The Pennsylvania State University
The Graduate School
Department of Architectural Engineering

Performance of a Solar Energy-Assisted Heat Pump Heating System --
Analysis and Correlation of Field-Collected Data

A Thesis in
Architectural Engineering
by
Ronald Craig Williams

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ABSTRACT

An analysis of building energy usage and thermal load for the Solar Building during the winter heating seasons of 1974-75 and 1975-76 is reported here. The one-story office building is located in Albuquerque, New Mexico. Its mechanical heating and cooling equipment is categorized as a solar-assisted heat pump system consisting of solar collectors, water thermal storage, a water-to-water heat pump and five smaller water-to-air heat pump packaged units.

Building energy usage was examined with emphasis on the time of day energy was consumed and the source from which the energy was obtained; i.e., from the electricity for lighting, office equipment and mechanical equipment, and from the heat output of the thermal storage and heat pumps. The rate of electrical energy consumption was found to be very dependent on building use. High rates of electrical energy usage during occupied periods required cooling during parts of even the coldest days. Mechanical equipment heating 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Energies supplied to and withdrawn from the building were examined and are presented for hourly, daily, and seasonal periods. A comparison of the two heating seasons was made.

Energy losses and gains from the building to the surroundings were examined for both steady-state and transient load profiles. Envelope conductive heat losses and losses due to infiltration and ventilation were calculated using actual weather data through the use of the Building Environmental Analysis Program (BEAP). The effect of building thermal storage on heating and cooling loads was examined and a set of building balance-point temperatures was established. Comparisons between the building energy consumption and a calculated load were made for hourly, daily, and seasonal periods. Agreement, in most areas, was found to be acceptable.

TABLE OF CONTENTS

	Page
Abstract	iii
List of Tables	viii
List of Figures	ix
Definition of Terms	xv
Abbreviations and Notations	xvi
Acknowledgements	xvii
 1.0 INTRODUCTION	
1.1 Scope	1
1.2 Need for Study	2
1.3 Objectives	4
1.4 Limitations	5
1.5 Approach	5
 2.0 THE SOLAR BUILDING	
2.1 Building and Site Information	7
2.1.1 Background	7
2.1.2 Building Dimensions	9
2.1.3 Construction	9
2.1.4 Location and Climate	12
2.2 HVAC System	14
2.2.1 General Descriptions	14
2.2.2 Water-to-Water Heat Pump	19
2.2.3 Water-to-Air Heat Pumps	25
2.2.4 Water Thermal Storage	25
2.2.5 Solar Collectors	26
2.2.6 Evaporative Water Cooler	27
 3.0 ENERGY SUPPLIED TO AND WITHDRAWN FROM THE SOLAR BUILDING	
3.1 Introduction	33
3.2 Methods of Energy Supply and Withdrawal	34
3.2.1 Water-to-Water Heat Pump	34
3.2.2 Water-to-Air Heat Pumps	39

	Page
3.2.3 Electrical Energy	43
3.2.4 Building Occupants	48
3.3 Net Energy Delivered as Compared to the Indoor-Outdoor Temperature Differential	48
3.3.1 1975-76 Winter Heating Season	48
3.3.2 Comparison of the Two Winter Heating Seasons	52
3.4 Equipment Heating and Cooling as a Function of Time of Day	64
3.5 Energy Delivered and Withdrawn as a Function of Time of Day	76
3.6 Energy Delivered and Withdrawn as a Function of Day of Year	96
 4.0 BUILDING THERMAL LOAD	
4.1 Thermal Losses and Methods of Calculation	108
4.2 Conduction, Ventilation, and Infiltration Losses	111
4.3 Thermal Storage	115
4.4 Steady-State Load Calculations	116
4.5 Balance-Point Temperatures	119
4.6 Transient Load Calculations	125
4.6.1 Hourly Comparisons of Net Energy Consumption and Calculated Thermal Load	125
4.6.2 Hourly Comparisons of Mechanical Equipment Heating and the Calculated Heating Load	146
4.6.3 Daily Comparisons of Net Energy Consumption and Calculated Heating Load	149
 5.0 CONCLUSIONS AND RECOMMENDATIONS	
5.1 Conclusions	156
5.2 Recommendations	163

APPENDIX A: BUILDING CONSTRUCTION INFORMATION.	166
APPENDIX B: INSTRUMENTATION, DATA ACQUISITION, AND DATA VALIDATION	172
B-1 Instrumentation	172
B-2 Data Acquisition	175
B-3 Data Validation	176
APPENDIX C: DATA REDUCTION AND EVALUATION PROGRAMS	189
C-1 Data Reduction Computer Programs.	189
C-2 Operational Mode Determination.	189
C-3 Heat Pump Heating and Cooling Computer Program.	195
C-4 Direct Heating Computer Program	196
C-5 Water-to-Water Heat Pump Coefficient of Performance Computer Program	196
C-6 Total Building Energy Computer Program. . .	197
C-7 Building Performance Factor Computer Program.	198
LITERATURE CITED.	199
BIBLIOGRAPHY.	201

LIST OF TABLES

	Page
1. Solar Building Room Dimensions.	11
2. Summary of Temperatures and Energy Supplied and Withdrawn for the 1974-75 Winter Heating Season	62
3. Summary of Temperatures and Energy Supplied and Withdrawn for the 1975-76 Winter Heating Season	63
4. Steady-State Load Equations	120
5. Balance-Point Temperatures.	123
6. Summary of Monthly and Seasonal Net Energy Usage and Calculated Loads.	160
7. Solar Building Exterior Wall Construction	167
8. Solar Building Wall and Roof Assemblies	168
9. Solar Building Room Construction.	170
10. Mechanical Equipment Flow Rates, Average Temperature Differentials, and Steady-State Error for the 1974-75 Winter Heating Season	187
11. Mechanical Equipment Flow Rates, Average Temperature Differentials, and Steady-State Error for the 1975-76 Winter Heating Season	188

LIST OF FIGURES

	Page
1. The Solar Building	8
2. Floor Plan of the Solar Building	10
3. Schematic Piping Diagram, Winter 1974-75	16
4. Schematic Piping Diagram, Winter 1975-76	18
5. Schematic Heat Pump Heating Circuit.	20
6. Schematic Heat Pump Cooling Circuit.	21
7. Schematic Direct Heating Circuit, Winter 1974-75	22
8. Schematic Direct Heating Circuit, Winter 1975-76	23
9. Schematic Solar Collection Circuit, Winter 1974-75 . . .	28
10. Schematic Solar Collection Circuit, Winter 1975-76 . . .	29
11. Schematic Evaporative Water Cooler, Heat Rejection Circuit, Winter 1974-75.	31
12. Schematic Evaporative Water Cooler, Heat Rejection Circuit, Winter 1975-76.	32
13. Rates of Heating Energy Supplied by the Water-to-Water Heat Pump as a Function of the Indoor-Outdoor Temperature Differential for Day Periods	37
14. Rates of Heating Energy Supplied by the Water-to-Water Heat Pump as a Function of the Indoor-Outdoor Temperature Differential for Night Periods	38
15. Rates of Energy Withdrawn by Water-to-Air Heat Pump Cooling as a Function of the Indoor-Outdoor Temperature Differential for Day Periods	41
16. Rates of Energy Withdrawn by Water-to-Air Heat Pump Cooling as a Function of the Indoor-Outdoor Temperature Differential for Night Periods	42
17. Rates of Electrical Energy Use as a Function of the Indoor-Outdoor Temperature Differential for Day Periods .	46
18. Rates of Electrical Energy Use as a Function of the Indoor-Outdoor Temperature Differential for Night Periods	47

