TO 375
          370 COST(ID)=DMD(I)*CD(I)
          375 CPER=CPER+COST(I)+COST(ID)
              CBLK(ID)=CBLK(I)+COST(ID)
          355 QPER=QPER-B(I)
270          DPER=DPER-DMD(I)
          350 CONTINUE
              IF(CPER.GE.CMIN) GOTO 380
              CPER=CMIN
              CMINTL=CMINTL+CMIN
275          380 CTOT=CTOT+CPER
              QPER=0
              OUT(4)=QTOT
              OUT(5)=CTOT
              OUT(6)=CCUSTL
280          OUT(7)=CFCATL
              OUT(8)=CBLK(1)
              OUT(9)=CBLK(2)
              OUT(10)=CBLK(3)
              OUT(11)=CBLK(4)
285          OUT(12)=CBLK(5)
              OUT(13)=CBLK(6)
              GO TO 305
          END

```

APPENDIX E

LIFE CYCLE COST ANALYSIS COMPUTER PROGRAM

LIFE CYCLE COST ANALYSIS (LCCA) COMPUTER PROGRAM

Life Cycle Cost Analysis (LCCA) is a computer program that provides a complete economic analysis of any two heating and/or cooling systems and compares the relative cost effectiveness of the two. The systems are referred to in the program as the baseline and alternative systems with the alternative system being the one the user believes to be the more economical. A listing of LCCA is presented on pages E-3 through E-10, followed by a sample printout on pages E-9 and E-10.

LCCA prints out all the inputs and their descriptions at the start of the run. It then generates cash flow tables for the baseline and alternative systems. The inputs and calculations of the cash flows for the two systems are independent, but parallel. That is, for each input and calculation of the baseline system, there is a parallel input and calculation for the alternative system. The tables tabulate on a year-by-year basis the major costs involved in owning the systems. These costs include loan payments (capital recovery), energy costs (fuel costs), income tax savings (a negative cost due to property tax, interest and tax credits) and maintenance costs.

After the cash flow tables for the two systems have been calculated, LCCA compares the two in a third cash flow table. The total yearly cost of the alternative system is subtracted from that of the baseline system. This represents the annual savings of using the alternative over the baseline system. If the savings are negative, then the owner would lose money that year by having the alternative system instead of the baseline system. The annual savings are used to compute cumulative savings (i.e. a running total of the annual savings), present worth of annual savings (discounted savings based on present worth factor) and cumulative present worth (a running sum of the discounted savings).

Next, LCCA calculates the payback period and internal rate of return of the alternative system over the baseline system. Payback period is defined in LCCA as the year in which the cumulative savings become positive and stay positive for the remainder of the simulation. LCCA checks for a payback period, and if there is one it is printed out at the end of the analysis. A similar procedure is employed to determine the internal rate of return of the alternative over the baseline system. Rate of return is the interest (discount) rate at which the cumulative present worth is zero for the last year of the simulation. To find this rate, LCCA uses rate of return guesses supplied by the user to discount the cash flow back to present worth. LCCA uses a Newtonian convergence method to find the rate which causes the

sum of the discounted annual savings to be zero. This is the rate of return.

Several things should be noted about this procedure. The actual rate of return must be between the initial guesses or it will not be found. If it is calculated to be above the high guess or below the low guess an appropriate statement will be printed out. The procedure discounts the negative saving flows at the inputted discount rate instead of the trial rate of return. This is done, because the negative savings are costs and therefore should be discounted at the rate of return that the owner can get on other investments. Because of this consideration, the rate of return produced is considered by finance theory to be accurate only if it is not greatly different from the discount rate. LCCA prints out a warning to this effect if the rate of return is more than ten percent greater or less than the discount rate.

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1      PROGRAM LCCA (INPUT,TAPE2,OUTPUT,TAPE5=INPUT,TAPE6=OUTPUT)
      DIMENSION TOTBLN(30),GUESS(3),ANSAV(31),ALTSYS(4),BLNSYS(4)
      DIMENSION CITY(4),JOBDES(4),CUMSA(40),BFI(5,3),AFI(5,3)
      DIMENSION IDATA(12),BFC(5),AFC(5)
      INTEGER CITY,BLNSYS,YEAR,BOWNED,AOWNED,PAYEAR
      IN=2
10     IC=5
      IP=6
      REWIND IN
      WRITE(IP,243)
243    FORMAT(1H1//,35X,39HLIFE CYCLE COST ANALYSIS (LCCA) PROGRAM)
15     WRITE(IP,245)
245    FORMAT(//30H COPYRIGHT 1978 BY GARY MILLS/21H ALL RIGHTS RESERVE
      1D)
      WRITE(IP,247)
247    FORMAT(//33H PROPERTY OF REMMERS ENGINEERING/36H 175 PAWNEE DR.
20     180ULDER, CO 80303)
      WRITE(IP,240)
240    FORMAT(1H0//30H EXPLANATION OF INPUT/OUTPUT:)
