EVALUATION OF THE SOLAR BUILDING
ALBUQUERQUE, NEW MEXICO

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ABSTRACT

During portions of the 1974-75 and 1975-76 winter heating seasons, a field evaluation was made of a solar-assisted heat pump heating system in a small commercial office building in Albuquerque, N.M. The system was comprised of one main water-to-water heat pump and five small water-to-air heat pumps. The liquid-type solar collector array had an area equivalent to about 10% of the building floor area. Other than the ethylene glycol/water solution circulated through the solar collector array, water was used in all parts of the system, including three thermal energy storage tanks.

Considerable information concerning this project has been disseminated through conferences, workshops, technical papers at professional society meetings, reports to the federal government and Master of Science theses, all of which are referenced in this report. This report summarizes the work done on this project over the period of the contract and includes pertinent information concerning the building, the solar-assisted heat pump system, data acquisition aspects, results and conclusions.

INTRODUCTION

In 1956 the first solar energy assisted heat pump commercial office building in the world was completed and occupied by Bridgers and Paxton Consulting Engineers, Inc.; designers, owners and since that time occupiers of the Solar Building, Albuquerque, N.M., (Fig. 1). Performance data were collected during the 1956-57 winter heating season and reported (Ref. 1). At that time, fossil fuels were so plentiful and at such low cost that the system was not economically viable. After a few years of operation the solar portion of the system was deactivated.

With the renewed interest in solar energy and the lack of actual performance data from solar energy systems, the U.S. National Science Foundation awarded a grant in April 1974 to The Pennsylvania State University, Stanley F. Gilman, Principal Investigator, to conduct a field evaluation of the Solar Building. With the formation of the U.S. Energy Research and Development Administration in January 1975, financial support was provided by ERDA and subsequently by the Solar Energy Division of the Department of Energy (DOE) up to September 30, 1978.

The first step was to refurbish the system and put it back into operation. It was then instrumented so actual performance data could be collected by a data acquisition computer and transmitted to The Pennsylvania State University for analysis. The refurbishing was implemented through a subcontract to Bridgers and Paxton Consulting Engineers, Inc. who were at the site. Instrumentation and on-site data collection supervision was effected by a subcontract with the University of New Mexico.

The principal objectives of the project were:

1. Determine the actual building system performance characteristics by collecting and analyzing field collected operating data.
2. Correlate the building system operation with computer predicted performance.
3. Utilize the computer model to analyze the performance and potential of such systems.

FIG. 1 The Solar Building, Albuquerque, N.M.

4. Develop information that consulting engineers and others could use to design solar energy assisted water-source heat pump systems for non-residential buildings.

Considerable information has been disseminated through the media of conferences, technical papers presented at professional society meetings, reports to ERDA and Master of Science theses (Refs. 2-27). Therefore, this report will summarize many of the findings and refer the reader to more specific information in appropriate major references. Aspects of the project not previously reported will be discussed in detail.

BUILDING AND SYSTEM

The Solar Building has a solar-heated floor area of 5,000 ft² that is separated from a conventionally heated and cooled portion by means of fixed partitions and doors equipped with door closers. The exposed part of the south wall of the building is inclined at an angle of 55 degrees from the horizontal and covered with single-glazed steel frame window units (Fig. 1). The 55 panels which comprise the collector are mounted behind the windows on a wood frame filled with insulation and covered on the back with gypsum board. The panels are aluminum Roll-Bond with integral channels but the ethylene glycol/water coolant solution passes through 3/4 inch diameter copper tubes affixed in a serpentine pattern to the backs of the panels by means of mechanical clips and an aluminum-filled epoxy bonding material. Use of the copper tubes was necessitated by development of numerous small leaks in the channels of the aluminum panels during the 1956-57 heating season, when they were used with untreated tap water. All panels are connected in parallel to the collector piping system. The 30 percent/70 percent by weight ethylene glycol/water solution is circulated through the panels at a rate of 0.04 gallons per minute per square foot of panel surface to produce a temperature rise of about 15°F when solar energy is absorbed and removed from the panel at a rate of 300 Btuh/ft².

A plan view of the building (Fig. 2) shows the subdivisions of the floor area and locations of the major components of the heating and cooling system and the air distribution system. The major duct system, which supplies mixed return and outside air to most occupied spaces, is connected

Fig. 2. Floor Plan of the Solar Building

to the air handling unit (AHU) located in the equipment room. The AHU is supplied with heated water from the tanks or from the water-to-water heat pump for heating and with chilled water from the heat pump for cooling. A schematic piping diagram is presented in Fig. 3.

During the first heating season of this project, 1974-75, only the original underground 6,000 gallon tank, designated Tank 1, existed. Hence, it was used both for storage of solar energy and as heat source and sink for the heat pump. For the second heating season, two above-ground tanks, designated Tanks 2 and 3, with capacities of 2,131 gallons and 2,861 gallons, respectively, were added with the sole purpose of storing solar energy. These tanks were added so that the temperature of solar storage could rise as high as possible, while the temperature of Tank 1 could be maintained relatively low to serve as a source and sink at a more suitable temperature for the heat pump. Energy is exchanged between the collector fluid and water from one of the tanks by means of a shell and tube heat exchanger.

The compressor, chiller and condenser in the right portion of Fig. 3 constitute the water-to-water heat pump which served the principal portion of the building.

Five small water-to-air heat pumps are used to maintain comfort control in various offices and the computer room. These units are connected in parallel to a pipe loop through which water from Tank 1 is continuously pumped. These units draw air primarily from the spaces they condition, but two of them draw part or all of their air from the main duct system to assure a continuous supply of fresh air to these spaces. The units provide either heating or cooling, under control of an automatic changeover thermostat. Since the computer room required almost continuous cooling, the net energy flow from the water-to-air heat pump loop to storage was nearly always positive.

The evaporative water cooler (upper right corner of Fig. 3) is used to reject excess heat from Tank 1 during the winter heating season as necessary to maintain its temperature in a suitable range to serve as a source and sink for the heat pumps. The upper limit for the winter heating season was about 100°F.

Fig. 3. Schematic Piping Diagram

Useful solar energy is collected by circulating fluid through the collectors whenever the temperature sensed on the back of one of the collector panels by one bulb of a differential temperature sensor exceeds the temperature of the glycol/water solution leaving the heat exchanger by about 10°F. Collection ceases when this temperature differential drops below about 2°F by opening a by-pass valve and turning off the collector fluid pump.

As shown in Fig. 4, during the 1974-75 heating season solar energy was stored in Tank 1, after having been transported to the tank by water returning from the heat pump or the AHU. As shown in Fig. 5, during the 1975-76 heating season it was normally stored in Tank 2 and/or Tank 3 depending on the storage capacity to be exercised, but provision was made to divert solar energy to Tank 1 in case its temperature dropped too low.

The modes of operation of the main heating and cooling system are determined by the temperature sensed at the main thermostat located in the southeast corner of the drafting room (Fig. 2). The pneumatic thermostat provides for three active winter modes of operation, including two stages of heating and one stage of cooling, depending on the branch pressure at the thermostat. A "dead band", or intermediate mode, is provided between heating and cooling with a temperature differential of about 1.5°F. In the first heating stage, i.e., the lowest temperature range of the thermostat, the heat pump absorbs heat from water circulated from Tank 1 through the evaporator and supplies heat to the air handling unit (AHU) through a closed loop connecting that unit to the condenser (Fig. 6). This mode is termed "Heat Pump Heating". During the second heating season, provision was made to supply the heat pump with energy from Tank 2 and/or 3 if the temperature of Tank 1 dropped too low. Simultaneously, solar energy storage was switched to Tank 1 to raise its temperature to a minimum level.

The second stage of heating provides for circulation of water directly from storage to the AHU, and is termed "Direct Heating". During the first heating season, Tank 1 served as the source (Fig. 7). During the second heating season Tank 2 and/or 3 served as the source (Fig. 8). As a result of experience gained during the first season, provision was made to override the Direct Heating mode during the second season if the source

Fig. 4. Solar Collection Circuit for 1974-75 Heating Season

Fig. 5. Solar Collection Circuit for 1975-76 Heating Season

HEAT PUMP HEATING

Fig. 6. Heat Pump Heating Mode

DIRECT HEATING 1974-75

Fig. 7. Direct Heating Mode 1974-75

DIRECT HEATING 1975-76

Fig. 8. Direct Heating Mode 1975-76

tank temperature dropped below about 80°F. It was found that the direct heating mode satisfied the building's heating requirements at night during the second heating season with storage temperatures as low as 85°F.

The cooling stage provides for absorption of energy from air flowing through the air handling unit by connecting this unit in a closed loop with the heat pump evaporator (Fig. 9). Operation of the heat pump in this mode is termed "Winter Cooling", and it occurs whenever building external and internal heat gains exceed heat losses. The associated outdoor dry-bulb temperature is termed the Balance Point Temperature (BPT). The BPT varies widely with non-residential buildings, with 20°F outdoor temperature not being unusual. Residences ordinarily have balance point temperatures of the order of 65°F, which is the basis of the Degree-Day method of predicting energy consumption for residences. With the high daily range of outdoor dry-bulb temperature in climates like Albuquerque, commercial buildings will typically require cooling during parts of days for much of the fall and spring and frequently in the winter periods.