	Page
19. Rates of Net Energy Supplied as a Function of the Indoor-Outdoor Temperature Differential for Day Periods.	49
20. Rates of Net Energy Supplied as a Function of the Indoor-Outdoor Temperature Differential for Night Periods	51
21. Rates of Adjusted Net Energy Supplied as Compared to Steady-State Heating Loads as a Function of the Indoor-Outdoor Temperature Differential for Day Periods, Winter 1974-75	53
22. Rates of Adjusted Net Energy Supplied as Compared to Steady-State Heating Loads as a Function of the Indoor-Outdoor Temperature Differential for Night Periods, Winter 1974-75	54
23. Rates of Adjusted Net Energy Supplied as Compared to Steady-State Heating Loads as a Function of the Indoor-Outdoor Temperature Differential for Daily Periods, Winter 1974-75.	55
24. Rates of Adjusted Net Energy Supplied as Compared to Steady-State Heating Loads as a Function of the Indoor-Outdoor Temperature Differential for Day Periods, Winter 1975-76.	56
25. Rates of Adjusted Net Energy Supplied as Compared to Steady-State Heating Loads as a Function of the Indoor-Outdoor Temperature Differential for Night Periods, Winter 1975-76.	57
26. Rates of Adjusted Net Energy Supplied as Compared to Steady-State Heating Loads as a Function of the Indoor-Outdoor Temperature Differential for Daily Periods, Winter 1975-76.	58
27. Comparison of Heating and Cooling with Temperature Differentials and Time for the Week of December 15, 1975	66
28. Comparison of Heating and Cooling with Temperature Differentials and Time for the Week of December 22, 1975	69
29. Comparison of Heating and Cooling with Temperature Differentials and Time for the Week of December 29, 1975	71
30. Comparison of Heating and Cooling with Temperature Differentials and Time for the Week of January 5, 1976.	73

	Page
31. Comparison of Heating and Cooling with Temperature Differentials and Time for the Week of January 12, 1976.	74
32. Comparison of Heating and Cooling with Temperature Differentials and Time for the Week of January 19, 1976.	75
33. Comparison of Heating and Cooling with Temperature Differentials and Time for the Week of January 26, 1976.	77
34. Comparison of Heating and Cooling with Temperature Differentials and Time for the Week of February 2, 1976.	78
35. Comparison of Heating and Cooling with Temperature Differentials and Time for the Week of February 9, 1976.	79
36. Comparison of Heating and Cooling with Temperature Differentials and Time for the Week of February 16, 1976	80
37. Comparison of Heating and Cooling with Temperature Differentials and Time for the Week of February 23, 1976	81
38. Comparison of Electrical Energy Usage to Heating Energy Supplied and Cooling Energy Withdrawn, Week of December 15, 1975	84
39. Comparison of Electrical Energy Usage to Heating Energy Supplied and Cooling Energy Withdrawn, Week of December 22, 1975	85
40. Comparison of Electrical Energy Usage to Heating Energy Supplied and Cooling Energy Withdrawn, Week of December 29, 1975	87
41. Comparison of Electrical Energy Usage to Heating Energy Supplied and Cooling Energy Withdrawn, Week of January 5, 1976	88
42. Comparison of Electrical Energy Usage to Heating Energy Supplied and Cooling Energy Withdrawn, Week of January 12, 1976.	89
43. Comparison of Electrical Energy Usage to Heating Energy Supplied and Cooling Energy Withdrawn, Week of January 19, 1976.	90
44. Comparison of Electrical Energy Usage to Heating Energy Supplied and Cooling Energy Withdrawn, Week of January 26, 1976.	91

45. Comparison of Electrical Energy Usage to Heating Energy Supplied and Cooling Energy Withdrawn, Week of February 2, 1976.	92
46. Comparison of Electrical Energy Usage to Heating Energy Supplied and Cooling Energy Withdrawn, Week of February 9, 1976.	93
47. Comparison of Electrical Energy Usage to Heating Energy Supplied and Cooling Energy Withdrawn, Week of February 16, 1976	94
48. Comparison of Electrical Energy Usage to Heating Energy Supplied and Cooling Energy Withdrawn, Week of February 23, 1976	95
49. Daily Energy Usage for January 1975.	99
50. Daily Energy Usage for February 1975	100
51. Daily Energy Usage for March 1975.	102
52. Daily Energy Usage for April 1975.	103
53. Daily Energy Usage for December 1975	104
54. Daily Energy Usage for January 1976.	105
55. Daily Energy Usage for February 1976	107
56. Steady-State Thermal Load as a Function of the Outdoor Temperature.	121
57. Steady-State Thermal Load with Corresponding Balance-Point Temperatures for Day Periods	124
58. Steady-State Thermal Load with Corresponding Balance-Point Temperatures for Night Periods	126
59. Steady-State Thermal Load with Corresponding Balance-Point Temperatures for Daily Periods	127
60. Comparison of Net Energy Usage with the Calculated Heating Load, Week of December 15, 1975.	132
61. Comparison of Net Energy Usage with the Calculated Heating Load, Week of December 22, 1975.	134
62. Comparison of Net Energy Usage with the Calculated Heating Load, Week of December 29, 1975.	136

63.	Comparison of Net Energy Usage with the Calculated Heating Load, Week of January 5, 1976.	137
64.	Comparison of Net Energy Usage with the Calculated Heating Load, Week of January 12, 1976	138
65.	Comparison of Net Energy Usage with the Calculated Heating Load, Week of January 19, 1976	139
66.	Comparison of Net Energy Usage with the Calculated Heating Load, Week of January 26, 1976	140
67.	Comparison of Net Energy Usage with the Calculated Heating Load, Week of February 2, 1976	142
68.	Comparison of Net Energy Usage with the Calculated Heating Load, Week of February 9, 1976	143
69.	Comparison of Net Energy Usage with the Calculated Heating Load, Week of February 16, 1976.	144
70.	Comparison of Net Energy Usage with the Calculated Heating Load, Week of February 23, 1976.	145
71.	Comparison of Heat Pump and Direct Heating with the Calculated Heating Load, Week of December 22, 1975 . . .	147
72.	Comparison of Net Energy Usage with Calculated Heating Load with Corresponding Indoor and Outdoor Temperatures, December 1975.	150
73.	Comparison of Net Energy Usage with Calculated Heating Load with Corresponding Indoor and Outdoor Temperatures, January 1976	151
74.	Comparison of Net Energy Usage with Calculated Heating Load with Corresponding Indoor and Outdoor Temperatures, February 1976.	152
75.	Water-to-Water Heat Pump Heating Condenser vs. Air-Handling Unit Energy Balance	180
76.	Water-to-Water Heat Pump Heating Chiller vs. Condenser Energy Balance	181
77.	Water-to-Water Heat Pump Cooling Chiller vs. Air-Handling Unit Energy Balance	183
78.	Water-to-Water Heat Pump Cooling Condenser vs. Chiller Energy Balance	184