      WRITE(IP,280)
280    FORMAT(6X,96HPROGRAM OUTPUT CONSISTS OF A LIST DESCRIBING THE INPU
25     1TS USED IN THIS RUN PLUS CASH FLOW TABLES FOR/6X,99HBOTH THE BASEL
      2INE AND ALTERNATIVE SYSTEMS AS WELL AS A CASH FLOW TABLE COMPARING
      3 THE TWO. THE PAY-/6X,99HBACK PERIOD AND RATE OF RETURN OF THE AL
      4TERNATIVE SYSTEM OVER THE BASELINE SYSTEM ARE ALSO OUTPUT.//)
      C...THESE LOOPS READ ALL DATA AND PRINT OUT VALUES AND DESCRIPTIONS
30     WRITE(IP,235)
      285 FORMAT(1H0/30H SYSTEM INDEPENDENT INPUTS://)
      DO 580 L3=1,16
      READ(IC,1300) (IDATA(L1),L1=1,12)
      WRITE(IN,1300) (IDATA(L1),L1=1,12)
35     WRITE(IP,1400) IDATA(5), (IDATA(L1),L1=6,12), (IDATA(L2),L2=1,4)
      530 CONTINUE
      1303 FORMAT(3A8,A4,2A2,6A8)
      1400 FORMAT(1X,A2,3X,A2,6A8,10X,3A8,A4)
      WRITE(IP,1500)
40     1500 FORMAT(1H1//27H BASELINE SYSTEM INPUTS://)
      DO 585 L3=1,19
      READ(IC,1300) (IDATA(L1),L1=1,12)
      WRITE(IN,1300) (IDATA(L1),L1=1,12)
45     WRITE(IP,1400) IDATA(5), (IDATA(L1),L1=6,12), (IDATA(L2),L2=1,4)
      585 CONTINUE
      WRITE(IP,1600)
      1600 FORMAT(1H0//30H ALTERNATIVE SYSTEM INPUTS://)
      DO 590 L3=1,19
      READ(IC,1300) (IDATA(L1),L1=1,12)
      WRITE(IN,1300) (IDATA(L1),L1=1,12)
50     WRITE(IP,1400) IDATA(5), (IDATA(L1),L1=6,12), (IDATA(L2),L2=1,4)
      590 CONTINUE
      WRITE(IN,1)
      1 FORMAT(80X)
      REWIND IN
55     C READ STATEMENTS READ DATA INTO PROGRAM VARIABLES
      READ(IN,100) (JOBDES(L1),L1=1,4), (CITY(L2),L2=1,4)

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      READ(IN,200) IYEAR
      READ(IN,300) M
60      READ(IN,400) TERM
      READ(IN,300) ITERM1, ITERM2
      READ(IN,400) RTINT, DR, GUESS(1), GUESS(2), PTAXAR, PTAXML, PTINFL
      READ(IN,300) INCTXR
      READ(IN,400) SLSTAX
65      READ(IN,100) (BLNSYS(L1), L1=1,4)
      READ(IN,900) BOWNED
      READ(IN,400) BLNCST, BLNDPT, BTXCRD, BPTAR
      READ(IN,400) BNHRUP, BLNMT, BLNMTI, BFC(1)
      READ(IN,800) (BFI(1,L1), L1=1,3)
70      READ(IN,400) BFC(2)
      READ(IN,800) (BFI(2,L1), L1=1,3)
      READ(IN,400) BFC(3)
      READ(IN,800) (BFI(3,L1), L1=1,3)
      READ(IN,400) BFC(4)
75      READ(IN,800) (BFI(4,L1), L1=1,3)
      READ(IN,400) BFC(5)
      READ(IN,800) (BFI(5,L1), L1=1,3)
      READ(IN,100) (ALTSYS(L1), L1=1,4)
      READ(IN,900) AOWNED
80      READ(IN,400) ALTCST, ALTDPT, ATXCRD, APTAR
      READ(IN,400) ALHRUP, ALTMAT, ALMTI, AFC(1)
      READ(IN,800) (AFI(1,L1), L1=1,3)
      READ(IN,400) AFC(2)
85      READ(IN,800) (AFI(2,L1), L1=1,3)
      READ(IN,400) AFC(3)
      READ(IN,800) (AFI(3,L1), L1=1,3)
      READ(IN,400) AFC(4)
      READ(IN,800) (AFI(4,L1), L1=1,3)
      READ(IN,400) AFC(5)
90      READ(IN,800) (AFI(5,L1), L1=1,3)
100      FORMAT(3A8,A4)
200      FORMAT(I4)
300      FORMAT(I2)
400      FORMAT(F8.2)
95      800      FORMAT(F8.2,2X,F8.2,2X,F8.2)
      900      FORMAT(I1)
C CONVERTS FUEL INFLATION RATES FOR THE THREE PERIODS INTO DECIMAL FORM
      DO 120 J=1,5
      DO 119 I=1,3
100      BFI(J,I)=BFI(J,I)/100.0+1.0
      AFI(J,I)=AFI(J,I)/100.0+1.0
      119 CONTINUE
      120 CONTINUE
      PTINFL=PTINFL/100.0+1.0
105      DF=DR/100.0+1.0
      DFSTOR=DF
      RTINT=RTINT/100.0
      STRDR1=GUESS(1)
      STRDR2=GUESS(2)
110 C CALCULATES CAPITAL RECOVERY FACTOR
      CRF=RTINT*(1.0+RTINT)**TERM/(((1.0+RTINT)**TERM)-1.0)
C...PRINTS OUT BASELINE HEADING
      WRITE(IP,890) (JOBDES(J), J=1,4), (CITY(J2), J2=1,4)
      890 FORMAT(1H1///32H LIFECYCLE COST ANALYSIS PROGRAM/1X,3A8,A4/1X,3A8.

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115      +A4/)
        WRITE(IP,12)
12      FORMAT(17H BASELINE SYSTEM:)
        WRITE(IP,110) (BLNSYS(I),I=1,4)
110     FORMAT(1H0,10X,3A8,A4//)
120     IF(BOWNED.EQ.1) WRITE(IP,1001)
1001    FORMAT(1H0,110HSINCE THE BASELINE SYSTEM IS CURRENTLY OWNED, UTIL
        1IZING IT IN THE FUTURE REQUIRES NO ADDITIONAL DEBT SERVICE.)
        WRITE(IP,30)
125     30      FORMAT(103HOEND      PRINCIPAL INTEREST      LOAN      FUEL      INCOME
        1PROPERTY TAX MAINTENANCE TOTAL COST      PRESENT)
        WRITE(IP,40)
140     40      FORMAT(104H OF      PAYMENT      PAYMENT      BALANCE      COST      TAX
        1ATTRIBUTABLE      EXPENSE      OF SYSTEM      WORTH OF)
        WRITE(IP,45)
130     45      FORMAT(5H YEAR,41X,60HSAVINGS      TO SYSTEM      FOR YEAR      FOR YEA
        1R      TOTAL COST/)
        M=M+1
        C...BASELINE SYSTEM CALCULATIONS.....
        ANBNPT=CRF*(BLNCST-BLNPT)
135     BLNBAL=BLNCST-BLNPT
        TOTC=0.0
        TOTM=0.0
        TOTF=0.0
        TOTPW=0.0
140     DO 50 J=1,M
        N=J-1
        C...SELECTS CORRECT INDEX FOR FUEL INFLATION RATES
        K=1
        IF(N.GT.ITERM1) K=2
145     IF(N.GT.(ITERM1+ITERM2)) K=3
        C CALCULATES TAX SAVINGS FOR BEGINNING OF FIRST YEAR
        IF(N.NE.0) GO TO 901
        BLNINT=0.0
        BNPRIN=0.0
150     BLNANF=0.0
        BLNMNT=0.0
        C CALCULATES TAX SAVINGS
        C...ONLY HALF OF BLNCST APPLIES TO SALES TAX
        BNTXSV=(SLSTAX/100.0+FLOAT(INCTXR)/100.0*BLNCST/2.)*BTXCRD
155     IF(BOWNED.EQ.1) BNTXSV=0.0