In a commercial application such as the Solar Building, a heat pump system will often be in the cooling mode during some of the occupied period of the building, and in the heating mode during the unoccupied period, particularly at night. By incorporating thermal energy storage, the heat pump system can effectively remove heat from the building as required during the occupied period, transfer this heat to the storage system, and then utilize this heat to satisfy the building nighttime heating requirements. The "Winter Cooling" mode puts the heat absorbed from the building into the storage tank and thus raises the temperature level of the tank. Normally, heat is rejected from the heat pump condenser to Tank 1, and stored there for later use. However, if the temperature of this tank rises above 100°F, energy is rejected from it to the outdoors. This was accomplished by turning on the evaporative water cooler and circulating an ethylene glycol/water solution between it and the heat exchanger (Fig. 10) (1974-75) or by circulating water from the heat pump condenser to the evaporative cooler when the heat pump is cooling or directly from Tank 1 (Fig. 11) (1975-76). Heat rejection ceases when the temperature of Tank 1 drops below about 90°F. It is necessary to maintain the temperature of the energy source for the heat pumps below approximately 100°F to avoid actuating their high pressure cut-off switches.

HEAT PUMP COOLING

Fig. 9. Heat Pump Winter Cooling Mode

Fig. 10. Evaporative Water Cooler Heat Rejection 1974-75

EVAPORATIVE WATER COOLER HEAT REJECTION 1975-76

Fig. 11. Evaporative Water Cooler Heat Rejection 1975-76

INSTRUMENTATION

Instrumentation which measures temperatures, water flow rates, electric energy consumption and insolation was used to obtain the data necessary to determine energy flow, component performance, system performance and thermal loads on the building. Temperature sensors used to determine thermal energy flow in the system were inserted in thermal wells located in the water lines at the inlet and outlet of each water-to-air heat exchanger in the system and at the inlets and outlets of the storage tanks. The sensors were linearized thermistor pairs having a maximum tolerance on accuracy of $\pm 0.66^{\circ}\text{F}$, but pairs which agreed within 0.33°F were selected to measure water temperature differences across heat exchangers to increase precision in determining heat flow rates. The wells protruded into the flow stream at least 2 inches. Thermistor temperature sensors were also used to determine air inlet and outlet temperatures at heat exchangers through which air flow occurred. Due to lack of uniformity of air temperature and velocity distributions, these observations were normally used qualitatively to verify modes of operation, rather than quantitatively to evaluate energy flow rates.

To provide data pertinent to building heating and cooling loads, indoor dry bulb temperature was monitored at five locations, and indoor dew point temperature was monitored at one location. Outdoor dry bulb and dew point temperatures were monitored at one location using sensors located inside a weather station mounted on top of the building. Dew point temperatures were determined by means of probes utilizing a temperature sensor located in the center of two concentric electrically-heated bobbins covered by wicks impregnated with lithium chloride.

Incident solar fluxes were measured by means of three pyranometers. One pyranometer was oriented horizontally; the other two were oriented parallel to the collector. One of the latter two was covered with a shadow band, so it normally received only diffuse radiation. The original bulb-type pyranometers were replaced by Eppley black and white pyranometers in March, 1975 when two of the original instruments failed. The latter instruments were checked twice in the horizontal plane against more precise Eppley PSP pyranometers, once at the beginning and again at the end of their use.

Water flow rates through the heat pump evaporator and condenser, the air handling unit, the water side of the heat exchanger between the collectors and the system, and the evaporative water cooler were measured by calibrated rotameters. These measurements were normally accurate to within approximately 1/2 gallon per minute or better than 3.5 percent. The principal contributing factor to uncertainty was minor fluctuations in flow rate which occurred even during steady state operation. Visual observation of these meters was employed, since flow rates were found to be nearly constant over periods several days long for a given mode of operation. Average flow rates through individual water-to-air heat pumps were obtained by regular visual observations of integrating water flow meters at least twice daily over periods of time at least several days long. These rates were checked against short term average flow rates which were measured occasionally using the same meters. The total flow rate through the water-to-air heat pump loop was obtained by summing the flow rates through individual heat pumps and checked using an energy balance.

Electrical energy use by the water-to-water heat pump compressor motor, the several pump and blower motors, computers, water-to-air heat pumps, and by all lights and single phase appliances combined, and by all three-phase equipment combined, was obtained by means of watt-hour meters. Automatic read-out of meter output was obtained by using a mechanical contact device attached to the meter disk drive. Periodic sampling of the state of a set of contacts on each meter produced a count of the number of disk revolutions occurring in a given period of time. Multiplication of this count by the meter constant yielded the energy use. Average power consumption was obtained by dividing the energy used during a given time period by the elapsed time.

More detailed information is provided in Ref. 14.

DATA ACQUISITION

An IBM System 7, Model A10, computer with twin disk pack storage and a bisynchronous communication module was selected as the data acquisition system. This system was selected because it was being used successfully in other monitoring projects of a similar nature, principally by Professor Sepsy

of Ohio State University, and because field-developed programs for data acquisition applications were available. It is a very powerful system, since it is fully programmable, its computational speed is high and its disk storage capacity is large.

The computer sampled the inputs, digitized the data and stored it temporarily in core. Inputs were sampled at intervals ranging from 1/2 second for rapidly varying quantities such as wind speed and direction to 6 minutes for room temperatures, which varied slowly. Most inputs were sampled at intervals of one, ten or sixty seconds. The data from each sensor were summed to produce six-minute totals, termed "records", prior to storage in disk. Thus, the minimum time resolution of the recorded data was six minutes. The six-minute records for all items of data were formed into files after several hours of continuous data collection. Twelve-hour long files were used in 1974-75 and three-hour long files in 1975-76. To avoid confusion in reducing the data and recovering it from permanent storage, only continuous files were retained; so if data collection was interrupted for any reason prior to completion of a file, all the data in the file were lost. This produced excessive losses of data in the first heating season, for reasons to be discussed later; so the file lengths were reduced thereafter.

To reduce the rental charges for the data acquisition computer, no provision was made for on-site data reduction. Hence, the only data output that could be obtained locally from the data acquisition system as originally configured was the most recent set of samples, in the hexadecimal number system. An associated decision was to transmit all the data via the telephone system to The Pennsylvania State University computer system for permanent storage and subsequent manipulation, display, reduction and analysis. Although an IBM 1130 could have been available for on-site data reduction at the initiation of the project, the building owner anticipated replacing it soon, so no provision was made for its use. Thus, the only way to obtain a comprehensive view of the data was by transmitting it to Penn State, printing it on hard copy and returning it by mail to the site.

This type of arrangement may be satisfactory for monitoring if no problems occur, but it was found to be quite unsatisfactory for initial

check-out and for diagnostic work. For these purposes, quick output of recently-sampled data is essential, because the utility of data for trouble-shooting decreases rapidly with time. Also, it was found to be desirable to have data from redundant sensors for purposes of checking the sensors' outputs. And, it is desirable to have essentially simultaneous data from several sensors that measure related variables, for purposes of computing an energy balance on an item of equipment or a subsystem. Finally, to gain an understanding of system and component operating patterns, it is desirable to output data from a few key sensors at regular periodic intervals, or upon command, for trouble-shooting. To fulfill this need, a data logger was borrowed from the Mechanical Engineering Department at The University of New Mexico and installed at the site early in the check-out phase. It remained on site for the duration of the first year of the project, at which time it was replaced by a data logger purchased with project funds. Although only a few essential channels of temperature data were monitored with the data logger, it proved to be a valuable complement to the capabilities of the data acquisition computer for purposes of check-out, diagnostics and quick scans of the operating pattern and the status of system operation. Thus the importance of providing a capability for local output of data at regular, periodic intervals and instantaneously, upon command, was learned.

In retrospect, it would have been desirable to transfer the data to an on-site or nearby computer, at least for conversion to engineering units and display. This would have permitted inspection of a more complete set of data by on-site personnel at an earlier time. This, in turn, would have facilitated diagnosis of problems by enabling earlier correlation of a broader set of observed events with the data. Also, transfer of the data to Penn State on magnetic tape would have aided in reducing losses of data associated with the transmission process. Details of the data acquisition and processing procedure are presented in Ref. 13 and 17.

DATA TRANSMISSION AND LOSSES

Data were transferred from disk storage on the System 7 to the Penn State central computer and subsequently to storage on 9-track tape. Each set of data was also stored on a second tape to avoid losses of data due to

destruction or erasure of the first tape. Transfer of data took place at 120 characters per second and was effected by a Bisynchronous Communications Module attached to the System 7 computer. Only complete data files, comprised of 12 hours of continuous six-minute records in 1974-75 and three hours of records in 1975-76, were transmitted. Since it was desired to make provision for temporarily saving the System 7 disk storage files that had been transmitted, pending their output and inspection at Penn State, it was necessary to mark the files on disk storage according to whether they were "empty," and therefore ready to receive data, or "filled," and therefore ready for transmission, or "transmitted," but being held temporarily pending inspection. Also, provision was made to automatically and immediately retransmit any portion of a file that was not received properly at the other end. If successful transmission was not achieved after 5 attempts, a time-out error occurred, which halted transmission.