	Page
79. Water-to-Air Heat Pump Loop Energy Balance	186
80. Water-to-Water Heat Pump Mode Selection, Method 1, 1974-75 and 1975-76 Winter Heating Seasons	191
81. Water-to-Water Heat Pump Mode Selection, Method 2, 1974-75 Winter Heating Season.	192
82. Water-to-Water Heat Pump Mode Selection, Method 2, 1975-76 Winter Heating Season.	193

DEFINITION OF TERMS

Adjusted Net Energy Supplied	Net Energy Supplied minus Ventilation and Infiltration
Daily Period	Twenty-four hour time interval, beginning at 6:00 a.m.
Day Period	Twelve-hour time interval, beginning at 6:00 a.m.
Direct Heating	Heating accomplished without heat pump thermal boosting
Heat Pump Cooling	Water-to-water heat pump operating in the cooling mode
Heat Pump Heating	Water-to-water heat pump operating in the heating mode
Net Energy Supplied	Total Energy Supplied minus Total Energy Withdrawn
Night Period	Twelve-hour time interval, beginning at 6:00 p.m.
Total Energy Supplied	Sum of mechanical equipment heating, electrical energy use, and sensible heating by building occupants
Total Energy Withdrawn	Sum of water-to-water and water-to-air heat pump cooling
Water-to-Air Heat Pump loop (also, water-to-air loop)	Five small tonnage water-to-air heat pumps with a combined capacity of 7 tons connected in parallel on a continuously flowing water loop
Water-to-Water Heat Pump	A single 7 1/2 ton water chiller capable of operating in both a heating and cooling mode by alternately connecting the evaporator and condenser to the thermal storage and air-handling unit through appropriate piping by connecting the water thermal storage directly to the water-to-water heat pump air handler.

ABBREVIATIONS AND NOTATIONS

AHPC	water-to-air heat pump cooling
CLG	water-to-air heat pump cooling mode sensor, Figure 3
DH	direct heating
DLG	direct heating mode sensor, Figure 3
ELEC	electrical energy supplied
FM	water flow meter, Figure 4
ft	feet
hr	hour
HTG	water-to-water heat pump heating mode sensor, Figure 3
INFIL	infiltration to building
kW	kilowatt
m	meter
MBtu	10^3 Btu (British thermal unit)
PEOP	sensible heat from building occupants
PM	power meter, Figure 4
Q	quantity of energy supplied or withdrawn
TW	water temperature sensor, Figure 4
TR	air temperature sensor, Figure 4
VENT	forced ventilation to building
WHPC	water-to-water heat pump cooling
WHPH	water-to-water heat pump heating

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1.0 INTRODUCTION

1.1 Scope

This thesis reports on a portion of a research project undertaken by the Department of Architectural Engineering of The Pennsylvania State University and entitled "Evaluation of the Solar Building, Albuquerque, New Mexico". The project's purpose was to conduct a field evaluation of an existing solar-assisted heat pump system in a commercial office building. Specifically, objectives of the project were to instrument and collect actual building data from an existing solar-assisted heat pump installation, to analyze the data collected, and then, based on actual performance, to develop a simulation model to make theoretical predictions of the performance of other commercial buildings in other geographical locations. The final phase of the project was to develop and publish a System Design Manual for Solar-Assisted Systems in Commercial Buildings.

This author's responsibility was to analyze the building's thermal performance in regard to energy use and building thermal load. It is the result of this research that is reported here.

The building was examined as an envelope subjected to various weather conditions and use. The energy delivered to or removed from the building was calculated for the two winter heating seasons of 1974-75 and 1975-76. Assumptions on infiltration and ventilation rates for occupied and unoccupied periods were based on measured flows. Actual

energy supplied to the building was then compared to calculated energy requirements. Agreement was made where possible and probable reasons for disagreement are presented. Calculations for operation and performance were made at hourly intervals throughout the two heating seasons.

Information presented in these areas will provide documentation beyond what is presently available. In particular, information on building energy usage documents which part of a heating load may be supplied in a typical commercial building: by internal loads directly, by previously stored energy removed from the building during cooling operation, by the solar collectors, and by an auxiliary back-up system. A computer building load program, Building Environmental Analysis Program (BEAP) (1) was used to calculate an hourly building load. This hourly building load was compared to actual energy usage.

1.2 Need for Study

The present principal source of design information for the consulting engineer is the American Society of Heating, Refrigeration and Air Conditioning Engineers (ASHRAE). Research committees collect relevant research information, and apply it to design and application engineering information which is then published in a series of ASHRAE handbooks, a monthly journal, or a yearly publication of transactions. Presently, there is limited design information specific to calculating energy requirements for buildings with energy-conserving designs such as solar or energy reclamation systems. Much of the information concerning solar systems available elsewhere deals with residential buildings or schools.

Where information is available for commercial installations, it is often sketchy or deals with only one or several components of the energy-conserving system.

In designing heating and cooling systems for buildings with conventional mechanical equipment, it is acceptable to calculate only the peak heating and cooling loads at design conditions. Most of these load calculation methods use steady-state temperature and weather profiles and do not consider the relative amounts or duration of these thermal loads. When designing energy recovery and/or solar-assisted systems, more information is needed to assist the engineer in making accurate predictions of the performance.

The optimization of this type of system requires not only calculation of maximum heating and cooling loads but also a knowledge of how often the loads occur and their durations. Information about internal heat generation is equally important since it is this energy which may be used to offset energy from a solar or auxiliary back-up system. In addition, selection of equipment becomes more difficult due to the wide range of operating conditions inherent in such systems.

To effect widespread utilization of energy recovery and solar-assisted systems for the heating and cooling of buildings, consulting engineers and other designers need assurance that the proposed system will meet or exceed performance specifications and will be installed and operated at reasonable cost. To aid the consulting engineer in making more accurate predictions of initial

and operating costs, this study analyzes building energy flows to and from the building, and then, correlates these energy flows to building thermal loads. Information presented in this study provides operating performance of an actual building in a field test.

1.3 Objectives

The overall objective of this study was to provide building system field information for the design of energy-conserving systems in commercial buildings. It was intended that this approach would provide an improved understanding of the relationship between internal loads, mechanical equipment operation, and building energy requirements. It is hoped that this improved understanding will lead to mechanical equipment systems which are designed to minimize energy usage, while at the same time provide desired comfort conditions at the design loads.

To achieve this, this research had two major goals. The first was to examine methods of energy supply to the building, to gain an understanding of the entire building as a complete system, and to analyze the effect of electrical power usage on mechanical equipment. The second goal was to correlate building energy usage to a calculated building load. Relationships between the actual energy supplied and building heat losses/gains were assessed. A set of balance-point temperatures for occupied and unoccupied periods was established, and the Building Environmental Analysis Program was validated as an acceptable procedure in the calculation of building energy transfers.

1.4 Limitations

This research was confined to building energy usage and thermal performance of the Solar Building. No attempt was made to analyze the effectiveness of individual components of the mechanical equipment system. The building itself was viewed as an envelope being subjected to weather conditions and use. No attempt was made to divide the building into individual zones served by individual pieces of mechanical equipment, nor was there any attempt to compare energy use, building load, or any other parameter to any other building.

1.5 Approach

The second chapter provides a general description of the building and the HVAC system.