        TOTBNC=BLNPT+BNHKUP-BNTXSV+SLSTAX/100.0*BLNCST
        IF(BOWNED.EQ.1) TOTBNC=BLNPT+BNHKUP-BNTXSV
        BLNPT=0.0
        SOTBNC=TOTBNC
160     YEAR=IYEAR
        GO TO 902
        C...INFLATES THE FUEL COSTS FOR THE CURRENT YEAR
        901 DO 903 I=1,5
        903 BFC(I)=BFC(I)*BFI(I,K)
165     C...CALCULATES PROPERTY TAX INCREASE DUE TO SYSTEM
        IF(N.GT.1) GO TO 905
        BLNPT=BLNCST*BPTAR/100.0*PTAXAR/100.0*PTAXML/1000.0
        905 BLNPT=BLNPT*PTINFL
        C FUEL COST EQUATION FOR BASELINE SYSTEM
        BLNANF=0
170     DO 910 I=1,5

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910 BLNANF=BLNANF+BFC(I)
    BLNINT=BLNBAL*RTINT
    BNPRIN=ANBNPT-BLNINT
175    BLNMNT=0.0
    BLNMAT=BLNMAT*(BLNMTI/100.0+1.0)
    IF(N.GE.1)BLNMNT=BLNMAT
    IF(FLOAT(N).GT.TERM)BLNINT=0.0
    IF(FLOAT(N).GT.TERM)BNPRIN=0.0
180    BNTXSV=FLOAT(INCTXR)/100.0*(BLNINT+BLNPT)
    TOTBLN(N)=BLNANF+ANBNPT-BNTXSV+BLNPT+BLNMNT
    YEAR=N+IYEAR
    BLNBAL=BLNBAL-BNPRIN
    TOTENC=TOTBLN(N)
185    902 PWENC=TOTENC*(1/(DF**N))
    WRITE(IP,1200) YEAR,BNPRIN,BLNINT,BLNBAL,BLNANF,BNTXSV,BLNPT,
    1BLNMNT,TOTBNC,PWENC
1200    FORMAT(1X,I4.3X,F7.2,4X,F7.2,1X,F8.2,2X,F8.2,1X,F7.2,4X,F6.2,8X,
    1F6.2,5X,F8.2,6X,F8.2)
190    TOTF=TOTF+BLNANF
    TOTM=TOTM+BLNMNT
    TOTC=TOTC+TOTENC
    TOTPW=TOTPW+PWENC
50    CONTINUE
195    WRITE(IP,1210)
    1210 FORMAT(1H ,36X,8H-----,24X,8H-----,5X,8H-----,6X,8H-----
    +--)
    WRITE(IP,1220) TOTF,TOTM,TOTC,TOTPW
1220    FORMAT(1H ,36X,F8.2,24X,F8.2,5X,F8.2,6X,F8.2)
200    EUAC=TOTPW*((DF-1.0)*CF**N/((DF**N)-1))
    WRITE(IP,1230) TOTPW
    1230 FORMAT(27H0 SYSTEM LIFE-CYCLE COST: ,F8.2)
    WRITE(IP,1240) EUAC
1240    FORMAT( 61H0 LIFECYCLE COST AS AN EQUIVALENT UNIFORM ANNUAL CASH
205    +FLOW: ,F8.2)
C...ALTERNATIVE SYSTEM HEADING PRINTOUT
    WRITE(IP,890) (JOBDES(J),J=1,4),(CITY(J2),J2=1,4)
    WRITE(IP,103)
210    103 FORMAT(20H ALTERNATIVE SYSTEM:)
    WRITE(IP,110) (ALTSYS(I),I=1,4)
    IF(AOWNED.EQ.1) WRITE(IP,1002)
    1002 FORMAT(1H0,112HSINCE THE ALTERNATIVE SYSTEM IS CURRENTLY OWNED, UT
    1ILIZING IT IN THE FUTURE REQUIRES NO ADDITIONAL DEBT SERVICE.)
215    WRITE(IP,30)
    WRITE(IP,40)
    WRITE(IP,45)
C...ALTERNATIVE SYSTEM CALCULATIONS.....
    ANALPT=CRF*(ALTCST-ALTDPT)
    ALTBAL=ALTCST-ALTDPT
220    TOTC=0.0
    TOTM=0.0
    TOTF=0.0
    TOTPW=0.0
    DO 70 J=1,M
    N=J-1
225    C...SELECTS CORRECT INDEX FOR FUEL INFLATION RATES
    K=1
    IF(N.GT.ITERM1) K=2

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230      IF(N.GT.(ITERM1+ITERM2)) K=3
C COST CALCULATIONS FOR BEGINNING OF FIRST YEAR
      IF(N)920,921,920
921      YEAR=IYEAR
      ALTINT=0.0
      ALPRIN=0.0
235      ALTANF=0.0
      ALTMNT=0.0
C...CALCULATES TAX SAVINGS
      ALTXSV=(SLSTAX/100.0*FLOAT(INCTXR)/100.0*ALTCST/2.0)+ATXCRD
      IF(AOWNED.EQ.1) ALTXSV=0.0
240      TOTALC=ALTDPT+ALHKUP-ALTXSV+SLSTAX/100.0*ALTCST
      IF(AOWNED.EQ.1) TOTALC=ALTDPT+ALHKUP-ALTXSV
      ALTPT=0.0
      ANSAV(1)=SOTBNC-TOTALC
      GO TO 922
245      C...INFLATES FUEL COST FOR THE CURRENT YEAR
      920 DO 950 I=1,5
      950 AFC(I)=AFC(I)*AFI(I,K)
C...CALCULATES PROPERTY TAX INCREASE DUE TO SYSTEM
      IF(N.GT.1) GO TO 955
250      ALTPT=ALTCST*APTAR/100.0*PTAXAR/100.0*PTAXML/1000.0
      955 ALTPT=ALTPT*PTINFL
C...FUEL COST EQUATION FOR ALTERNATIVE SYSTEM
      ALTANF=0.0
      DO 960 I=1,5
255      960 ALTANF=ALTANF+AFC(I)
      ALTINT=ALTBAL*RTINT
      ALPRIN=ANALPT-ALTINT
      ALTMNT=0.0
      ALTMAT=ALTMAT*(ALMTI/100.0+1.0)
260      IF(N.GE.1)ALTMNT=ALTMAT
      IF(FLOAT(N).GT.TERM)ALTINT=0.0
      IF(FLOAT(N).GT.TERM)ALPRIN=0.0
      ALTXSV=(FLOAT(INCTXR)/100.0)*(ALTINT+ALTPT)
      TOTALC=ALTANF+ANALPT-ALTXSV+ALTPT+ALTMNT
265      YEAR=IYEAR+N
      ALTBAL=ALTBAL-ALPRIN
      ANSAV(J)=TOTBLN(N)-TOTALC
      922 PWALT=TOTALC*(1/(DF**N))
      WRITE(IP,1200) YEAR,ALPRIN,ALTINT,ALTBAL,ALTANF,ALTXSV,ALTPT,
270      1ALTMNT,TOTALC,PWALT
      TOTF=TOTF+ALTANF
      TOTM=TOTM+ALTMNT
      TOTC=TOTC+TOTALC
      TOTPW=TOTPW+PWALT
275      70 CONTINUE
      WRITE(IP,1210)
      WRITE(IP,1220) TOTF,TOTM,TOTC,TOTPW
      EUAC=TOTPW*((DF-1.0)*DF**N/((DF**N)-1))
      WRITE(IP,1230)TOTPW
280      WRITE(IP,1240) EUAC
C...PRINTS OUT HEADING FOR ALTERNATIVE AND BASELINE COMPARISON
      WRITE(IP,890) (JOBDES(J),J=1,4),(CITY(J2),J2=1,4)
      WRITE(IP,930)
285      930 FORMAT(40H ALTERNATIVE SYSTEM VS. BASELINE SYSTEM:)
      WRITE(IP,940) (ALTSYS(I),I=1,4),(BLNSYS(J),J=1,4)

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940  FORMAT (1H0,3A8,A4,5H VS. ,3A8,A4)
      WRITE(IP,310)
290  310  FORMAT(1H0,55HEND      ANNUAL      CUMULATIVE      PRESENT      CUMULAT
      +IV:)
      WRITE(IP,320)
      320  FORMAT(1X,52HOF      SAVINGS      SAVINGS      WORTH OF      PRESENT)
      WRITE(IP,330)
      330  FORMAT(1X,4HYEAR,29X,20HANNUAL      WORTH OF)
      WRITE(IP,340)
295  340  FORMAT(34X,19HSAVINGS      SAVINGS)
      CUMSAV=0.0
      CUMSVN=0.0
      C...ALTERNATIVE TO BASELINE COMPARISON CALCULATIONS
      DO 301 J=1,M
300      DF2=(1.0/DF)**(J-1)
      CUMSAV=CUMSAV+ANSV(J)