In general, this scheme for transmitting data and marking files yielded quite satisfactory results when the interface between the data acquisition computer and the telephone system performed well. However, for about three months at the start of data acquisition in 1974-75 and again in the summer of 1975, serious problems were encountered with this interface. At times, transmission would occur without interruption. At other times, it was essentially impossible to transmit data, due to failure to establish an adequate connection, or due to time-out errors, or due to cessation of operation of the System 7. These failures were frequently interactive, thus tending to compound the difficulty of transmitting data on some occasions. They were interactive in the sense that time-out errors tended to occur when it was difficult to achieve a satisfactory telephone connection, and repeated attempts to reestablish communication and transmit data tended to cause the System 7 to fail. Shut-down of the System 7 prior to completion of a file caused loss of data, due to the facts that incomplete files were not saved and that no data were collected when the computer was shut down. Data losses due to failure to complete a file increase with increasing length of the file, so the 12-hour file length used in 1974-75 was reduced to three hours in 1975-76. The longer file length in 1974-75, combined with the difficulties encountered with transmission early in this period, caused significant losses of data. The losses due to unplanned

shut-downs were increased by the occasional need to shut down the System 7 to permit diagnostic work on it. Whenever possible, the computer was shut down just after a file had been completed, but this was not always possible, particularly with the 12-hour file length, due to the need for key personnel at both ends of the line to be available. Very little data were lost from storage or in the actual transmission process, due to the capability of the computer to automatically store information on disk in the event of a power failure and due to the provision to re-mark and save files that had been transmitted, if it was felt that the transmission had been unsatisfactory. The principal problems with the transmission process were noisy telephone connections and a lack of mutual understanding of interface requirements on the part of the computer manufacturer and telephone company personnel.

TECHNIQUES FOR CHECKING AND VALIDATING DATA

The simplest technique for validating data, and one that is extremely useful in monitoring systems that should operate continuously, is use of redundant sensors. Implementation of redundancy checks may require use of duplicate sensors or instruments, but such checks can be performed by comparing readings obtained during certain operating modes from sensors that are not redundant in the usual sense. For example, the readings from two temperature sensors in a fluid stream on opposite sides of a heat exchanger may be compared under conditions in which flow continues but negligible heat is exchanged. Such conditions occur when an item of equipment cycles off, but the system remains in the same basic mode of operation. This type of redundancy check was implemented both during normal system operation and during special check runs, to be described later. It was especially helpful to be able to perform these checks during normal system operation by observation of data output from the data logger at least once daily. It was found that enough cycling occurred to permit checking most of the critical temperature sensors in this manner. Use of six-minute average data returned from Penn State was helpful in performing redundancy checks on additional sensors.

Longer-term checks were performed on the readings obtained from the watt-hour meters by performing visual readings over periods a few hours in length and comparing the differences in these readings with the energy

use indicated by the automated data acquisition system for the same period. Shorter term checks were performed by counting revolutions of the watt-hour meter disk for six minutes and comparing with the computer output. These checks are on the data acquisition system alone, and do not constitute redundancy checks. Although it was planned to implement redundancy checks on water flow rate readings by comparing outputs from turbine flow meters with rotameter readings, only one turbine meter was obtained late in the second heating season, so these checks were performed in a very limited fashion. However, it was possible to perform redundancy checks on three of the rotameter readings during much of the first heating season, since two of the water streams joined to form a mixed stream, and there was a rotameter in each of the three lines. These checks were performed frequently during the first six weeks of the 1974-75 heating season, when visual readings were made hourly during occupied hours due to lack of receipt of watt-hour meters with pulse outputs. These readings also established that the various water flow rates for each of the modes of operation were relatively stable, which alleviated the perceived need for automated measurement of water flow rates. But it would have been helpful to have had such measurements at times, such as when portions of the control system malfunctioned.

Since redundancy checks were not performed on electrical power, it was considered doubly important to implement, wherever possible, energy balance checks to validate simultaneous measurements of temperature, flow rate and electrical energy use. This was felt to be the case particularly for the water-to-water heat pump. However, initial attempts to obtain an energy balance on this unit lasted for several weeks. In the process, every temperature sensor, watt-hour meter and water flow meter associated with this heat pump was recalibrated or checked by all the measures available. By the time the problem producing the failure to achieve an energy balance was identified and remedied, the necessity for performing energy balances had been deeply engraved in the project team's thinking. The problem was found to be primarily due to a missing check valve and not the instrumentation.

An energy balance on the subsystem of water-to-air heat pumps, termed the heat pump loop, was considered important, since this loop was a major

source of thermal energy for the 6,000 gallon storage tank. This occurred due to the continual need for cooling the computer room. Such a check was performed by measuring the water flow rate through, and the temperature difference across, each water-to-air heat pump, to evaluate the rate of heat flow to or from the unit. The net heat flow to the water obtained by algebraically summing the heat flow rates to the five units was compared with the heat flow obtained by multiplying the temperature difference across the entire loop by the total water flow rate through the loop, to check the energy balance. Although the difference in temperature between the loop inlet and outlet was frequently on the order of 2°F , this check generally yielded satisfactory results due to careful selection and insertion of the temperature sensors at the inlet and outlet of the heat pump loop.

Because of difficulty in obtaining an initial energy balance on the water-to-water heat pump, a procedure of performing special runs to check out critical items of instrumentation was adopted. These runs were subsequently performed at fairly regular intervals during a period when the building was not normally occupied, such as after noon on Friday. The first step was to manually force the system into a certain mode of heat pump operation to maintain operation of the pumps by resetting the main thermostat. The power to all blowers was maintained, while the power to all heat pump compressors was cut off. This step resulted in negligible heat exchange to and through all components except the collector loop, the heat exchanger between the collectors and system, and the storage tank for solar energy. After transients in the temperatures of the air and water streams had died out, it was possible to perform redundancy checks on the temperature sensors at the inlets and outlets of all heat pumps, the water-to-air heat pump loop and the heat exchanger in the main air handler. If these checks proved unsatisfactory, the sensors were checked further, frequently by interchanging the locations of sensors that should have agreed, but did not. Placement of most of the sensors in loose fitting wells filled with oil facilitated such interchanges. Faulty sensors were replaced until satisfactory agreement within 0.33 degrees F was achieved. Following completion of the redundancy checks, power flow to all compressors was resumed. Continuous operation of the heat pumps in either the heating

or cooling mode was maintained by adjusting thermostat settings until steady state operation of each unit was achieved, as indicated by steady temperature readings. Flow rates observed visually during these runs were also recorded. The steady state data on temperatures, electrical energy use and flow rates obtained from these runs were used to perform energy balances on the water-to-water heat pump and the water-to-air heat pump loop. Following completion of a check run, the system was returned to normal operation.

Energy balance checks were performed occasionally on the thermal storage tanks by observing the inlet and outlet temperatures for a period of a few hours, using the temperature difference together with the water flow rate through the tank to compute a net energy flow rate and comparing the result with the change in internal energy computed from the water storage capacity and temperature change of the tank. Near-uniform tank temperature permitted use of the outlet temperature for the tank temperature.

The only critical component on which an energy balance check was not performed was the heat exchanger between the collector loop and the main system. This type of check was not possible, since the turbine flow meter planned for use in measuring the collector fluid flow rate was never obtained. The potential for errors in the heat flow rate through this heat exchanger was reduced by observing redundant temperature measurements at both the inlet and outlet of the water side of the heat exchanger.

All of the checks described above were performed throughout data acquisition periods. Subsequent to acquisition of much of the data, extensive computer-based data analysis programs were developed, the results of which permitted observation of redundancy and energy balance checks for periods of normal operation. Six-minute data were used in these analysis programs, and results were produced in terms of six-minute and hourly energy flow rates and performance factors and hourly average temperatures.

Examples of the results obtained from these programs are presented in the format of energy balances in Figures 12, 13, 14 and 15. Figure 12 shows a comparison between values for the net heat flow rate to the water-to-air heat pump loop, using the two methods for evaluating this heat flow rate described above. Each point represents a corresponding pair of hourly total values computed from the data obtained during a 40 day period in 1976. If

FIG. 12 Energy Correlation for Water-to-Air Heat Pump Loop.

FIG. 13 Energy Correlation for Water-to-Water Heat Pump Condenser.

FIG. 14 Energy Correlation for Water-to-Water Heat Pump Chiller.

FIG. 15 Calculated vs Measured Temperatures for Tank 1.

perfect agreement existed, all the points would lie along a 45 degree line. The scatter observed in the data is considerable, but does not seem unusual, considering the low differences observed between the heat pump loop inlet and outlet temperatures and the transients which occurred in these temperatures due to equipment cycling.

Figure 13 shows a comparison of thermal energy output and the sum of the thermal and electrical energy inputs to the water-to-water unit, for the heat pump heating mode of operation. It is observed that the data exhibit a systematic deviation between the output and input energies. Similar disagreement was observed in the results from the special check runs, which involved steady-state conditions. The difference could be due in part to thermal losses from the compressor, since the input energy is higher than the output. But an estimate of such losses indicates that this mechanism could not account for all of the difference. Hence, this difference remains unexplained.

Figure 14 shows a comparison between the heat input to the water-to-water heat pump evaporator and the difference between the heat output from the condenser and the electrical input to the compressor, for the heat pump cooling mode of operation. Fewer data points appear, since less cooling operation occurred during the portion of the year to which this data pertains. The agreement is considered satisfactory, except for the points which lie along a line with a slope of about three. These points were obtained during a period when a portion of the control system malfunctioned, so that the water flow rates which occurred were quite different from those input manually to the data analysis program, which were the normal values.