Evaluations of energy supplied to or withdrawn from the Solar Building appear in Chapter 3. This chapter includes methods of energy supply and energy withdrawal, net energy delivered as compared to the indoor-outdoor temperature differential, equipment heating and cooling as a function of time of day, energy supplied and withdrawn as a function of time of day, and a comparison of energy use for the two winter heating seasons.

Chapter 4 discusses losses and gains from the building to the surroundings. Analysis presented in this chapter consisted of both

steady-state and transient loads. The transient loads were calculated hour-by-hour using actual weather data recorded at the building site. Conductive losses through the building's walls, roofs, and windows were examined as were losses associated with infiltration and ventilation. A set of balance-point temperatures was established and is presented for both occupied and unoccupied periods. The effect of thermal storage on mechanical equipment load is discussed. And a detailed comparison of net energy consumption with the calculated thermal load is presented both as a function of time of day and day of year.

The final chapter, Chapter 5, discusses the conclusions and recommendations for further study.

2.0 THE SOLAR BUILDING

2.1 Building and Site Information

2.1.1 Background

The Solar Building, a one-story commercial office building, depicted by Figure 1, is believed to be the first solar heated office building in the world. The building and its systems were designed by the owner and occupant, the firm of Bridgers and Paxton, Consulting Engineers (2). The building was first occupied in August 1956, and the solar system operated successfully for several years. The solar system was eventually shut down due to water leakage problems which developed as a result of degradation of the aluminum flat plate solar panels. Since fossil fuels were comparably inexpensive at that time the solar system was discontinued.

During the period extending from April through September 1974, the solar collectors were refurbished in preparation for the research project of which this thesis is a part. In addition, an elaborate instrumentation and data acquisition system was designed and installed. Actual on-site data monitoring and data collection began in November of that year. Winter heating information was collected for the 1974-75 and 1975-76 seasons. Data collection also occurred during the summer of 1975; that summer cooling information is not reviewed in this thesis.

Figure 1. The Solar Building

2.1.2 Building Dimensions

The building, consisting of 4300 sq ft (400 m^2) of usable floor area resembles an "L" in shape. Including a small attic space which is used as a return air space and houses some mechanical equipment (all water-to-air heat pumps and supply ducts for the water-to-water heat pump), the building's total volume is $40,120 \text{ ft}^3$ (1136 m^3). A drafting area located at the west side of the building comprises approximately 44% of the floor area. The rest of the building is divided into smaller spaces. These include individual offices, a reception area, a conference room, a computer room, and two rest rooms. The floor plan with an equipment legend appears in Figure 2. The plan shows the overall length of the building to be approximately 140 ft (42.7 m), orientated in an east-west direction. The width varies from 21 ft to 48 ft (6.4 m to 14.6 m). The dimensions, areas, and volumes of the individual spaces appear in Table 1.

2.1.3 Construction

The building envelope is of modern construction. The roof assembly consists of an insulated built-up roof supported by structural steel beams which run to ground on 18 ft (5.5 m) centers. The east wall consists of two layers of brick. Other exterior walls contain 2 in. by 4 in. (5 cm by 10 cm) wood framing with batt-type insulation. Fifty-five panels comprising the solar collector are located on the south wall. They are made of aluminum plates covered with single glass and housed in steel frames. This south wall is inclined at an

Figure 2. Floor Plan of the Solar Building

Table 1. Solar Building Room Dimensions

Space	Length ft	Width ft	Height ft	Floor Area ft ²	Volume ft ³
Office #1	11.10	11.50	7.75	128	989
Office #2	11.10	11.50	7.75	128	989
Office #3	10.00	11.60	10.75	116	1,247
Office #4	10.00	11.40	7.75	114	883.5
Office #5	10.00	11.75	7.75	118	910
Office #6	9.75	11.75	7.75	115	888
Office #7	13.25	8.10	7.75	107	882
Office #8	13.25	17.80	7.75	206	1,828
Drafting Room Ext.	19.75	30.25	7.75	598	4,630
Drafting Room	21.50	37.00	12.75	878	10,143
Computer Room	21.70	18.20	12.75	395	5,035
Conference Room	12.70	11.40	7.75	145	1,122
Reception Room	13.20	29.75	7.75	453	3,043
Print Room	13.25	8.10	7.75	107	832
Corridors (2)	5.50	24.00	7.75	178	1,023
Rest Room and Hall	12.00	11.40	7.75	137	1,058
Mechanical Room	42.75	6.00	4.8 (avg)	256	1,218
Total Wall Area				600	
Return Air Space (Attic)					2,075

Net Usable Floor Area 4,300 ft²

Total Building Area 4,900 ft²

Total Building Volume 39,800 ft³

angle of 55° (0.96 radians) from the horizontal. The west end of the south wall, the west wall, and the north wall are all finished with 1 in. (2.5 cm) of stucco backed with fiberboard sheathing. All interior surfaces are gypsum board. A portion of the south wall is separated from a conventionally heated and cooled addition by fixed partitions and doors equipped with closers.

The windows form a total area of 355 ft^2 (32.9 m^2). The largest number of windows, comprising 264 ft^2 (24.5 m^2) of the total, are located on the north facing wall. Six skylights are located in the drafting room and the reception area. These form a combined area of 50 sq ft (4.6 m^2). The remaining window area is found on the east facing wall. Detailed wall, window, and roof construction information is contained in Appendix A.

2.1.4 Location and Climate

The Solar Building is located $35^\circ 03'$ north latitude and $106^\circ 37'$ west longitude at an elevation of 5300 ft (1615 m) above sea level. It is positioned on flat land within the city of Albuquerque, New Mexico.

The only solar obstruction to the building is an adjoining one-story conventionally heated and cooled office building to the south. This building shades some of the lower collectors during morning hours.

The climate is described as arid continental. The Trewartha designation of climate is BSK (dry, semi-arid, middle latitude and meager rainfall) (3).

The average monthly precipitation varies from less than 1/2 in. (1.3 cm) per month during the winter to over 1 1/4 in. (3.2 cm) during July and August. Annual mean rainfall and snowfall is 8.33 in. (21.2 cm) (based on 82 years ending 1975) (4).

Temperatures in Albuquerque are typical of dry, high altitude continental climates with large daily ranges but rare extremes. Normally there is less than one day a year when temperatures fall below 0°F (-18°C) or reach 100°F (38°C). Daytime temperatures during winter months average near 50°F (10°C) with only a few days when temperatures do not reach above freezing (4).

Albuquerque experiences a large number of clear days with a high percentage of sunshine throughout the year. Clear sunny weather predominates throughout the winter months. For the year 1975, there was 78% of possible sunshine with 184 clear days, 98 partly cloudy days, and 83 cloudy days (4). These numbers agree well with yearly means.

Throughout the year, wind averages approximately nine miles per hour. In late winter and spring, the wind speed is somewhat higher. The prevailing wind direction is southeast and for 46 days a year on an average the wind speed reaches 32 miles per hour (51 km) (4).