      CUMSA(J)=CUMSAV
      ANSVN=ANSV(J)*DF2
      CUMSVN=CUMSVN+ANSVN
305      YEAR=IYEAR+J-1
      WRITE(IP,345) YEAR,ANSV(J),CUMSAV,ANSVN,CUMSVN
      345  FORMAT(1X,I4,4X,F8.2,4X,F9.2,4X,F8.2,4X,F8.2)
      301  CONTINUE
      C...CALCULATES PAYBACK PERIOD
      M=M-1
310      IC=0.
      350  NYEAR=9999
      351  IC=IC+1
      IF(IC.GE.M) GO TO 354
315      IF(CUMSA(IC).GE.0.0) GOTO 352
      GO TO 351
      352  NYEAR=IC
      353  IC=IC+1
      IF(CUMSA(IC).LT.0.0) GO TO 350
320      IF(IC.GT.M) GOTO 354
      GO TO 353
      354  IF(NYEAR.NE.9999) GO TO 355
      WRITE(IP,6000)
325  6000  FORMAT(1H0//)
      WRITE(IP,600)M
      600  FORMAT(79H0THE USE OF THE ALTERNATIVE SYSTEM WILL NOT PAYBACK -PAY
      + FOR ITSELF- WITHIN THE,I3,26H YEAR TERM OF THE ANALYSIS)
      GO TO 501
      355  NYEAR=NYEAR-1
330      WRITE(IP,500) NYEAR
      500  FORMAT(22H0THE PAYBACK PERIOD IS,I3,7H YEARS.)
      C CALCULATES RATE OF RETURN
      501  BPVSAV=0.0
      GUESS(1)=GUESS(1)/100.0+1.0
335      GUESS(2)=GUESS(2)/100.0+1.0
      DF=GUESS(1)
      DO 710 N=1,20
      BPVSVN=0.0
      C NEGATIVE CASH FLOWS ARE DISCOUNTED AT THE GIVEN DISCOUNT RATE, NOT AT
340  C THE INTERNAL RATE OF RETURN
      C THIS LOOP CALCULATES SUM OF PW OF ANNUAL SAVINGS WITH THE CURRENT ROR
      DO 401 K=1,M

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345         IF(ANSAV(K).LT.0.0)BPVSVN=ANSAV(K)/(DFSTOR**(K-1))+BPVSVN
           IF(ANSAV(K).GE.0.0)BPVSVN=ANSAV(K)/(DF**(K-1))+BPVSVN
401     CONTINUE
           IF(.NOT.((BPVSVN.LE.10.0).AND.BPVSVN.GE.(-10.0))) GO TO 420
           GO TO 740
350     420   IF(N.EQ.20) WRITE(IP,210)N
           210  FORMAT(36H0SPECIFIED CONVERGENCE NOT MET AFTER,12,10HITERATIONS)
           IF (N.NE.1) GO TO 701
           GUESS(3)=GUESS(2)
           DF=GUESS(2)
           GO TO 710
355     C SELECTS NEW RATE OF RETURN (DF)
           701  IF(BPVSVN.GT.0.0) GO TO 720
           DF=(GUESS(1)+DF)/2.0
           GUESS(2)=GUESS(3)
           GO TO 650
360     720   DF=(GUESS(2)+DF)/2.0
           GUESS(1)=GUESS(3)
           650   GUESS(3)=DF
           IF(N.NE.2) GO TO 710
           IF(BPVSAV.LT.0.0) WRITE(IP,4002) STRDR1
365     4002  FORMAT(77H0THE FIRST RATE OF RETURN GUESS IS TOO HIGH. THE RATE O
           IF RETURN IS LESS THAN ,F6.2,19H PERCENT PER ANNUM.)
           IF(BPVSAV.LT.0.) GO TO 3001
           IF(.NOT.(BPVSVN.LT.(0.0).AND.BPVSAV.LT.(0.0).OR.BPVSVN.GT.(0.0).AN
           10.BPVSAV.GT.(0.0))) GO TO 710
           IF(BPVSVN.GT.0.0.AND.BPVSAV.GT.0.0) WRITE(IP,4001) STRDR2
370     4001  FORMAT(66H0RATE OF RETURN ON THE DIFFERENTIAL NET CASH FLOW IS GRE
           1ATER THAN ,F6.2,19H PERCENT PER ANNUM./104H THEREFORE, THE USE OF
           2THE BASELINE SYSTEM IN LIEU OF THE ALTERNATIVE SYSTEM IS NOT ECONO
           3MICALLY VIABLE.)
           GO TO 3001
375     710   BPVSAV=BPVSVN
           740   RTRTRN=(DF-1.0)+100.0
           WRITE(IP,750) RTRTRN
           750   FORMAT(/57H THE RATE OF RETURN ON THE DIFFERENTIAL NET CASH FLOW
           11S ,F5.2,19H PERCENT PER ANNUM.)
380     IF(RTRTRN.GT.DR+10.0) WRITE(IP,5020) DR
           5020  FORMAT(/ 77H THE RATE OF RETURN ON THE DIFFERENTIAL NET CASH FLO
           1W IS SIGNIFICANTLY LARGER/33H THAN THE GIVEN DISCOUNT RATE OF ,F5
           2.2,9H PERCENT./95H THE RATE OF RETURN CALCULATION IS NOT ACCURAT
           3E. HOWEVER, THE PRESENT VALUE FIGURES ARE VALID.)
385     3001  STOP
           END

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LIFE CYCLE COST ANALYSIS (LCCA) PROGRAM

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175 PAVNEE DR. BOULDER, CO 80303

EXPLANATION OF INPUT/OUTPUT:

PROGRAM OUTPUT CONSISTS OF A LIST DESCRIBING THE INPUTS USED IN THIS RUN PLUS CASH FLOW TABLES FOR BOTH THE BASELINE AND ALTERNATIVE SYSTEMS AS WELL AS A CASH FLOW TABLE COMPARING THE TWO. THE PAY-BACK PERIOD AND RATE OF RETURN OF THE ALTERNATIVE SYSTEM OVER THE BASELINE SYSTEM ARE ALSO OUTPUT.

SYSTEM INDEPENDENT INPUTS:

1	JOB DESCRIPTION	DOE HEAT PUMP STUDY
2	JOB LOCATION	ALBUQUERQUE, NM
3	STARTING YEAR	1979
4	TERM OF ANALYSIS (YR)	20.0
5	TERM OF LOAN (YR)	20.0
6	FIRST FUEL INFLATION PERIOD (YR)	10
7	SECOND FUEL INFLATION PERIOD (YR)	10
8	INTEREST RATE ON LOAN (PERCENT)	10.0
9	DISCOUNT RATE (PERCENT)	6.0
10	LOW RATE OF RETURN GUESS (PERCENT)	0.1
11	HIGH RATE OF RETURN GUESS (PERCENT)	50.0
12	PROPERTY TAX APPRAISAL RATE	100.
13	PROPERTY TAX MILL LEVY	63.090
14	PROPERTY TAX INFLATION RATE	5.0
15	OWNER INCOME TAX RATE (PERCENT)	20.6
16	SALES TAX RATE (PERCENT)	4.25

BASELINE SYSTEM INPUTS:

17	BASELINE SYSTEM DESCRIPTION	STAND ALONE H. P. (1.5 TON)
18	BASELINE SYSTEM PRESENTLY OWNED (YES=1 NO=0)	0
19	BASELINE SYSTEM COST (\$)	1850.
20	BASELINE SYSTEM DOWN PAYMENT (\$)	185.
21	BASELINE SYSTEM TAX CREDIT (\$)	0.0
22	SYSTEM PROPERTY TAX ASSESSMENT RATE (PERCENT)	33.3
23	UTILITY HOOKUP CHARGE (\$)	0.0
24	BASELINE SYSTEM MAINTENANCE COST (\$/YR)	92.5
25	MAINTENANCE COST INFLATION RATE (PERCENT)	5.0
26	ANNUAL FUEL COST, FIRST COMPONENT (\$)	1170.
27	FIRST COMPONENT INFLATION RATES	7.5 5.8 5.8
28	ANNUAL FUEL COST, SECOND COMPONENT (\$)	0.0
29	SECOND COMPONENT INFLATION RATES	0.0
30	ANNUAL FUEL COST, THIRD COMPONENT (\$)	0.0
31	THIRD COMPONENT INFLATION RATES	0.0
32	ANNUAL FUEL COST, FOURTH COMPONENT (\$)	0.0
33	FOURTH COMPONENT INFLATION RATES	0.0
34	ANNUAL FUEL COST, FIFTH COMPONENT (\$)	0.0
35	FIFTH COMPONENT INFLATION RATES	0.0

ALTERNATIVE SYSTEM INPUTS:

36	ALTERNATIVE SYSTEM DESCRIPTION	ANY. NO TAX CREDIT
37	ALTERNATIVE SYSTEM OWNED (YES=1 NO=0)	0
38	ALTERNATIVE SYSTEM INITIAL COST (\$)	3000.0
39	ALTERNATIVE SYSTEM DOWN PAYMENT (\$)	300.00
40	ALTERNATIVE SYSTEM TAX CREDIT (\$)	0.0
41	SYSTEM PROPERTY TAX ASSESSMENT RATE (PERCENT)	33.3
42	UTILITY HOOKUP CHARGE (\$)	0.0
43	ALTERNATIVE SYSTEM MAINTENANCE COST (\$/YR)	150.
44	MAINTENANCE COST INFLATION RATE (PERCENT)	5.0
45	ANNUAL FUEL COST, FIRST COMPONENT (\$)	1043.
46	FIRST COMPONENT INFLATION RATES	7.5 5.8 5.8
47	ANNUAL FUEL COST, SECOND COMPONENT (\$)	0.0
48	SECOND COMPONENT INFLATION RATES	0.0
49	ANNUAL FUEL COST, THIRD COMPONENT (\$)	0.0
50	THIRD COMPONENT INFLATION RATES	0.0
51	ANNUAL FUEL COST, FOURTH COMPONENT (\$)	0.0
52	FOURTH COMPONENT INFLATION RATES	0.0
53	ANNUAL FUEL COST, FIFTH COMPONENT (\$)	0.0
54	FIFTH COMPONENT INFLATION RATES	0.0

DOE HEAT PUMP STUDY
ALBUQUERQUE, NM

BASILINE SYSTEM:

STAND ALONE H. P. (1.5 TON)

END OF YEAR	PRINCIPAL PAYMENT	INTEREST PAYMENT	LOAN BALANCE	FUEL COST	INCOME TAX SAVINGS	PROPERTY TAX ATTRIBUTABLE TO SYSTEM	MAINTENANCE EXPENSE FOR YEAR	TOTAL COST OF SYSTEM FOR YEAR	PRESENT WORTH OF TOTAL COST
1979	0.00	0.00	1665.00	0.00	7.38	0.00	0.00	255.78	255.78
1980	29.07	166.50	1535.33	1257.75	41.28	40.91	37.12	1549.79	1462.07
1981	31.38	163.53	1503.35	1352.38	41.23	42.35	101.38	1651.19	1469.56
1982	35.18	160.40	1568.78	1453.49	41.08	44.99	107.08	1760.05	1477.77
1983	38.59	156.88	1530.08	1562.50	40.82	47.24	112.43	1876.92	1488.70
1984	42.58	153.01	1487.52	1679.69	40.52	49.60	119.08	2002.39	1498.31
1985	46.82	148.75	1440.71	1805.86	40.17	52.08	123.98	2137.11	1506.58
1986	51.50	144.07	1389.21	1941.09	39.75	54.69	130.16	2281.75	1517.50
1987	56.65	138.92	1332.58	2086.87	39.27	57.42	136.86	2437.06	1529.04
1988	62.31	133.26	1270.24	2243.17	38.71	60.29	143.50	2603.82	1541.20
1989	68.55	127.02	1201.69	2411.41	38.07	63.31	150.67	2782.89	1553.95
1990	75.40	120.17	1126.29	2581.27	37.33	66.48	158.21	2934.19	1549.70
1991	82.94	112.63	1043.35	2699.24	36.49	69.80	166.12	3094.24	1537.74
1992	91.23	104.34	952.12	2855.80	35.52	73.29	174.42	3263.55	1530.08
1993	100.36	95.21	851.76	3021.43	34.43	76.95	183.14	3442.87	1522.70
1994	110.39	85.18	741.37	3196.68	33.20	80.90	192.30	3632.15	1515.57
1995	121.43	74.14	619.93	3382.08	31.80	84.84	201.92	3832.82	1508.70
1996	133.58	61.99	486.35	3578.25	30.22	89.08	212.01	4044.70	1502.06
1997	146.93	48.64	339.42	3785.78	28.43	93.54	222.51	4269.07	1495.84
1998	161.63	33.94	177.79	4005.36	26.43	98.21	234.74	4506.46	1489.44
1999	177.79	17.78	-0.00	4237.57	24.18	103.12	245.43	4757.81	1483.45
				51107.07			3211.53	99116.02	30427.50

SYSTEM LIFE-CYCLE COST: 30427.50

LIFECYCLE COST AS AN EQUIVALENT UNIFORM ANNUAL CASH FLOW: 2652.81

ALTERNATIVE SYSTEM:

ANY, NO TAX CREDIT

END OF YEAR	PRINCIPAL PAYMENT	INTEREST PAYMENT	LOAN BALANCE	FUEL COST	INCOME TAX SAVINGS	PROPERTY TAX ATTRIBUTABLE TO SYSTEM	MAINTENANCE EXPENSE FOR YEAR	TOTAL COST OF SYSTEM FOR YEAR	PRESENT WORTH OF TOTAL COST
1979	0.00	0.00	2700.00	0.00	12.75	0.00	0.00	414.75	414.75
1980	47.14	270.00	2652.86	1121.23	67.24	68.18	157.50	1594.81	1504.54
1981	51.86	265.29	2601.00	1205.32	56.35	69.49	155.38	1690.37	1504.42
1982	57.04	260.10	2543.36	1295.72	56.61	72.98	173.64	1792.85	1505.31
1983	62.74	254.40	2481.22	1392.39	66.20	76.81	182.33	1902.77	1507.17
1984	69.02	248.12	2412.20	1497.36	65.71	80.44	191.44	2020.67	1509.95
1985	75.92	241.22	2335.28	1609.68	65.14	84.48	201.01	2147.14	1513.55
1986	83.51	233.63	2252.77	1730.39	64.46	88.69	211.07	2282.82	1518.20
1987	91.88	225.28	2160.30	1860.17	63.88	93.12	221.52	2428.37	1523.59
1988	101.05	216.09	2059.85	1999.58	62.77	97.78	232.70	2584.52	1529.77
1989	111.16	205.99	1948.69	2149.66	61.73	102.66	244.33	2752.07	1536.74
1990	122.27	194.87	1826.42	2274.34	60.53	107.80	256.55	2895.29	1525.20
1991	134.50	182.64	1691.92	2406.25	59.17	113.19	269.38	3048.79	1514.16
1992	147.95	169.19	1543.98	2545.31	57.61	118.85	292.35	3207.04	1503.58
1993	162.74	154.40	1381.23	2693.47	55.84	124.79	296.99	3376.55	1493.45
1994	179.02	138.12	1202.21	2849.69	53.83	131.03	311.84	3555.87	1483.74
1995	196.92	120.22	1005.29	3014.97	51.56	137.58	327.43	3745.56	1474.43