Figure 15 displays plots of measured (uniform) tank temperature and the temperature computed from an energy balance on the tank. Although differences developed between the computed and measured temperatures, particularly on date 51, the two curves exhibit the same behavior. The difference developed due to drift of a temperature sensor. Thus the agreement is considered to be acceptable, in general.

The above results indicate that, despite the transients which occurred in normal operation, the temperature, flow rate and electrical energy data obtained were generally valid. But it would have assisted in obtaining

valid data if there had been a capability for performing automated data analysis on a continuous schedule during data acquisition to help identify problems.

Pyranometer outputs were checked using a comparative test, twice during the project. One such check was performed about midway through the first heating season when a new set of pyranometers was installed, after obviously invalid readings were obtained from the used instruments originally installed. The new instruments were Epply Black and White pyranometers. This check consisted of placing all the pyranometers side by side on a horizontal plane having an unrestricted view of the sky. Their outputs were compared with each other and with the output of an Epply PSP Pyranometer. The agreement was within about three percent initially, and the difference was only slightly larger at the end of data collection, about 14 months later. A check of the output at non-zero tilt angle was not performed. Subsequently, it was learned that black and white pyranometer output is sensitive to tilt angle.

A detailed report on data losses and sensor faults is presented in Ref. 15 and of operational experiences in Ref. 20. The principal conclusion reached in Ref. 20 was that the price of good data is continual vigilance, using personnel at the monitoring site who are thoroughly familiar with the operating patterns of the HVAC system and the characteristics of the data acquisition system. A second major conclusion was that a variety of checks must be utilized to maintain acquisition of valid data. These checks include observations of data from redundant sensors and calculation of energy balances. To achieve the steady state conditions essential for performing rigorous energy balances, it may be necessary to override an automatic control system; and this may require setting aside test period intervals. A third major conclusion was that capability for prompt local output of recently-sampled data in an easily-interpreted form is essential for diagnostic work. It is also helpful to have capability to output all data locally on a continuing, periodic, schedule. This aids in gaining an understanding of system behavior more rapidly and therefore aids in interpreting results and diagnosing problems. Finally, when transmission of data over a distance is necessary, it is desirable either to provide a mechanism for temporarily storing all transmitted data until satisfactory transmission has been ascertained in the

case of telephone transmission, or to transmit the data on a permanent medium such as magnetic tape.

RESULTS

Component Performance

Solar Collectors. During refurbishing of the system prior to installing instrumentation, it was discovered that many of the copper tubes affixed to the aluminum tube-in-sheet absorber plates either had leaks or the bond between the tubes and the plate was defective. Consequently, all panels were removed, inspected and, when necessary, repaired and re-installed. Since the construction of copper tubes bonded to tube-in-sheet plates is far from optimum, it was not expected that the performance of this collector array would be as good as designs of today. It was, therefore, important to determine the performance of the collector array as a separate component.

The results of this evaluation are reported in Ref. 16 and are summarized in Fig. 16. It is clear from the figure that the performance of the Solar Building collector array (B and P curve) is substantially below that of solar collectors available today. The data, however, are based on a gross area of 750 ft² and performance would have been improved if the data had been based on the net area of 570 ft². In addition, the ethylene glycol/water heat exchanger is included in the results.

The poor collector performance in no way affects the end results. As will be discussed subsequently, for this building, system and climate, the operating problem was typically an excess of thermal energy rather than a deficiency.

Performance data for the 1974-75 and 1975-76 winter heating test periods are presented in detail in Ref. 19. For the first test period, the average collector efficiency was 40% for those hours during which thermal energy was being collected, which was approximately 30% of the possible sunshine hours. For the second test period (1975-76), the average collector efficiency was 43% for those hours during which thermal energy was being collected, which was approximately 19% of the possible sunshine hours.

FIG. 16 Solar Collector Performance Comparison

As pointed out above, the operating problem was typically an excess of thermal energy rather than a deficiency. When the storage tank temperature reached about 100F during the 1974-75 heating season, solar collection was stopped by the automatic controls. Storage temperatures higher than 100F would prevent the heat pumps from rejecting heat into storage during the typical daytime cooling mode operation. The weather during the second test period was unusually mild and this is reflected in about 50% less solar collection operating time (19% of possible collection time vs 30% for the first test period.)

Water-to-Water Heat Pump. One of the important factors influencing the performance and economic viability of solar energy assisted heat pump systems is the Coefficient of Performance (COP) of the heat pump when "lifting" thermal energy from a low temperature level to a higher temperature level. The heating COP is defined as the condenser heat output divided by the heat equivalent of the compressor power input, in consistent units, as COP is dimensionless. A detailed explanation is contained in Ref. 7.

The heat pump COP for heating mode operation as a function of condenser leaving water temperature minus chiller entering water temperature, is shown in Fig. 17. This temperature difference is convenient for design and computer modeling of the system, as the condenser leaving water temperature is essentially the AHU entering water temperature, and the chiller entering water temperature is essentially the storage tank leaving water temperature. Also, the abscissa closely approximates the log mean temperature difference, or "temperature lift" of the machine. The solid line is a least squares fit of the data points and is very nearly linear. Each data point represents a minimum of 10 contiguous six-minute periods in which the heat pump operated. The set of points covers a time span of 203 days in 1974-75 and 137 days in 1975-76. The longest contiguous period was 430 six minute periods; i.e. 43 hours. The COP values are for the heat pump only and do not include auxiliary energy such as water pump power.

The heating COP varies from about 3.5 to 4.7, with a median of approximately 4.2. Thus, the water-to-water heat pump supplies heating energy to the building at approximately $1/4.2$ or 24% of the electrical energy which would be required by an electric resistance heating system. This would

FIG. 17 Heating COP for Water-to-Water Heat Pump.

be the maximum percentage as considerable energy can be supplied during the Direct Heating mode when the storage tank used for direct heating is at or above approximately 85F as a result of solar energy collection and internal building loads.

Water-to-Air Heat Pumps. In addition to the main water-to-water heat pump that served a large portion of the building, there were five water-to-air heat pumps that handled particular areas. Two of these had nominal cooling capacities of 0.8 tons of refrigeration, two had nominal cooling capacities of 1.5 tons and one had a nominal cooling capacity of 3.3 tons. The last was installed by the project team to replace a gas-electric packaged roof top unit that was serving the drafting room expansion area (west 31 ft portion, Fig. 2). By so doing, the extra complication of one unit not being a heat pump was avoided. All water sides were connected into one water piping loop to and from the underground storage tank.

Each water-to-air heat pump was individually controlled by an electric automatic changeover thermostat in the particular space being served; thus, any of the five heat pumps could be on "heating", "cooling" or "off" at any time. This substantially complicated the collection and analysis of data. However, there were some favorable aspects. Some of the heat pumps rarely operated. One, a 1.5 ton unit, served the computer room; and with the addition of the data acquisition computer and its resultant power input and hence heat release, this heat pump operated continuously on the cooling cycle. This was the predominant source of energy into the storage system; in fact, the net energy into the piping loop was always positive; i.e. heat was always being added to the loop, and produced about a 2F rise in water temperature from the inlet to the outlet. As pointed out previously, the operating condition was typically an excess, rather than a deficiency, of thermal energy, and the data acquisition computer heat release added to this excess.

The catalog data of the two different manufacturers of these heat pumps were used to calculate the heating COP as a function of inlet water temperature. The manufacturers' nominal ratings are certified by the Air-Conditioning and Refrigeration Institute (ARI). The heating COP values are very close to 3.0 over a range of water temperatures. As further checks on overall performance, the 3.3 ton unit was tested in the Thermal Laboratory

at The Pennsylvania State University prior to being installed in the Solar Building. Also, some relatively short period data were obtained on site for some of the heat pumps (Ref. 11).

Thermal Storage. During the first test period (1974-75) the system operated only with the underground storage tank. Prior to the second test period (1975-76) it was decided to add two additional thermal storage tanks. For ease of installation, these were installed above ground and heavily insulated with two inches of foamed polyurethane. Two tanks rather than one were selected so the effect of variable storage volume could be evaluated. Unfortunately, due to delivery delays and other problems, the project was well into the second test period before the above ground tanks were operative.

The logic behind the additional tanks was the following: With the typical daytime cooling cycles there was a need by the heat pump for a low temperature sink for easy heat rejection and to maintain high COP. On the other hand, there was a need for a high temperature source for direct heating of the building to minimize the amount of operation in the Heat Pump Heating mode and thus increase the overall annual efficiency of the system, i.e., the Seasonal Performance Factor (SPF). This logic is particularly applicable to commercial building systems that have a low balance point temperature; i.e., require periodic cooling operation during much of the winter heating season.

The 1975-76 test period weather turned out to be very mild and excess energy again showed up as a problem. The underground tank, which had been selected as "sink" because of the existing piping from the heat pumps, gained so much energy that its temperature remained high and periodically rose to a level that required the evaporative water cooler (Fig. 11) to operate and reject heat to the ambient. In due course, the evaporative water cooler was operated continuously in an attempt to substantially lower the storage temperature. However, with the thermal energy pouring in from the water-to-water and water-to-air heat pumps, little temperature reduction was achieved.

As to the above ground tanks, due to a misunderstanding about the intended control system modifications, the temperatures in the tanks were limited to about 100F; i.e., the same maximum temperature allowed for the underground tank for the first test period.

Because of the above, no conclusive results were obtained regarding the potential of a multiple tank, multiple temperature level system. However, purely on an engineering basis, consideration should be given to such a concept.