2.2 HVAC System

2.2.1 General Descriptions

The HVAC system is a water type with the following major components:

1. A water-to-water heat pump
2. Five small tonnage water-to-air heat pumps
3. Three water thermal storage tanks
4. A solar collector assemblage
5. An evaporative water cooler

The water-to-water heat pump consists of a 7 1/2 ton water chiller with a reciprocating compressor and is located in the mechanical room. Since a single zone thermostat cannot adequately control the temperature throughout the building, the five water-to-air heat pumps are used to control the temperature of selected spaces. Their combined cooling capacity is 6 1/2 tons and they are located in the attic space above the spaces they serve. The thermal storage tank consisted of one 6000 gallon (22.7 m^3) underground tank during the winter of 1974-75. For the second heating season, 1975-76, two aboveground tanks were added with nominal capacities of 2000 and 3000 gallons (7.6 m^3 and 11.4 m^3). During the first season, all thermal energy was stored and removed from the one tank. During the second year, the original underground tank was used for heat pump heating and cooling only. Solar collected energy was stored in the additional aboveground tanks; this energy was used for direct heating. The solar array, consisting of 55 panels, comprises 750 sq ft (69.7 m^2) of collectors on the building's

south wall. The evaporative water cooler serves as the heat rejector and cycles on when the underground storage tank temperature rises above a specified level.

Figures 3 through 12 portray schematically the various system operational modes. The first two figures show all of the piping configurations and heating and cooling equipment for the two heating seasons. The remaining eight diagrams show operational modes individually. Figure 3 portrays the heating and cooling system for the winter of 1974-75. In this figure, supply lines are shown by solid lines, and return lines are shown by broken lines. The location of all pumps and valves operating the system are shown. Both the solar collectors and evaporative water cooler are joined to the heat exchanger and contain an ethylene glycol-water mixture to prevent freeze-up. Individual pumps to both systems allow the two systems to function individually. During the first year, water flowed continuously in a loop through the shell of the shell-and-tube heat exchanger connecting the storage tank and the heating and cooling equipment.

After the water left the underground thermal storage, it was split into two flow streams; one served the water-to-air heat pump loop and the other served the water-to-water heat pump loop. Each of the two flow streams had individual continuously operating pumps. In the water-to-air loop the pump operation allowed the individual heat pump units to cycle on and off as needed. During periods of direct heating the water-to-water heat pump loop pump supplied water from the water thermal storage to the air handling unit directly. When operating in

Figure 3; Schematic Piping Diagram, Winter 1974-75

the water-to-water heat pump heating and cooling modes, an additional pump was required. This additional pump connected the condenser during heating and the chiller during cooling to the air handling unit. The other pump supplied water from the storage tank to the chiller during heating and to the condenser during cooling. Valves are identified in the figures by "HTG", "CLG", and "DLG", which represent heat pump heating, heat pump cooling, and direct heating, respectively. The valves were open for the mode specified and, otherwise, were closed.

The two additional tanks are shown on the piping schematic for the winter of 1975-76 (Figure 4). In addition to the tanks, new piping configurations are shown. Additional piping connected the solar collectors to the two aboveground tanks; valves which connected the collectors to the underground tank were closed. For this second collection period, the pumps for the collector-to-heat exchanger loop and the heat exchanger-to-tank loop operated only when solar energy was collected. Since energy for direct heating was now supplied from the aboveground tanks, additional piping connected these tanks to the air handling unit.

The location of instrumentation is also shown, in Figure 4, for this second year. "PM" signifies a power meter located on pumps and compressors; "TW" signifies a thermal well for recording water temperatures; "FM" signifies a flow meter to record water flow rates; "TR" signifies a temperature-recording device in the airstream.

Figure 4. Schematic Piping Diagram, Winter 1975-76

2.2.2 Water-to-Water Heat Pump

The water-to-water chiller functioned as a heat pump by alternately connecting the condenser and the evaporator to the air handling unit in a closed water loop. In the heating mode the condenser supplied heat to the air handling unit by using the underground storage tank as an energy source (Figure 5). During periods requiring cooling, the machine operated as a chiller connecting the condenser to the thermal storage and the evaporator to the air handling unit (Figure 6). A provision was made in the piping and control scheme to allow the heat pump to be bypassed for direct heating. In this mode the thermal storage was connected directly to the air handling unit and operated when the storage was at significantly high temperatures to satisfy the heating requirements without thermal boosting. The direct heating mode is shown in Figure 7 for the 1974-75 heating season and in Figure 8 for the 1975-76 heating season.

To prevent the heating and cooling modes from opposing each other, a dead band between direct heating and heat pump cooling was provided in the pneumatic thermostat. This allowed the building temperatures to drift through a set temperature differential before switching into an opposing mode. The piping configuration during dead band selection was that of heat pump cooling without compressor operation. The reasons for this arrangement were threefold. First, if the system was going from heating to cooling, the air handling unit and piping may have had water temperatures in excess of 100°F (38°C). The dead band allowed dissipation of this heat preventing compressor start-up.

Figure 5. Schematic Heat Pump Heating Circuit

Figure 6. Schematic Heat Pump Cooling Circuit

Figure 7. Schematic Direct Heating Circuit, Winter 1974-75

Figure 8. Schematic Direct Heating Circuit, Winter 1975-76

with high chiller temperatures. (These high chiller temperatures would cause the compressor to shut down.) Next, as the building cooling requirements were satisfied, the mode switching from heat pump cooling to the dead band allowed the cold water in the chiller to be dissipated. Then, after the transients were reduced, this mode allowed circulation of water completely isolated from any thermal source or sink.

The water-to-water heat pump was located in the mechanical equipment room and was connected to a pneumatic four-positioned thermostat located in the southwest corner of the drafting room. The control modes of the heat pump, starting at 15 lb. pressure (6.8 kg) and going towards 0 lb. pressure (0 kg) were heat pump cooling, dead band, direct heating, and heat pump heating. In addition, protective circuits were built into the control for the heat pump compressor which shut down the motor if condenser temperatures rose too high or fell too low.

The ductwork supplied all conditioning to the drafting room, reception area and hall; it supplied all other areas with ventilation and some conditioning. Extending nearly the entire length of the building, the attic space served as a return air plenum. The mechanical equipment room acted as a mixing chamber for the return air and the ventilation air. Ventilation air was supplied at a rate of 1700 cfm for the first season and 1000 cfm for the second. An axial fan supplied ventilation air during the first heating season and was equipped with a manual switch operated by the building occupants. For the second heating season a squirrel-cage centrifugal fan supplied the ventilation. This fan was equipped with an automatic clock which operated the fan only

during occupied hours (7:30 a.m. to 5:30 p.m. weekdays).

2.2.3 Water-to-Air Heat Pumps

Five water-to-air heat pumps were used to maintain comfort levels in various offices, a computer room, and other smaller spaces. One of these units is shown for each year in the system schematics. The units were connected in parallel to a pipe loop through which water flowed continuously. The units drew their air primarily from the spaces they served, but also drew additional air for ventilation from the main duct system. As mentioned above, the main duct assured a continuous supply of fresh air. Unlike the main water-to-water heat pump, the water flow direction did not change. Changeover from heating to cooling was provided by an automatic changeover thermostat connected to a reversing valve in the refrigerant circuit. Due to the heat generation of the computer facilities and other internal spaces, these units operated predominately in the cooling mode. This water loop may be thought of as an "energy source" which removed excess energy continuously from the building and supplied thermal energy to the water storage for later use.