1996	216.61	100.53	788.68	3189.84	49.00	144.46	343.80	3946.24	1465.49
1997	238.27	78.87	550.41	3374.85	46.11	151.58	360.99	4158.55	1456.32
1998	262.10	55.04	288.31	3570.59	42.36	159.27	379.04	4383.18	1448.70
1999	288.31	28.83	-0.00	3777.58	39.21	167.23	397.99	4620.84	1440.80
				45559.33			5207.89	58547.04	30374.59

SYSTEM LIFE-CYCLE COST: 30374.59

LIFECYCLE COST AS AN EQUIVALENT UNIFORM ANNUAL CASH FLOW: 2648.19

ALTERNATIVE SYSTEM VS. BASELINE SYSTEM:

ANY, NO TAX CREDIT VS. STAND ALONE H. P. (1.5 TON)

END OF YEAR	ANNUAL SAVINGS	CUMULATIVE SAVINGS	PRESENT WORTH OF ANNUAL SAVINGS	CUMULATIVE PRESENT WORTH OF SAVINGS
1979	-158.99	-158.99	-158.99	-158.99
1980	-45.02	-204.00	-42.47	-201.45
1981	-39.17	-243.17	-34.86	-236.32
1982	-32.80	-275.97	-27.54	-263.85
1983	-25.85	-301.82	-20.47	-284.33
1984	-18.28	-320.10	-13.68	-297.99
1985	-10.04	-330.13	-7.37	-305.05
1986	-1.07	-331.20	-0.71	-305.77
1987	8.69	-322.50	8.45	-300.31
1988	19.30	-303.20	11.42	-288.89
1989	30.83	-272.38	17.21	-271.68
1990	38.90	-233.48	23.49	-251.19
1991	47.45	-186.02	23.58	-227.60
1992	56.52	-129.51	25.50	-201.10
1993	66.12	-63.39	29.24	-171.86
1994	76.29	12.90	31.83	-140.03
1995	87.08	99.98	34.27	-105.75
1996	98.45	198.41	36.55	-69.20
1997	110.52	308.93	39.72	-30.48
1998	123.28	432.20	40.74	10.27
1999	136.78	568.98	42.65	52.91

THE PAYBACK PERIOD IS 15 YEARS.

THE RATE OF RETURN ON THE DIFFERENTIAL NET CASH FLOW IS 6.34 PERCENT PER ANNUM.

APPENDIX F

PERFORMANCE DATA CALCULATIONS FOR HIGH EFFICIENCY HEAT PUMP

The operating characteristics of the high efficiency heat pump are defined by the following general equation

$$Y = a_1 + a_2Z + a_3Z^2 + b_1X + b_2XZ + b_3XZ^2 + c_1X^2 + c_2X^2Z + c_3X^2Z^2$$

where as shown in Table F-1

a,b,c = constants

X = air temperature entering outdoor coil, °F

Z = air temperature entering indoor coil, °F

Y = unknown term (Q_{evap} , Q_{cond} , Q_{elec} and SST) where:

Q_{evap} = The net amount of free energy extracted from the outdoor ambient through the outdoor coil when operating in the heating mode. In the cooling mode, Q_{evap} is the net energy extracted from the space after accounting for the loss associated with the indoor fan wattage.

Q_{cond} = The total amount of energy delivered to the space including contributions from the refrigeration system and indoor fan wattage when operating in the heating mode. In the cooling mode, Q_{cond} includes only the refrigerant component.

Note: It is assumed that the refrigerant compressor casing losses are absorbed by the accumulator and therefore become a part of the discharge gas. This requires that the compressor and accumulator be housed in the same compartment.

Q_{elec} = The sum total of all electrical input to the system, regardless of the mode of operation.

SST = The saturated suction temperature corresponding to the suction pressure in the evaporator. SST is used to determine the need for defrost cycling.

Note: Heating performance should be penalized for defrost cycling whenever SST has remained at or below 32°F and less than the dew point for a period of 90 minutes. If these conditions exist, the heat pump should be reversed, the outdoor fan stopped and the unit operated for 6 minutes in the cooling mode, assuming an air temperature entering the outdoor coil (X) of 72°F and an air temperature entering the indoor coil (Z) of 77°F. The defrost penalty will amount to Q_{elec} plus any energy extracted from the load including storage if configured as part of a load-side heat pump/rock bed storage system.

TABLE F-1. PERFORMANCE DATA FOR HIGH EFFICIENCY 1.5 TON AIR-TO-AIR HEAT PUMP

CONSTANT	HEATING MODE (3°F ≤ X ≤ 67°F / 68°F ≤ Z ≤ 98°F)				COOLING MODE (72°F ≤ X ≤ 117°F / 47°F ≤ Z ≤ 77°F)			
	Q _{evap} (Btu/hr)	Q _{cond} [*] (Btu/hr)	Q _{elec} (kw)	SST (°F)	Q _{evap} ^{**} (Btu/hr)	Q _{cond} (Btu/hr)	Q _{elec} (kw)	SST (°F)
a ₁	12.595 × 10 ³	17.303 × 10 ³	2.157	-7.812	15.126 × 10 ³	19.846 × 10 ³	1.574	-15.29
a ₂	-77.0	-108.5	-1.086 × 10 ⁻²	4.441 × 10 ⁻²	-103.6	-181.0	-8.067 × 10 ⁻⁴	86.17 × 10 ⁻²
a ₃	1.524 × 10 ⁻²	21.57 × 10 ⁻²	5.667 × 10 ⁻⁵	1.619 × 10 ⁻⁵	2.554	2.914	-1.619 × 10 ⁻⁵	-1.629 × 10 ⁻³
b ₁	48.26	-57.21	-2.696 × 10 ⁻²	95.13 × 10 ⁻²	-186.3	-216.3	7.429 × 10 ⁻⁴	2.247 × 10 ⁻²
b ₂	1.495	4.264	8.066 × 10 ⁻⁴	-4.969 × 10 ⁻⁴	5.638	7.266	1.286 × 10 ⁻⁴	3.266 × 10 ⁻⁴
b ₃	-6.476 × 10 ⁻³	-1.871 × 10 ⁻²	-3.619 × 10 ⁻⁶	1.191 × 10 ⁻⁶	-4.635 × 10 ⁻²	-5.543 × 10 ⁻²	0.0	0.0
c ₁	2.83	3.767	0.0	-1.583 × 10 ⁻³	61.84 × 10 ⁻²	81.81 × 10 ⁻²	0.0	2.939 × 10 ⁻⁴
c ₂	-1.533 × 10 ⁻²	4.122 × 10 ⁻²	0.0	4.004 × 10 ⁻⁶	-2.836 × 10 ⁻²	-3.445 × 10 ⁻²	0.0	1.02 × 10 ⁻⁶
c ₃	-3.143 × 10 ⁻⁵	1.212 × 10 ⁻⁴	0.0	5.87 × 10 ⁻⁸	2.201 × 10 ⁻⁴	2.674 × 10 ⁻⁴	0.0	0.0
[*] Heating capacity (assumes no degradation for defrost energy). ^{**} Cooling capacity (sensible cooling only). The effect of latent cooling is not included because when configured as part of a heat pump/rock bed storage system, most of the cooling energy will be expended to cool storage (essentially a closed container) and most of the moisture will be extracted in a relatively short time.								