It could be argued that one tank would do as well if provisions were made for appropriate temperature stratification. When the above ground tanks were being installed they, as well as the underground tank, were equipped with temperature measuring devices spaced approximately four inches apart in a vertical arrangement. Several vertical piping connections were made to the two above ground tanks (Fig. 3) so water could be diverted into or out of each tank at selected vertical levels. All piping was deliberately designed using normal water piping practice; i.e., approximately four feet per second water velocity.

The conclusion from the data on storage tank temperature was that essentially no stratification occurred in any of the tanks. Data are presented in Ref. 19.

It is conceivable that water temperature stratification could be effected by devices such as diffuser sections to reduce inlet velocities and turbulence, or by elaborate baffling, but at an increase in capital cost.

System Performance

The Solar Building system was designed to be a stand-alone system; i.e., to not require auxiliary thermal energy input at any time. In actuality, the system did require auxiliary heat for a four-hour period during an unusual cold spell, during which the computer room heat source was not available due to frosting of the heat pump coil.

Fig. 18 depicts a spell of cold weather and considerable cloudiness for Albuquerque. The average daily temperature was about 25F over the period with a low of 11F just before daybreak on the third day, which temperature is below the ASHRAE design temperature for this locality. The tank water temperature fell from 85F to 58F during this period, or a net loss of 1,350,000 Btu from the system. Fortunately, the next few days were sunny and mild, so the system regained much of its lost energy.

FIG. 18 Typical Performance During Cold Weather.

The performance over a four-day period of intermediate weather is shown in Fig. 19. During this period, the daily average outdoor temperature was approximately 40F. The lower portion of Fig. 19 shows the incident solar radiation and the daily solar collection efficiency. The upper portion depicts the variation in water storage tank temperature, which is an index of the thermal energy input or output from storage. During the period shown, the system held its own very easily and, in fact, had a net increase of storage tank temperature of about 8F, equivalent to 400,000 Btu.

Comparison of the Figs. 18 and 19 tank water temperature variations illustrates the complicated interrelationship of variability of weather and system energy usage. In Fig. 19, the water tank temperature was at or above 85F for approximately half of the time. For these conditions, the heating requirements could be satisfied much of the time by the Direct Heating mode; i.e., with the heat pump off. Thus, only relatively little auxiliary power (pumps and AHU fan) was required by the system to supply heat to the building.

In contrast, in Fig. 18, the tank water temperature is at 85F only at the beginning of the period, and practically all of the building heating requirements during this period were supplied under the heat pump heating mode; i.e., with heat pump power being consumed as well as auxiliary pump power.

As previously stated, generally in this building system there was an excess, rather than a deficiency, of thermal energy. During the first test period (1974-75) the temperature of the underground water storage, which served as a sink during the heat pump cooling mode, was 90F or above at the beginning of the day (midnight) on 65 days out of the 148 day test period. During the second test period (1975-76), the temperature was 90F or above on 74 days of the 135 day test period. Referring to Table 1, which is discussed in detail below, Col. 4 shows this tank temperature for a portion of the first test period. As pointed out above, an unusually cold spell was experienced during the first week of January and the computer room heat source was accidentally disabled. As the month progressed, however, the storage tank temperature regained relatively high levels. This is shown in Col. 4, Table 1 for the period Date 16 (January 16) to Date 23.

FIG. 19 Typical Performance During Mild Weather.

TABLE I

-1- DATE	-2- HPH-AHU KWH	-3- DH-AHU KWH	-4- THK1 KWH	-5- SYS-AHU KWH	-6- COMP KWH	-7- HTG ON HRS	-8- HPH ON HRS	-9- AHU KWH	-10- P1 KWH	-11- P2 KWH	-12- TOT AUX KWH	-13- SYS1 HPP
353.	417.04	30.56	82.8	133.49	28.69	7.00	6.20	5.62	3.12	1.86	10.60	3.3977
354.	365.57	101.43	84.7	136.83	27.04	10.20	4.90	7.34	4.00	1.47	12.89	3.4267
355.	455.11	103.57	84.4	163.69	33.44	12.20	6.30	8.70	4.00	1.89	15.55	3.3408
356.	435.70	97.62	84.8	156.26	30.92	12.80	6.40	9.22	5.12	1.92	16.26	3.3121
357.	889.00	85.18	79.0	205.43	64.55	15.10	10.90	10.87	6.04	3.27	20.18	3.3687
358.	1359.17	20.15	72.9	406.48	97.42	17.00	15.80	12.82	7.12	4.74	24.68	3.3293
359.	1641.08	0.0	64.6	481.07	127.87	19.00	19.80	14.26	7.92	5.94	28.12	3.0040
360.	1499.72	1.15	58.7	439.75	116.57	18.30	18.20	13.18	7.32	5.46	25.96	3.0055
361.	1192.31	0.40	63.0	349.61	92.16	14.50	14.40	10.44	5.80	4.32	20.56	3.1016
362.	1621.63	0.0	62.1	475.13	125.96	20.30	20.30	14.62	8.12	6.09	20.83	3.0696
363.	1696.24	0.0	57.1	496.99	133.95	20.00	20.00	14.40	8.00	6.00	28.40	3.0612
364.	603.06	0.0	54.4	200.37	57.06	9.40	9.40	6.77	3.76	2.82	13.35	2.8458
365.	1104.41	0.0	55.8	347.03	92.84	14.60	14.60	10.51	5.84	4.38	20.73	3.0555
366.	1174.41	0.0	48.0	344.10	87.52	14.00	14.00	10.08	5.60	4.20	19.00	3.2040
2.	838.02	0.0	53.3	245.77	64.12	9.90	9.90	7.13	3.96	2.97	14.06	3.1439
3.	022.38	0.0	57.2	240.96	59.83	0.70	8.70	6.26	3.48	2.61	12.35	3.3300
4.	1582.50	0.0	58.5	463.67	122.47	18.60	18.60	13.39	7.44	5.58	26.41	3.1143
5.	1523.71	0.83	61.1	446.69	115.00	18.60	18.50	13.39	7.44	5.55	26.30	3.1398
6.	1269.24	10.40	65.9	374.93	91.40	16.90	16.00	12.17	6.76	4.80	23.73	3.2566
7.	1291.81	0.02	64.2	378.74	96.19	16.60	16.50	11.95	6.64	4.95	23.54	3.1632
8.	923.26	17.47	66.8	275.63	69.23	14.20	12.50	10.22	5.68	3.75	19.65	3.1011
9.	520.01	63.59	70.1	170.99	37.22	12.60	6.80	9.07	5.04	2.04	16.15	3.2036
10.	1334.01	0.0	64.0	390.86	97.52	15.00	15.80	11.30	6.32	4.74	22.44	3.2503
11.	2032.72	0.0	58.1	595.50	144.00	22.90	22.90	16.49	9.16	6.87	32.52	3.3741
12.	2129.40	0.0	53.9	623.93	150.77	24.00	24.00	17.28	9.60	7.20	34.00	3.3754
13.	544.13	0.0	56.4	160.95	42.41	6.40	6.90	4.97	2.76	2.07	9.80	3.0830
14.	746.34	42.56	70.0	231.15	40.92	12.10	9.10	8.71	4.84	2.73	16.28	3.5449
15.	795.74	107.49	77.3	264.64	49.86	13.50	9.70	9.72	5.40	2.91	18.03	3.8981
16.	637.02	53.52	82.6	202.33	37.91	11.00	7.70	7.92	4.40	2.31	14.63	3.8511
17.	485.26	3.39	89.2	143.17	27.04	6.10	6.00	4.39	2.44	1.80	8.63	4.0141
18.	633.94	79.40	88.0	209.01	36.50	12.40	8.20	8.93	4.96	2.46	16.35	3.9546
19.	508.44	11.08	89.2	152.22	28.73	6.70	6.30	4.82	2.68	1.89	9.39	3.9929
20.	417.04	0.0	92.3	122.43	22.86	4.00	4.00	2.88	1.60	1.20	5.68	4.2097
21.	745.23	14.34	87.0	222.55	42.41	9.00	8.50	6.48	3.60	2.55	12.63	4.0436
22.	102.30	83.52	87.2	77.08	10.84	5.60	2.30	4.03	2.24	0.69	6.96	4.3759
23.	14.22	24.76	94.0	11.42	0.86	1.40	0.40	1.01	0.56	0.12	1.69	4.4755

34599.59 969.74 10421.69 2512.95 474.30 420.50 341.49 189.72 126.15 657.36

TOTAL SYSTEM AHU KWH 10421.69
 TOTAL HPH-AHU KWH 10137.59
 TOTAL DH-AHU KWH 284.13

TOTAL COMPRESSOR KWH 2512.95
 TOTAL AUXILIARY KWH 657.36
 TOTAL SYSTEM 1 PERFORMANCE 3.29

At temperature levels of 95F to 100F, depending on the control differential, it was necessary to cease solar collection and, with the water-to-air heat pump loop further adding thermal energy, periodically actuate the evaporative water cooler (Figs. 10 and 11) and reject heat to the ambient in spite of the fact that the storage volume was over 10 gal per sq ft of net solar collector area. There are several reasons for the excess energy. First, the system was deliberately designed to be a stand-alone system; i.e., not require auxiliary heat. Second, the data acquisition computer heat release imposed a continuous cooling load of about 2 tons of refrigeration on the system. Third, during the second test period, the mild weather was a significant factor.