2.2.4 Water Thermal Storage

The original 6000 gallon (22.71 m^3) underground storage tank (Tank 1) served as both the heat source and sink for all mechanical equipment during the 1974-75 heating season. During the heating season, Tank 1 would rise to temperatures in excess of 100°F (38°C). At these tempera-

tures direct heating operated well, but the elevated temperatures caused short cycling for both heat pump heating and cooling. Before the start of the 1975-76 heating season, two aboveground tanks (Tank 2 and Tank 3) were added. The capacities of these two tanks were 2131 gallons (8.067 m^3) and 2861 gallons (10.33 m^3). They were added for the purpose of solar energy collection and subsequent direct heating. These additional tanks allowed the solar tank temperatures to rise freely while the temperatures in Tank 1 remained at suitable temperatures for heat pump operation.

2.2.5 Solar Collectors

The 750 sq ft (69.7 m^2) configuration of solar panels provided the building with its solar collection facility. Due to use of city water during the 1956-57 heating season, the integral channel aluminum Roll Bond developed numerous corrosion leaks and became inoperative. As a result, the plates had been refitted with 3/4 in. (19 mm) diameter copper tubes, affixed in a serpentine fashion by means of mechanical clips. An aluminum-filled epoxy was used to create a thermo-bond along the entire length of tube. As a first stage of this research project, the surfaces which had deteriorated were cleaned and resprayed with a flat black paint and the bonding of copper tubes renewed where necessary.

The coolant was a 30% mixture (by volume) of ethylene glycol and treated water. The panels were connected in parallel and were joined by common headers. The circulation rate was $0.04 \text{ gallons/min (ft}^2\text{)}$ ($1.7 \text{ liters/min[m}^2\text{]}$). Energy exchange between the collector loop fluid

and circulating water was accomplished through a shell-and-tube heat exchanger. During the 1974-75 heating season the exchanger was located on the water-to-water heat pump return line. A separate loop provided the transfer to the aboveground storage for the 1975-76 season. The piping arrangements for the two years are shown in Figures 9 and 10.

During the first year of the study, solar energy collection would commence when the sensor, located on the back of one of the panels, exceeded the temperature sensor at the heat exchanger by 10°F (5.6°C). During that year, collection ceased when this temperature differential fell to 2°F (1.1°C). The remote temperature sensor was moved to the aboveground tank for the second year. The temperature differential remained the same; the sensor, however, now controlled the water loop pump between aboveground tanks and heat exchanger in addition to the collector-heat exchanger loop pump.

2.2.6 Evaporative Water Cooler

The evaporative water cooler served as the heat rejector. While for direct heating higher water temperatures are more beneficial, it was necessary to maintain a lower range for the heat pumps. Generally, the cooler switched on when the underground tank temperature rose above 100°F (38°C) for the first year, although maximum temperatures were varied, for certain tests. For the second year the aboveground direct heating tanks were allowed to rise without limit. The evaporative water cooler again cycled on to cool the underground tank when temperatures rose above the 100°F (38°C) maximum setting, resulting from

Figure 9. Schematic Solar Collection Circuit, Winter 1974-75

Figure 10.. Schematic Solar Collection Circuit, Winter 1975-76

excess energy being supplied to the water storage. The last two Figures, 11 and 12, show the evaporative water cooler circuit for the winters of 1974-75 and 1975-76, respectively.

Figure 11. Schematic Evaporative Water Cooler, Heat Rejection Circuit, Winter 1974-75

Figure 12. Schematic Evaporative Water Cooler, Heat Rejection Circuit, Winter 1975-76

3.0 ENERGY SUPPLIED TO AND WITHDRAWN FROM THE SOLAR BUILDING

3.1 Introduction

This chapter describes actual energy consumption of The Solar Building for the winter heating seasons of 1974-75 and 1975-76. Data collection methods were improved for the second heating season, and information presented mostly reflects that year. Comparisons are made between the two years and show consistency in the data. Although data collection for the first heating season began near the end of November, several reoccurring problems prevented the analysis of reliable data until near the end of December. Data collection the second year began December 15, 1975, and then continued through April. The most important period for winter heating was from December to the end of February. Several areas of analysis did continue through the end of March, allowing an extended period of study for some parameters. During March, the weather became mild, which resulted in the cooling operation becoming dominant.

The decision not to show weekday and weekend periods separately was based on examination of data showing downward indoor temperature swings on weekends. This reduction in building temperature produced a significant release of thermally stored energy from within the building. Since the field test was not designed to estimate thermal storage in a precise manner, information as to the actual amount of energy released could not be discerned. This problem was further complicated by building occupants manually adjusting the thermostat

downward on weekends in an uncontrolled and variable manner. Separation into weekday and weekend periods would have provided misleading information by overpredicting weekday energy and underpredicting weekend energy.

3.2 Methods of Energy Supply and Withdrawal

3.2.1 Water-to-Water Heat Pump

Heating energy to overcome the building thermal losses was supplied by the water-to-water heat pump, electrical energy, and, to a smaller extent, heat generated by the occupants. Although five water-to-air heat pumps were able to supply heating as well as cooling, only on rare occasion did these units operate in the heating mode. Therefore, the water-to-water heat pump was the primary piece of mechanical equipment which supplied heating to the building. This energy was supplied by using water from the thermal storage as the energy source for water-to-water heat pump operation; the heat pump then supplied energy to the building at the air-handling unit. A system of ductwork distributed this heated air to individual spaces. This unit was capable of operating in two heating modes, a direct heating mode and a heat pump heating mode. The direct heating mode operated first as the building temperature dropped, supplying energy to the building by connecting the thermal storage directly to the air-handling unit coil. If the building temperature continued to fall because of the thermal storage not having a sufficiently high temperature to provide the needed energy release at the air-handling unit, the unit would switch to the heat pump

heating mode. In this mode, the thermal storage would form a closed loop with the heat pump evaporator, the air-handling unit a closed loop with the heat pump condenser, and compressor operation would commence.

No auxiliary backup heating system existed in the building. If the thermal storage tank fell below 45°F (7°C) and freezing of the storage water at the evaporator became possible, city water would be added directly to the thermal storage tank. The city water had a temperature close to 55°F (13°C) in winter and provided an adequate energy source for the heat pump heating operation. Only on one occasion during the two-year field test did the storage tank temperature fall below 45°F (7°C), requiring the addition of city water. Energy supplied by the solar collectors and rejected to the tank by previous heat pump cooling operation provided adequate source energy for all other occasions.

Hourly average rates of mechanical equipment heating; i.e., water-to-water heat pump and direct heating; for day and night periods are shown as a function of the indoor-outdoor temperature differential in Figures 13 and 14. These figures represent information collected over a 3 1/2 month period from December 17, 1975, to March 31, 1976. The use of an indoor-outdoor temperature differential was chosen for presentation since indoor temperature varied somewhat, and thermal losses were a function of this temperature differential and not of outdoor temperature alone. Generally, the indoor temperature was around 75°F (24°C); and this temperature may be used as a reference to approximate outdoor temperature on these plots.

Figure 13 shows how little the heat pump actually operated during the day periods. In most cases, heat pump equipment operation was less than 20 MBtu/hr when averaged over the 12-hour period. The lower points on the graph represent equipment operation during occupied day periods. The water-to-water heat pump operated in either or both the direct heating or heat pump heating mode during early morning hours before the increased electrical consumption during the occupied hours, and solar radiation decreased the equipment heating load. Therefore, although the rate of heating energy delivered during these early hours may have been high, the 12-hour daytime average was low. Usually within one or two hours after the building became occupied, the heat pump would cycle off heating until evening. The scattering of points above the 20 MBtu/hr level represents weekends and holidays when electrical power usage was significantly less. This reduction in internal gains on weekends and holidays required heating in either the direct heating or heat pump heating modes to supply energy to the building throughout most of the day. This is shown by comparing points on this plot to those in Figure 17.