APPENDIX G

**CAPABILITY OF ROCK BED STORAGE
TO DEHUMIDIFY**

by

R.E. Jones, Consultant

Objective: Determine the capability of rock bed storage as a cold storage medium to dehumidify for any length of time.

Solution: The analysis is based on the following assumptions:

- 250,000 Btu's of rock bed heat storage at $30^{\circ}\text{F } \Delta T$ (90 to 120°F); therefore, 167,000 Btu's of cooling storage at $20^{\circ}\text{F } \Delta T$ (65 to 45°F).
- Space comfort conditions— 78°F db, 65°F wb, 58°F dp; $W_{dp} = 0.0105$ lb of water/lb of dry air.
- Dew point leaving rock bed at fully charged condition = 45°F ; $W = 0.0065$ lb of water/lb of dry air.
- Two ton heat pump with 500 cfm/ton.
- Specific heat of rock = 0.21 Btu/lb $^{\circ}\text{F}$; density = 167 lb/ft 3 .
- Rock size is 3/4-in. diameter (assumed relatively spherical with 40 percent void).

$$\text{Rock volume} = \frac{4\pi r^3}{3} = \frac{4\pi \left[\frac{0.75}{2(12)} \right]^3}{3} = 1.278 \times 10^{-4} \text{ ft}^3$$

$$\text{No. of rocks/ft}^3 \text{ of storage} = (1.0-0.4)/(1.278 \times 10^{-4}) = 4694$$

- Rock surface area -

$$\text{Surface area of sphere} = 4\pi r^2 = 4\pi \left[\frac{0.75}{2(12)} \right]^2 = 1.227 \times 10^{-2} \text{ ft}^2$$

$$\text{Surface area/ ft}^3 \text{ of storage} = (1.227 \times 10^{-2})(4694) = 57.6 \text{ ft}^2$$

Two conditions exist which effectively increase the rock surface area, in other words, the moisture holding capacity of each rock prior to enough water being deposited for droplets to form and starting of elimination.

They are:

- (1) The surface crevices or pits in the rocks will capture and hold moisture due to surface tension, and
- (2) Where the rocks touch, surface tension will also form a meniscus.

It is estimated that these factors will combine to increase the effective surface area approximately threefold, assuming thin layers of water less than 0.010-in. thick.

$$\text{Effective surface area/ft}^3 \text{ of storage} = 3 (57.6) = 172.8 \text{ ft}^2$$

Assuming a flow velocity of 15 ft/min through storage at 1000 cfm, the frontal area = 66.7 ft^2 or 8.165 ft square.

$$\begin{aligned} \text{Total storage volume} &= \frac{167,000}{(0.21)(20)} = 39,760 \text{ lb} \\ &= \frac{39,760}{(167)(0.6)} = 397 \text{ ft}^3 (=7.35 \text{ ft cube}) \end{aligned}$$

$$\text{Flow length} = \frac{397}{66.7} = 5.95 \text{ ft}$$

Therefore, assuming 1-in segments in the direction of flow there are approximately 71 of these segments, each with the following characteristics:

$$\text{Volume/segment} = \frac{397}{71} = 5.56 \text{ ft}^3$$

$$\begin{aligned} \text{Actual surface area/segment} &= 5.56 (57.6) = 320.3 \text{ ft}^3 \\ \text{Effective surface area/segment} &= 5.56 (172.8) = 960.8 \text{ ft}^2 \end{aligned}$$

$$\text{Thermal storage/segment} = \left[\frac{167,000}{397} \right] 5.56 = 2339 \text{ Btu}$$

Since dehumidification ends at a rock temperature equal to the dew point (58°F), the rock ΔT is 13°F and the effective storage capacity of each segment = $\frac{13(2339)}{20} = 1520 \text{ Btu}$

At the worst case, all 1520 Btu's is dehumidifying energy

$$\text{Enthalpy (avg)} = \frac{1067.81 + 1060.47}{2} = 1064.14 \text{ Btu/lb.}$$

$$\text{Weight of water vapor (max)} = \frac{1520}{1064.14} = 1.43 \text{ lb}$$

$$\text{Volume of water vapor (max)} = \frac{1.43}{62.4} = 0.0229 \text{ ft}^3$$

If this amount of moisture were spread evenly over the entire surface area of rock, the thickness of water would be between

$$\frac{0.0229(12)}{320.3} = 8.6 \times 10^{-4} \text{ in.}$$

and

$$\frac{0.0229(12)}{960.8} = 2.9 \times 10^{-4} \text{ in.}$$

Conclusion: These thicknesses are probably not enough to form a droplet.

APPENDIX H

HEAT TRANSFER CHARACTERISTICS OF ROCKS USED FOR SENSIBLE HEAT STORAGE

Rock beds have been used extensively for thermal energy storage, particularly in conventional air-based solar systems, and it has been determined that rocks found in river beds are ideal for this application. The most common constituents of river gravel are limestone, sandstone and granite. The chart below lists the thermodynamic properties of the three types along with two terms (thermal capacitance and thermal diffusivity) which describe their heat transfer characteristics.

Material	Thermal Conductivity (k) Btu/hr-ft. ⁰ F	Average Density (ρ) lb/ft ³	Specific Heat (c_p) Btu/lb. ⁰ F	Thermal Capacitance (ρc_p) Btu/ft ³ . ⁰ F	Thermal Diffusivity ($k/\rho c_p$) ft ² /hr
Limestone	0.54	169	0.22	37.2	0.015
Sandstone	1.00	141	0.20	28.2	0.035
Granite	1.71	169	0.20	33.8	0.051

Thermal capacitance (ρc_p) is a measure of the heat content of a unit volume of rock. Thermal diffusivity ($k/\rho c_p$) is a measure of the ability of the rock to absorb the heat transferred to it by hot air passing through it and to release that heat to cooler air.

Of the three types, limestone has the highest thermal capacitance, which means it has the best capability for storing heat. It will store approximately 10 percent more heat per degree temperature difference than granite, and approximately 25 percent more than sandstone.

On the other hand, limestone has the lowest thermal diffusivity. Therefore, it will not absorb or release heat as quickly as sandstone or granite. The diffusivity of sandstone is more than twice that of limestone, and granite's diffusivity is more than three times greater than that of limestone.