The relative proportions of thermal energy supplied to the building by Direct Heating and Heat Pump Heating is a function of the building, the system and the weather. In order to determine the actual amount of each, every six minute data period (240 periods per day) had to be analyzed for its major mode of operation during that period; i.e., Direct Heating, Heat Pump Heating, Heat Pump Cooling or Off. In order to accomplish this, a Mode Decision Flow Diagram was developed. For example, if during a six-minute data period, the main heat pump compressor showed less than 0.10 kWh power consumption (less than 10% operating time) it was presumed that the dominant mode was other than Heat Pump Heating or Heat Pump Cooling. The two remaining possible modes were Direct Heating or Off. Considering the tolerances of the temperature sensors and other factors, a minimum temperature rise of 2F across the air handling unit (AHU) water coil was selected to confirm Direct Heating. The operational mode computer program, as well as the data reduction and analysis programs, are described in Appendix C of Ref. 23.

Once the mode was determined, the analysis program computed the amount of thermal energy delivered to the building during the 10 six-minute data periods constituting one hour and also determined the compressor, pump(s) and fan power required to deliver this amount of thermal energy to the building. Summations were then made to obtain daily performance. For any period of time the Heating COP of the water-to-water heat pump was

$$HCOP = \frac{\text{Heat Pump Heating Energy Supplied, kWh}}{\text{Heat Pump Power Consumption, kWh}}$$

and for the system as a whole, the Performance Factor was

$$PF = \frac{\text{Thermal Energy Supplied to the Building, kWh}}{\text{Power Consumption of Heat Pump and Auxiliaries, kWh}}$$

Some daily results are shown in Table 1 for the 1974-75 heating season. The second column is the heat pump heating output to the air handling unit and the third column is direct heating. Note that from Date 362 through Date 4 there was no direct heating. This was the time of unusually cold weather and low tank temperature that was previously pointed out. The fifth column is the total system output for the day expressed in kilowatt hours, followed in the sixth column by the measured compressor kilowatt hours. There is a fan in the air handling unit and two water pumps that use electric power; values are shown in Col. 9, 10 and 11 with the total shown in Col. 12. By summing the compressor energy use and the total energy use of the auxiliaries, a system COP can be determined by dividing Col. 5 by the total power, Col. 12. Values are shown in Col. 13. Results are summarized at the bottom of the table. During this period, 97 percent of the heat supplied to the building was by Heat Pump Heating. Despite this, the overall System Performance Factor was 3.29. Energy use of the auxiliaries was 20 percent of the total compressor power consumed and is an important factor to consider in the design of systems.

Over the first test period which lasted 148 days beginning in Mid-December 1974, the average COP of the main heat pump was 4.1 and the System Performance Factor was 4.3. Heat supplied to the building was 27,560 kWh or 94×10^6 Btu, of which 54% was supplied by Heat Pump Heating and 46% by Direct Heating. Pump and fan power consumption was 43% of the total, again indicating the importance of minimizing auxiliary power in order to increase the SPF.

For the mild winter during the second (1975-76) test period, which was also a slightly shorter test period of 135 days beginning in Mid-December, heat supplied to the building was 16,940 kWh or 58×10^6 Btu, which was only 62% of that supplied during the first test period. Of this heat, 83% was supplied by Heat Pump Heating and only 17% by Direct Heating. Auxiliary power consumption was 30% of the total. The average COP of the main heat pump was 4.1, identical with the first test period. However, the system

Seasonal Performance Factor was 3.6, about 80% of that determined for the first test period. This lower SPF is directly attributable to the lower amount of operating time in the Direct Heating mode and larger amount of operating time in the Heat Pump Heating mode.

From the above it is concluded that:

1. To obtain a high Seasonal Performance Factor, a Direct Heating Mode is very desirable for such systems. From the characteristics of heat pumps and commercial building loads, a multiple tank or multiple temperature level, thermal storage system has the potential of considerably improving the overall performance of such systems.
2. Auxiliary power for fans and pumps can seriously affect the SPF and must be minimized in the system design.

Building Performance

A comprehensive analysis of the Solar Building energy usage vs its thermal load was made for both winter heating test periods and is reported in a M.S. thesis (Ref. 23). Building energy usage was examined with emphasis on the time of day energy was consumed and the source from which the energy emanated; i.e., from lights, office equipment, thermal storage and the several heat pumps. The rate of electrical consumption was found to be very dependent on the building use. High rates of electrical energy usage during occupied periods required cooling during parts of even the coldest days. Heating by the mechanical equipment was found to vary as a function of building usage as well as a function of the indoor-outdoor temperature differential.

Energies supplied to and withdrawn from the building were calculated and analyzed for hourly, daily and seasonal periods for both winter season test periods.

Energy losses and gains from the building to the surroundings were analyzed for both steady-state and transient load profiles. Envelope conduction heat losses and losses due to infiltration and ventilation were calculated using actual weather data input to the Building Energy Analysis Program (BEAP) of the Architectural Engineering Department of The Pennsylvania State University. This computer program was developed several

years ago under a National Science Foundation grant and has been improved and refined systematically since then.

The effect of building thermal storage on heating and cooling loads was examined and a series of building Balance Point Temperatures (BPT) was established. Comparisons between actual building energy consumptions and calculated loads were made for hourly, daily and seasonal periods. Agreement in most areas was found to be acceptable.

Conclusions and recommendations resulting from this portion of the project, and reported in Ref. 23, are:

Heating energy dissipated within the building spaces, as a result of high rates of electrical energy use for lighting and for operation of office equipment, drastically reduced the heating energy which had to be supplied by mechanical equipment. During almost all daytime periods electrical energy use resulted in energy being dissipated which exceeded the building load requirements even during the coldest periods. Since the mechanical heating and cooling equipment was equipped with a dead band between the heating and cooling modes, the space temperature could rise approximately five degrees before cooling operation began. The excess energy caused the interior space temperature to rise and resulted in substantial internal energy storage in walls, floors, and furnishings. The internal storage of energy reduced the required cooling during occupied periods and reduced mechanical heating during unoccupied periods. As the building temperature fell from the cooling setpoint, internally stored energy was released to the building spaces and thereby reduced the amount of heating energy which had to be supplied by mechanical equipment.

After the indoor space temperature rose, mechanical cooling began. In this mode of operation the heat pump units withdrew thermal energy from the interior spaces and rejected the energy to the water thermal storage. This operation allowed the storage of excess thermal energy remotely for later use.

When mechanical heating was necessary, energy was withdrawn from the water storage. During periods with low or moderate heating loads and high water source temperatures, the stored water could be circulated directly; i.e., direct heating, to satisfy the building load. During other periods thermal boosting of heating energy was performed in the heat pump heating mode.

Thus, excess energy during daytime periods was rejected to water storage to maintain proper occupied space temperatures and withdrawn from the water storage to maintain desired night space temperatures. This system, equipped with water source heat pumps and water thermal storage, maintained building comfort conditions at substantially lower cost than could have been achieved using electric resistance heating and electric air conditioning without thermal storage. In addition, the water-to-water heat pump operated very efficiently with an average coefficient of performance of 4.2 without auxiliaries (Ref. 16).

Throughout the winter heating season most of the energy supplied by the heating equipment was available as a result of previous cooling. The remainder of the heating energy supplied was available as a result of solar collection and necessary electrical energy input to the mechanical equipment.

Total energy available from internal loads, heat pump compressor operation, and solar collection exceeded the heating requirements of the building; and the building required no auxiliary heating system. After January of both test periods, average rates of energy supplied to storage as a result of mechanical equipment cooling exceeded that necessary for mechanical equipment heating. The addition of necessary compressor and fan energy for heat pump operation to the energy stored from previous mechanical cooling reduced the need for the solar system further. Approximately 80% of mechanical heating was performed in the heat pump heating mode. Using an average compressor coefficient of performance (COP) of 4.2, an additional 19% energy was available as a result of electrical energy dissipated by the compressor. Applying this factor for heat pump compressor operation to the average hourly rate of mechanical heating and cooling for January 1975, December 1975, and January 1976, the energy available from previous cooling together with necessary compressor energy exceeded the heating requirements for all except January 1975.

The solar energy collection system did provide some advantage by helping to supply energy during cold periods when the building was unoccupied, and only limited source energy was provided by mechanical equipment cooling.

Thermal loads were successfully calculated using both steady-state and transient procedures through the use of the Building Energy Analysis Program

(BEAP). The steady-state thermal load calculations, although not providing a precise accounting of energy for a particular hourly or daily period, did provide a clear representation of average thermal load. These steady-state calculations were primarily used for determination of the building balance-point temperatures and for comparing the net energy use to the average building thermal load as a function of the indoor-outdoor temperature differential. Using the seasonal average indoor-outdoor temperature differential and comparing the seasonal average hourly net energy usage and steady-state calculated heating load using five days occupied, two days unoccupied, and a 10 mph wind speed, the steady-state calculations predicted a load within 7% of that measured. Transient load calculations performed using actual weather data were also found to be within 10% of that measured.

Variations observed between the calculated transient heating load and measured net energy usage were largely caused by the storage and re-release of thermal storage within the building and by variations in infiltration rates. For short periods of time, one hour to 12 hours, thermal storage would cause substantial differences between net energy usage and calculated heating load. For the daily period, the thermal storage had a reduced effect. When viewed over a monthly or seasonal interval, the effect due to building thermal storage tended toward zero. Infiltration, on some occasions, produced large discrepancies between calculated and measured energy flows.