Since night periods have a reduced electrical consumption (Figure 14), the heat pump heating and direct heating operation was much more responsive to the indoor-outdoor temperature differential. It must be remembered that these points do not represent the total thermal loss from the building, but only that part of the thermal loss which was supplied by the water-to-water heat pump unit. A significant portion of building energy needs was supplied to the building by electrical power which was

Figure 13. Rates of Heating Energy Supplied by the Water-to-Water Heat Pump as a Function of the Indoor-Outdoor Temperature Differential for Day Periods

Figure 14. Rates of Heating Energy Supplied by the Water-to-Water Heat Pump as a Function of the Indoor-Outdoor Temperature Differential for Night Periods

dissipated as heat and then stored as thermal energy within the building's structure during day periods. This stored energy and its release from walls, floors, and the building's furnishings was quite substantial, as shown by night periods with temperature differentials as high as 38°F (3°C), requiring no heating equipment operation.

During the day, the water-to-water heat pump operated in the cooling mode, withdrawing room energy and rejecting this energy to the thermal water storage. This operation served to cool primarily the drafting room and the reception area which became overheated due to high lighting levels and solar radiation. The unit operated in a cyclic manner which began in late morning and continued throughout the remaining occupied period. During extremely cold periods, very little cooling was observed. As weather became milder, this operation increased, reflecting reductions in building energy loss.

3.2.2 Water-to-Air Heat Pumps

The water-to-air heat pump loop consisted of five small tonnage water source heat pumps with a combined capacity of approximately 7 tons. The function of these units was to maintain comfort conditions in individual spaces and to maintain proper operating temperatures for the computer facilities. The water-to-air heat pump serving the computer room operated almost continuously in the cooling mode. Others operated on a part-time basis, predominately during occupied hours. The energy rejected by these units provided much of the thermal energy stored in the building's underground tank. Energy rejected during cooling, supplemented by the building's solar collectors, provided energy for

subsequent direct and heat pump heating.

All ventilation air was supplied through the water-to-water heat pump duct system to all building spaces including those served by individual units. Hot air was, therefore, supplied through the ductwork providing heating to those spaces whenever the water-to-water heat pump unit operated in the heating mode and cool air whenever this unit operated in the cooling mode. By this method, all spaces could be heated by the water-to-water heat pump which had a higher coefficient of performance than water-to-air units. The resulting problem was that warm air was supplied to some spaces at times when cooling, or neither heating nor cooling, was required, resulting in increased cooling loads on the individual water-to-air units. This problem produced higher cooling loads for the water-to-air units in January and February, two predominate heating months, than were experienced in March, a month with slightly higher ambient temperatures. This increase was 13% for daytime periods and 20% for 24-hour daily periods. This opposing operation was not observed during periods when the water-to-water heat pump unit operated in the cooling mode.

Day and night hourly averages of water-to-air heat pump loop cooling are shown as a function of indoor-outdoor temperature differential in Figures 15 and 16. These figures represent cooling by the individual heat pump units for the period of December 17, 1975, to March 31, 1975. The operation is shown to have little dependence on the indoor-outdoor temperature differential.

Figure 15. Rates of Energy Withdrawn by Water-to-Air Heat Pump Cooling as a Function of the Indoor-Outdoor Temperature Differential for Day Periods

Figure 16. Rates of Energy Withdrawn by Water-to-Air Heat Pump Cooling as a Function of the Indoor-Outdoor Temperature Differential for Night Periods

There was a strong dependence on electrical energy consumption, especially by that of the computer facilities. A day-to-day analysis showed the water-to-water heat pump cooling values tracking upward and downward following electrical consumption. Cooling from the units is generally greater for occupied, than for unoccupied, periods.

3.2.3 Electrical Energy

The maximum and minimum values for electrical consumption for the two years were similar; no substantial changes in use over the two seasons of data collection were observed. The second year is presented since examination of both years produced essentially the same results.

The electrical consumption of this building was due to interior lighting, building equipment, and office machines, including an IBM 1130 computer used by the building occupants and an IBM 7 computer which performed data collection functions for this field test. In addition, electrical power consumption included the building's mechanical equipment -- fans, heat pump compressors for the water-to-air heat pump units, and all water pumps. The power consumption required by the water-to-water heat pump compressor was accounted for separately and is not included in electrical power consumption information. This choice was made since energy supplied or withdrawn by the water-to-water heat pump was calculated on the load side of the compressor. For heating, compressor energy appeared as part of the total energy supplied by the heat pump's fan coil to the building. During cooling, compressor energy was dissipated to the tank condenser loop and does not appear as energy

supplied to the building. The water-to-air heat pump compressor powers were included since energy flows to and from the building were measured on the storage tank side of the compressors.

Fully occupied day periods had electrical loads peaking in early afternoon with maximum energy draw typically around 39 kW (133 MBtu/hr). At the beginning and end of the work day, this consumption averaged around 22 kW (75 MBtu/hr). The daily 12-hour fully occupied periods had average electrical energy consumption varying roughly between 24 kW and 38 kW (roughly 82 MBtu/hr to 130 MBtu/hr). Minimum occupied electrical consumption was more difficult to assess than peak consumption since distinguishing between full and partial occupation was impossible. When daytime power consumption fell below 20 kW (68 MBtu/hr), average power draw, the assumption that the building was fully occupied was doubtful. During periods when the average power draw was between 14 kW and 20 kW (47.6 MBtu/hr and 68 MBtu/hr), the building was probably partially occupied. This electrical consumption averages to 8.1 watts per square foot ($27.5 \text{ Btu/ft}^2[\text{hr}]$), based on 4300 square feet (400 m^2) of usable floor area during fully occupied periods.

Electrical power consumption was significantly less during weekends, holidays, and night periods. During these periods, electrical consumption rates fell roughly between 9 kW and 13 kW (30 MBtu/hr and 44 MBtu/hr) and represented the base consumption level of the building. Points at which some of the occupants were in the building provided scatter between the occupied and unoccupied periods.

This information is shown by Figures 17 and 18 for the 1975-76 winter heating season. This data represents the period from December 15 through March 31 (108 days). The first figure shows average electrical power consumption for day periods as a function of the indoor-outdoor temperature differential. The highest average power consumption observed is just over 40 kW (138 MBtu/hr). The scales for electrical consumption in MBtu/hr on the left form the grid.

The upper set of points show fully occupied day periods, the middle points show partially occupied day periods, and the lowest show those of weekends and holidays. Dividing this information into months, the average day electrical power consumption was 22.3 kW (76 MBtu/hr) for December, 29.3 kW (100 MBtu/hr) for January, 28.4 kW (97 MBtu/hr) for February, and 27.5 kW (94 MBtu/hr) for March. The lower value for December reflects the high percentage of holidays representing unoccupied periods. The figure shows, as expected, little relation between electrical energy consumption and the indoor-outdoor temperature differential. There is a slight decrease in peaks and averages as temperature differential decreases due to warmer temperatures, reducing mechanical equipment power. The effect is, however, relatively small. The unoccupied day periods at the bottom of the figure correspond very well to observed power for night periods shown in Figure 18.