Although it is impossible to determine the relative importance of thermal capacitance and thermal diffusivity without knowing the exact operating conditions of a given rock bed storage (configuration, air flow rate, temperature range, etc.), it is known that temperature stratification is critical to the successful performance of rock bed storage and thermal diffusivity is primarily responsible for this phenomenon. The higher the diffusivity, the larger the temperature gradients will be across storage. Therefore from a heat transfer standpoint, granite with its higher diffusivity is the preferred storage medium for most heat pump/rock bed storage applications.

APPENDIX I

TRADEOFFS BETWEEN OVERSIZING AND UNDERSIZING
HEAT PUMPS

By

R. E. Jones, Consultant

Sizing a heat pump is a decision based on economics as well as on physics. Unlike sizing an air conditioner where the unit is usually sized to offset the peak instantaneous cooling load (assuming no storage), heat pump sizing is not as easily defined. There are tradeoffs or penalties associated with oversizing and undersizing heat pumps. These are discussed in the following paragraphs.

Oversizing

Heating Mode. Two advantages of oversizing the heat pump in the heating mode are: 1) an increase in the yearly free energy fraction, and 2) the capability of using a deeper night setback because of the greater recuperative power offered by the higher tonnage machine. The economic benefits of these two pros can only be determined through a number of year-long simulations.

Disadvantages include an increase in the initial cost of the equipment and an increase in the cost of the distribution system (assuming there is no separate distribution system feeding the load from a storage container). There are also two operating cost disadvantages: 1) the loss of unrecoverable energy, and 2) an increase in maintenance costs. Both are attributable to the cycle that occurs in satisfying the load directly. These operating cost disadvantages are not a factor if the heat pump feeds storage, but they are if the heat pump uses the storage as a source of low level energy.

Another disadvantage of oversized equipment is excessive air movement near the thermostat. Air movement directly affects cycling and indirectly affects the comfort of the occupants. Figure I-1 shows that heat pump cycling is affected more by changes in air movement, which is effectively the same as changes in thermal capacitance (CAP), than it is by changes in the load (UA). The comfort portion of the argument is a subjective matter and cannot be evaluated other than from a marketing standpoint.

Cooling Mode. There is only one apparent advantage to oversizing the heat pump when operating in the cooling mode, i.e., the occupant is always assured of being cool. There are two disadvantages: 1) the amount of cycling increases, and 2) insufficient dehumidification occurs during periods requiring a great deal of latent cooling. The discomfort associated with the lack of humidity control can be evaluated only from a marketing standpoint. The cause, however, can be explained as follows.

- Latent gain in the space is constant and dependent on the internal sources of moisture and on the pressure difference of the water vapor inside and outside of the conditioned enclosure.
- The dew point inside the enclosure is a function of the surface temperature of the refrigerant or water coil that is the agent for energy (sensible and latent) removal from the space. The dew point cannot fall below that temperature if dehumidification is taking place, no matter how long the refrigeration unit is in operation.
- If the sensible load is satisfied, but the latent load keeps piling up, then the occupants will have a cold and clammy environment. This will generally occur on humid, cloudy days when both the temperature and solar gain are lower than design, but the latent gain is equal to the design criteria.
- The thermostat is sensitive to only sensible gains or losses, and therefore does not react to an increase in the humidity ratio. An undersized piece of equipment would be active longer to satisfy the sensible requirements, and would therefore remove more moisture.

These comfort arguments do not apply in areas with low, summertime, relative humidities such as Albuquerque.

Undersizing

Heating Mode. The only real disadvantage to undersizing the heat pump in the heating mode is that the overall yearly free energy fraction will decrease. The cost of the auxiliary fuel will determine how great the disadvantage is.

The advantages are decreased distribution network first costs, a smaller possibility of frequent cycles of the machine, and slower air movement affecting cycling and human comfort.

Cooling Mode. Undersizing can cause discomfort due to lack of capacity, however, that may be overshadowed by the increased ability to remove moisture and the small number of cycles of the machine. Undersizing only applies to systems that do not take advantage of a thermal flywheel downstream of the heat pump. If peak load shaving can be accomplished with thermal storage, then many of the above arguments become meaningless.

Another factor that affects the installed size of a heat pump is altitude. For example, the air handling equipment has to be approximately 20 percent larger at 5000 feet elevation than it does at sea level. The size of the refrigeration equipment is not affected by altitude.

If an air source heat pump is to handle domestic water heating or a forced infiltration component as well as space heating and cooling, the usual procedure of heat pump sizing based on the balance point is not valid. The capacity of the heat pump remains directly proportional to the ambient temperature outside the conditioned space, but the calculated static heat loss of the space is no longer in direct synchronization with the outdoor ambient.

In summary, the proper sizing of a heat pump is not only an engineering decision, but it is also economic and marketing sensitive, depending on the local cost of electrical and auxiliary energy. A good starting place for picking the proper size of equipment for a residential application would be about one nominal cooling ton for every 900 square feet of living area. If the cooling load is quite low, then a dual speed or partial capacity unit should be considered for proper operation in the cooling season.

HEAT LOAD ($\overline{UA}$)

- $\overline{UA} = 380$ (0 A/C per hr infiltration)
- .- $\overline{UA} = 513$ (.3636 A/C per hr infiltration)
- $\overline{UA} = 747$ (1.091 A/C per hr infiltration)

THERMAL CAPACITANCE (CAP)

- $\triangle$ CAP = 3364 Btu/ $^{\circ}$ F (slow air motion)
- $\triangle$ CAP = 1682 Btu/ $^{\circ}$ F (medium air motion)
- $\triangle$ CAP = 841 Btu/ $^{\circ}$ F (fast air motion)

FIGURE I-1. EFFECT OF HEAT LOAD AND THERMAL CAPACITANCE ON HEAT PUMP CYCLING

APPENDIX J

SIZING CALCULATIONS FOR BASELINE

STAND-ALONE HEAT PUMP SYSTEM

Table J-1 summarizes the cooling load of the baseline house as calculated using the method of Manual J, Load Calculations for Residential Winter and Summer Air Conditioning published by the National Warm Air Heating and Air Conditioning Association (NWAHACA). The design conditions used in the calculations were:

No. of Occupants - 4

Building Volume - 13,330 ft³

Ventilation - One air change per hour

Electric Light & Appliance Load - 1.2 kw

Indoor Air - 77°F db; 50% relative humidity

Outdoor Air

Albuquerque - 95°F db; 65°F wb; medium daily temperature range

Madison - 90°F db; 75°F wb; medium daily temperature range

Ground Water Temperature - 53°F

TABLE J-1. HEAT GAIN SUMMARY FOR BASELINE HOUSE*

HEAT SOURCE	HEAT GAIN (BTU/HR)	
	ALBUQUERQUE	MADISON
<u>Sensible Heat:</u>		
Conduction - walls	6440	4640
roof	1500	1080
basement	-1270	-1270
Solar gain	12490	10830
Ventilation (infiltration)	3680	2660
Electric lights & appliances	4100	4100
People	900	900
Total RSH	27840	22940
<u>Latent Heat:</u>		
Ventilation (infiltration)	---	3230
Appliances	---	820
People	---	900
Total RLH	0	4950
Total Cooling Load	27840	27890
ARI Capacity Multiplier**	0.75	0.69
ARI Standard Rating	20880 (1.74 tons)	19240 (1.60 tons)
<p>*Based on calculations performed in accordance with Manual J, NWAHACA. **Based on a 60F indoor temperature swing.</p>		

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