Infiltration resulted in increased energy losses which increased the building load. Although care was taken to take into account varying infiltration rates, which included the measurement of infiltration using tracer gas techniques (Ref. 12), variations in building use, indoor-outdoor temperature differentials, wind speed, and wind direction did affect its magnitude in unpredictable ways. Since the wind speed used was that measured at the Albuquerque Airport Weather Station, some variations did occur between the building site and the airport about three miles away. Also, daily averages of wind speed used in calculation of thermal loads did not account for variations within the daily periods. Wind direction also had an effect. For this study the predominant wind direction was used. As a result, most of the daily discrepancies between the actual daily load and the calculated daily load were attributed to poor agreement between actual and calculated infiltration rates.

Building balance-point temperatures were identified for occupied and unoccupied day, night, and daily periods. Using an indoor temperature of 75°F, an average wind speed of 10 mph, and allowing all electrical energy to offset the heating load, the following outdoor balance-point temperatures resulted.

<u>Period</u>	<u>Outdoor Temperatures</u>
Day (occupied)	17°F
Day (unoccupied)	47°F
Night (unoccupied)	52°F
Daily (24 hour with occupied day)	38°F
Daily (24 hour with unoccupied day)	50°F

This field study showed the solar collection system not to be efficient because of the building's high internal energy sources, warm daytime temperatures, and large amounts of solar insolation. The climate in Albuquerque, New Mexico, has large daily ranges, often 30°F to 40°F with nights below freezing and typical daytime temperatures between 40°F and 50°F. This, accompanied with solar radiation, reduced the daytime heating loads to low levels. The high rates of internal heat gains, especially lighting levels, and associated cooling resulted in large amounts of energy being supplied to storage to satisfy nighttime loads. The efficient transfer of energy to the thermal storage showed the water source heat pump system combined with water storage alone to be an excellent system.

Thus, for this building, the solar collection system was not very effective because other energy sources could and did supply most of the needed heating. The high internal heat generation capability resulted in large amounts of energy which could be reclaimed and stored by the building's heat pumps and water thermal storage. The water-to-water and water-to-air heat pumps, combined with water thermal storage alone, provided a system which could cool as well as heat. This system resulted in large energy savings by providing an excellent method of energy recovery.

Recommendations resulting from this thesis (Ref. 23) project were:

The lighting system was responsible for much of the internal heat gain during occupied hours. Although the heat pump system efficiently cooled

spaces which became overheated and stored this energy for later use, only for a few weeks each year was all this excess energy utilized. During all other periods, high lighting levels resulted in excess heating energy. Reduction in electrical energy use for lighting would provide monetary savings which would be realized in three ways. First, reduction in lighting levels would reduce the cost of electrical energy consumed for lighting. Second, the reduced heating of spaces would reduce the cooling requirements with a corresponding reduction in heat pump electrical energy usage especially during summer periods. Third, a reduced cooling load would result in a reduction in the size of the mechanical cooling equipment and thereby reduce its cost. The switching from high levels of general area lighting to high levels only at task locations may result in a significant reduction in electrical energy consumption. Further study is needed to assess work performance and comfort resulting from such changes.

If all the energy rejected by mechanical cooling could have been stored and then made available to supply heating loads at a later time, dependence on solar energy would have been reduced further. This building's thermal storage range, however, was not large enough to accommodate long-term storage. (The water storage facility was sized for approximately one-day's energy storage.) Periodic energy rejection to the environment was necessary to maintain proper operating temperatures of thermal storage for heat pump cooling operation. Consequently, one or two days of severe weather conditions and little building use would result in depletion of the thermal storage. Additional studies of storage sizing and resultant system performance are recommended.

A detailed building analysis, preferably including a computer simulation, should be performed during the design stage of a building. Transient computer simulations should be made, if possible, to predict peaks, duration, and frequency of heating and cooling loads. Use of building materials with large thermal storage capabilities should be considered to reduce both heating and cooling peaks as well as for storing excess daytime energy which would reduce both daytime cooling and nighttime heating. A solar-assisted heat pump system would not be economically attractive and, therefore, should not be recommended for buildings whose internal loads require significant cooling during most occupied periods. A water source heat pump combined with

water thermal storage provides an excellent heating and cooling system in these cases. Large energy savings are possible with this type of system as a result of energy recovery characteristics.

Buildings with lower internal loads which require additional energy input during much of the heating season may be well suited for a solar-assisted heat pump system. For this type of building the energy recovery capabilities of the heat pumps and energy collection capabilities of the solar collectors may complement each other. Thus, a solar-assisted system may be economically attractive and should be considered.

This field study was located in Albuquerque, New Mexico. The area is typical of high altitude, hot and dry climates with predominately clear, sunny weather and with large daily temperature ranges. This study should be extended to similar buildings in other climates as well as to buildings with low internal loads in order to accurately assess the energy savings potential of heat pumps and solar-assisted heat pump systems with thermal storage in other climatic regions of the United States.

Computer Modeling and Correlation

One objective of the project was to try to computer model the building and system and correlate the simulation with field collected data. The system used was that for the first test period; i.e., with the single underground storage tank. Details of this aspect are presented in Refs. 19 and 22.

The program organization is shown in Fig. 20 and the steps are:

Step 1: Load Calculations. The WTHRED (Weather Data Reader) program was developed to read weather data tapes, but it is not limited to tapes. Through commands input to the program, the user defines the format and the order of the required data. The weather tapes used are modified NOAA Test Reference Year (TRY) weather tapes.

The WEATHR (Weather Data Generator) program was developed through a National Science Foundation grant to The Pennsylvania State University some years ago. It is based on monthly average values of outdoor dry-bulb temperature and dew point, wind velocity and average daily solar insolation (Ref. 25).

FIG. 20 Program Organization.

The BEAP (Building Energy Analysis Program) load calculation and energy consumption program has been under continual development and updating in the Architectural Engineering Department of The Pennsylvania State University since the late 1960's. Additions to this program specifically made for this project were: 1) glass shading devices, 2) summer/winter design conditions, 3) workday/weekend and holiday temperature and load profiles, 4) improved ventilation/infiltration load calculations, and 5) interface capability with WEATHR, WTHRED, and TRNSYS.

Step 2: System Simulation. Version 8.1 of TRNSYS, the University of Wisconsin developed simulation program, was modified to interface with the other programs. Hourly system simulation occurs until steady state is reached in that hour, then the next hour is simulated. New models were developed for several components, including the heat pumps, liquid pumps and control valves.

The initial phase was to model the components. After this the entire system was modeled, data generated and then compared with field collected data for selected time periods.

Solar Collector. The solar collector array is comprised of aluminum absorber plates painted black with copper tubes soldered to them and covered with a single pane glass. A sample comparison of modeled vs actual solar collection is shown in Fig. 21. The collector entering water temperature was 94F at solar collection start-up. Since the storage tank water (collector entering water temperature) was high the fraction of incident solar radiation collected for that period was only 23%.

Storage Tank. The insulated underground storage tank holds 6000 gallons of water, which corresponds to about 10 gal/sq ft of collector. Actual versus modeled storage tank operation is shown in Fig. 22. The correlation between the modeled and actual tank leaving water temperatures is very good. The tank is modeled as non-stratified. During model testing, two and three layer stratification modes were modeled, but the single layer (non-stratified) provided the best model. Test data on the three storage tanks with thermistors at each 4-in. increment of height also confirmed no temperature stratification when utilizing common water piping design practices, such as standard fitting selections and water velocities (Ref. 19).

FIG. 21 Solar Collector Model Comparison.

FIG. 22 Storage Tank Model Comparison.

Water-to-Water Heat Pump. The water-to-water heat pump (HP#1) is a reciprocating compressor packaged water chiller having a nominal cooling capacity of 7-1/2 tons. The model assumes the cooling or heating capacity of the evaporator or condenser is the effect produced at the coil in the air handling unit, i.e., there are no piping heat gains or losses. In actuality, the heat pump and the air handling unit are very close together. The air handling unit fan heat is included in the analysis. A plot of the heat pump model heating capacity against evaporator entering water temperature is shown in Fig. 23. The entering water temperature (EWT) is the evaporator EWT for the heating mode and the condenser EWT for the cooling mode.

Water-to-Air Heat Pumps. The five water-to-air heat pumps in the Solar Building are identified as follows:

	Model	Nom.Clg. (Btuh)	Nom.Htg. (Btuh)	Nom. CFM	Nom. GPM
HP#2	CHP-100	10,000	8,900	350	2.0
HP#3	CHP-1.5	19,000	16,200	600	3.5
HP#4	CHP-1.5	19,000	16,200	600	3.5
HP#5	CHP-100	10,000	8,900	350	2.0
HP#6	HW-40	40,000	47,000	1,200	9.0

All of the units have the same basic operating characteristics and the heating and cooling performance data files have the same requirements as the water-to-air heat pump model. Performance data were generated for each of the five water-to-air heat pumps in a form similar to Fig. 23.

Building Model. Using the construction details of the building, plus assigned infiltration, people and lighting loads, the building was modeled by the BEAP program. Peak load output data were checked by hand calculation.

The floor plan of the building is shown in Fig. 24 and indicates the six zones. The zone and the equipment serving that zone are as follows:

<u>Zone</u>	<u>Unit</u>
1	HP#1 water-to-water
2	HP#2 water-to-air
3	HP#3 water-to-air
4	HP#4 water-to-air
6	HP#6 water-to-air

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FIG. 23 W/W Heat Pump Model - Heating Cycle.

FIG. 24 System Zones for Solar Building.

HP#5 did not operate during the period modeled. It is used as a booster device on Zone 1; thus this area has been included in Zone 1.

System Model. The system is modeled by a modified version of the TRNSYS Program and has the following operating modes:

Direct Heating
Heat Pump Heating
Heat Pump Cooling

The system logic model is shown in Fig. 25. Two major provisions made in developing this model are: 1) the glycol-to-water heat exchanger is included in the solar collector model, and 2) the system can be in only one mode during a single hour.

Because each component's performance is dependent on entering water temperature, the approach to modeling was to use temperatures and mass flow rates rather than energy flow.

Results. The initial modeling of the system and comparison with actual collected data was conducted for a six-day time period between midnight Wednesday February 5, 1975, and midnight Tuesday February 11, 1975. Field collected data (monitored system operation) for this period are shown in Fig. 26.

Results for the first attempt at modeling are presented in Fig. 27. Comparison with Fig. 26 shows that the tank temperature remained high, thus excluding solar energy collection except for the first day. The tank temperature, per Fig. 26, should drop during the night as the building has a nighttime heating requirement. The modeled tank temperature, Fig. 27, indicates a too low nighttime heating requirement. This was confirmed by plotting out the BEAP modeled cooling and heating loads, which showed very little building heating load at night. A check of collected wind velocity data showed very little wind, none in some cases, thus practically no infiltration heat losses in the BEAP model. The wind speed and direction device installed at the building operated erratically or not at all throughout the project, so it was always suspect. Consequently, wind data from Local Climatological Data published in NOAA, and taken about three miles from the building was examined. Average daily wind speeds were found to

FIG. 25 Solar Building System Model.

<u>I. D.</u>	<u>LABEL</u>	<u>SCALE</u>
1	Outdoor Temp	150
2	Collector EWT	150
3	Tank LWT	150
4	S. E. Incident	300,000
5	S. E. Useful	300,000

FIG. 26 Monitored System Operation.

FIG. 27 Modeled System Operation.

range from 3 to 12 miles per hour for the period of analysis. These data were used instead of the collected data and new modeling results generated. Comparative data are shown in Fig. 28 and the modeled tank temperatures track the measured values fairly well (compare Fig. 28 with Fig. 26). It was thus concluded that infiltration was the problem.

Tank temperature is an index of the energy flows in the system because all the thermal energy passes through the tank. On an energy basis, during the six-day heating and cooling period the model accounted for 85% of the building energy requirements.

Overall, it is concluded that, considering the complexity of the system and the fact that the building required heating some times and cooling at other times, correlation for this period was good.

Due to recurrent problems with the data acquisition computer and the Telephone data transmission system (Ref. 15), the other periods examined were necessarily later in the year when the weather was mild. Results were inconclusive. A major deterrent was the erratic operation of the evaporative water cooler. Since more solar energy was collected in mild weather than could be utilized, periodically the cooler would, under control of a thermostat in the water storage tank, reject heat to the environment and reduce the tank temperature. Operation was unpredictable and the tank temperature after a heat rejection cycle varied widely from cycle to cycle. Thus, although the above period of comparison is limited in extent, it involves the most suitable set of continuous data available for the 1974-75 heating season.

Conclusions. Short or long term performance of heat pumps in solar-assisted heat pump systems can be accurately measured in actual buildings being operated in their normal fashion.

Correlation of component models with actual component performance can be obtained within reasonable limits.

Good correlation of field collected data with a computer simulation for the entire building system was obtained for an early February period. Results in mild weather were inconclusive.

<u>I.D.</u>	<u>LABEL</u>	<u>SCALE</u>
1	Outdoor Temp.	150
2	Collector EWT	150
3	Tank LWT	150
4	S.E. Incident	300,000
5	S.E. Useful	300,000

FIG. 28 Final Modeled System Operation (ave. wind).

Since only six zones are representative of a small building and a relatively simple system, this experience indicates that difficulty could be experienced in attempting to correlate field collected data with computer simulations on a zone-by-zone basis for large buildings with a multiplicity of zones.

Some of the possible reasons for difficulty in correlating simulations with field collected data for the solar building are as follows:

Unknown flow of people in and out of the building during a normal working day. This caused variable infiltration rates. Although door openings were measured during the test data collection period, the amount of outdoor air coming in was not determinable. This would depend on the wind speed and direction and whether or not other doors in the building were open at the same time.

Interaction of the several zones. It is not specifically known at any one period of time whether or not a particular office has its lights on, its door open or closed, or is occupied or unoccupied. Thus the actual performance of a particular water-to-air heat pump serving that zone may deviate substantially from the predicted performance based upon the averaged assumptions of lights on, office occupied and door closed.

Thermostat set points and differentials. With the possible interaction between zones discussed above, one zone may supply heating or cooling to another zone so that the unit for the latter is "off" when it would be predicted to be "on". In addition, the thermostats in the six zones will not have the same differentials, and may not have the same set points (in terms of actual temperatures sensed by the thermostats). Thermal capacitance effects can then cause unexplained long off or on periods of operation.

Frequent cycling of the heat pumps. This caused considerable transients in the collected data as well as small temperature differences across heat exchangers until steady state was approached, thus posing a data accuracy problem.

System Design Manual

An important objective of the project is to produce a System Design Manual based on the information and experience obtained from the field evaluation of the Solar Building and the resultant analysis, correlation of data and system modeling. Drafts of this manual were sent some time ago to a specially-appointed ASHRAE Review Team and their comments and suggestions, where feasible, have been incorporated into the current draft. The final draft, which is entitled, Handbook for the Design of Solar Energy Assisted Water-Source Heat Pumps for Non-Residential Buildings is under preparation and is to be sent to the ASHRAE Review Team for final editing and potential publication in a form to be determined by DOE and ASHRAE. The analysis and results for building systems in various climatic zones will be detailed in Ref. 24.

Workshops

During the course of the project, two Workshops were held at The Pennsylvania State University.

The first, a Workshop on Solar Energy Heat Pump Systems for Heating and Cooling Buildings, was held on June 12-14, 1975 and yielded 33 papers and discussions which were published in a Proceedings (Ref. 9) which has been widely disseminated.

The second, a Workshop on the Design of Solar Assisted Heat Pump Systems, was held on July 25-29, 1977. The objective of this conference was to present draft material from the System Design Manual and obtain input from the attendees. No Proceedings were published.

CONCLUSIONS

1. Collection of valid data from systems having multiple HVAC units serving multiple zones is difficult because of the interaction between the HVAC units and the building zones, the frequent cycling of equipment and small temperature differences across heat exchangers.

2. On-site supervision with personnel knowledgeable of HVAC and data acquisition systems is essential and even then considerable difficulty may be experienced.

3. A variety of checks and cross-checks must be utilized to maintain acquisition of valid data over long periods of time; e.g., a winter heating season.

4. A capability for prompt on-site output of data in an easily interpretable form is essential for system trouble shooting and data checking to enhance its validity.

5. When data are to be transmitted over a distance, they should first be put on a permanent medium such as magnetic tape in relatively short files to reduce data losses.

6. Since the solar building is small and the HVAC system relatively simple in comparison with large building systems, this experience indicates great difficulty could be experienced in attempting to evaluate large building system performance by field tests.

7. The Seasonal Performance Factor of the main water-to-water heat pump was good - about 4.2. Considerable auxiliary electric power was used by water pumps and fans; this power must be minimized by careful system design in order to achieve good overall system performance.

8. The internal loads of the solar building, the climate of Albuquerque and the heat output of the data acquisition computer combined to cause an excess, rather than a deficiency, of thermal energy much of the time. Consequently, the contribution of building utilized thermal energy by the solar collector array was small; i.e., the heat pump system with water storage could effectively supply most of the heat requirements of the building.

9. With usual water piping design practice, temperature stratification in all of the three water tanks was essentially non-existent.

10. For optimum system performance, heat pump systems require a relatively low source or sink temperature for heat pump heating or cooling, respectively. However, to maximize the Seasonal Performance Factor, as much direct heating as possible should be utilized. Consequently, a multiple tank or multiple temperature level thermal storage system has considerable potential for increasing overall system performance.

11. Commercial buildings typically have high internal loads and hence may require cooling in many zones throughout much of the year. Consequently, excess thermal energy may be produced, particularly during intermediate seasons. Provisions therefore must be made to reject excess heat to the environment as needed to maintain the storage (sink) temperature at a level into which the heat pump can reject heat when on the cooling cycle.

12. It is important to recognize in the system design process that on the heating cycle the heat equivalent of the heat pump compressor power appears in the condenser and thus is useful heat to the space. This can be credited against the total heat requirements and will have the effect of reducing the amount of solar collector area required.

13. Due to temperature swings in the building associated with thermostat differentials, thermal storage effects caused substantial differences between the measured thermal energy input and the calculated heating load over short periods of time; i.e., up to 12 hours. For daily and longer periods, thermal storage had far less effect. Over many days, computer calculation methods predicted loads within 10% of that measured.

14. Good correlation of field collected data with a computer simulation of the entire building and system was obtained for a cold weather period. Results in mild weather were inconclusive.

15. Commercial buildings with low internal loads and located in colder climates are potentially better suited for solar-assisted heat pump systems than the converse.

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