Similarly, little correspondence was observed between electrical energy consumption and the indoor-outdoor temperature differential for night periods. When the entire season was divided into monthly averages, there was a slight decrease in these night period power consumption

Figure 17. Rates of Electrical Energy Use as a Function of the Indoor-Outdoor Temperature Differential for Day Periods.

Figure 18. Rates of Electrical Energy Use as a Function of the Indoor-Outdoor Temperature Differential for Night Periods

averages as the weather became more mild. For the months of December and January, the average consumption was 12.9 kW and 13.2 kW (44 MBtu/hr and 45 MBtu/hr), respectively. In February and March, the night averages fell slightly to 11.1 kW (38 MBtu/hr) and 10.8 kW (37 MBtu/hr).

3.2.4 Building Occupants

In comparison to other internal gains, energy released by the occupants was small. Since it was impossible to predict how many occupants were in the building at any given time, the occupancy load was based on a constant number of people. This variable was set at 27 which is thought to reflect the average number of occupants. The decision as to whether the building was occupied or unoccupied was based on a mean electrical power consumption of 14.65 kW (50 MBtu/hr). When electrical power draw was in excess of this value, the building was considered occupied; when below this value, the building was considered unoccupied. Based on 250 MBtu/hr (73 W) of sensible energy per person, 6.75 MBtu/hr (2 kW) was added to the building energy input when the building was considered occupied. This was believed to reflect the occupancy loads fairly well.

3.3 Net Energy Delivered as Compared to the Indoor-Outdoor Temperature Differential

3.3.1 1975-76 Winter Heating Season

The average hourly net energy delivered to the building (Figure 19)

Figure 19. Rates of Net Energy Supplied as a Function of the Indoor-Outdoor Temperature Differential for Day Periods

shows how little relation there was between temperature differential and the net energy consumption for day periods. This figure shows that substantially more energy was delivered than was needed to overcome thermal heat loss. The average hourly net energy delivered was the result of adding all heat pump heating, direct heating, electrical energy consumption and heat generation by occupants, and subtracting cooling by both the water-to-water and water-to-air heat pumps. The lower points show minimum values of energy delivery and are closer to that needed to overcome normal heating loads.

It remains, however, that all energy delivered must be stored or dissipated to the surroundings. Energy not removed by conditioning equipment was accounted for in two ways. First, excess energy was stored, as mentioned before, in the building and furnishings by raising their temperatures which then was dissipated at a later time. The remaining energy was dissipated through increased ventilation and infiltration. Forced ventilation by the ventilation fan represented about 36.8 MBtu/hr (10.8 kW) of cooling at a 40°F (22°C) temperature differential. Increased traffic through doors during occupied hours and the occasional leaving open of exterior doors by occupants permitted some additional cooling.

The average hourly net energy delivered to the building during night periods (Figure 20) has much better correlation between energy delivered and the indoor-outdoor temperature differential than those for the day. This is to be expected since the building was closed to traffic; internal gains are much reduced, and the effect of solar gains were not present.

Figure 20. Rates of Net Energy Supplied as a Function of the Indoor-Outdoor Temperature Differential for Night Periods

3.3.2 Comparison of the Two Winter Heating Seasons

This section compares the calculated steady-state energy loss predictions to energy supplied to the Solar Building for the two winter heating seasons of 1974-75 and 1975-76. The comparisons are divided into day, night, and daily periods. No distinction was made in this section between weekday and weekend or holidays. The steady-state loss predictions were shown with 12 or 24-hour hourly averages plotted as points on the respective plots. This set of six figures, Figure 21 through Figure 26, portrays net energy delivered to the building minus ventilation and a nominal value for infiltration for each of the two heating seasons. These values were designated as the adjusted net energy since the ventilation air rate changed from the first to the second year. Subtracting ventilation and a nominal value of infiltration allows direct comparison of the two seasons.

For the six figures in this section the adjusted net energy supplied to the Solar Building was defined as:

$$\text{adjusted net energy} = \text{WHPH} + \text{DH} + \text{ELEC} + \text{PEOP} - \text{WHPC} - \text{AHPC} - \text{VENT} - \text{INFIL}$$

where: WHPH - heat pump heating by the water-to-water heat pump

DH - direct heating from thermal storage to the water-to-water heat pump fan coil unit

ELEC - electrical energy supplied to the building

PEOP - sensible heating by building occupants

WHPC - water-to-water heat pump cooling

AHPC - water-to-air heat pump cooling

Figure 21. Rates of Adjusted Net Energy Supplied as Compared to Steady-State Heating Loads as a Function of the Indoor-Outdoor Temperature Differential for Day Periods, Winter 1974-75

Figure 22. Rates of Adjusted Net Energy Supplied as Compared to Steady-State Heating Loads as a Function of the Indoor-Outdoor Temperature Differential for Night Periods, Winter 1974-75

Figure 23. Rates of Adjusted Net Energy Supplied as Compared to Steady-State Heating Loads as a Function of the Indoor-Outdoor Temperature Differential for Daily Periods, Winter 1974-75

Figure 24. Rates of Adjusted Net Energy Supplied as Compared to Steady-State Heating Loads as a Function of the Indoor-Outdoor Temperature Differential for Day Periods, Winter 1975-76

Figure 25. Rates of Adjusted Net Energy Supplied as Compared to Steady-State Heating Loads as a Function of the Indoor-Outdoor Temperature Differential for Night Periods, Winter 1975-76

Figure 26. Rates of Adjusted Net Energy Supplied as Compared to Steady-State Heating Loads as a Function of the Indoor-Outdoor Temperature Differential for Daily Periods, Winter 1975-76

VENT - forced ventilation air supplied to the building (occupied periods)

INFIL - infiltration losses based on a nominal value of 500 cfm (14.1 m³/min (unoccupied periods)

During the first heating season, the forced ventilation rate was measured to be 1700 cfm (47.9 m³/min) when the ventilation fan operated. Although this fan was operated by a manual switch, it was assumed to be switched to the "on" position during all occupied hours. For the second heating season, the ventilation fan was replaced with a fan having a nominal airflow rate of 1000 cfm (28.3 m³/min). This second year, the fan was controlled by a seven-day clock which switched the fan on at 7:30 a.m. and off at 5:30 p.m., Monday through Friday. These calculations follow that operation and allow direct comparison of the two heating seasons. During unoccupied hours, the infiltration rate was chosen to be the nominal 500 cfm (14.1 m³/min) since the computer program used to calculate the building energy use did not have the capability of regulating the rate to wind speed or temperature differential. These air rates allowed the comparison of day, night, and daily periods without regard to occupied and unoccupied periods.

The solid lines on the figures reflect this adjustment for infiltration and ventilation being subtracted from the energy input to the building. The lower line represents losses from the building envelope including the rates of infiltration and ventilation outlined above. The second line shows the effect of an additional 500 cfm (14.1 m³/min) for a total of 1000 cfm (28.3 m³/min) for unoccupied periods, 2200 cfm

(62.3 m³/min) for occupied periods during the first year, and 1500 cfm (42.5 m³/min) for occupied periods in the second year. The third line shows another additional 500 cfm (14.1 m³/min) for a total of 1500, 2700, and 2000 cfm (42.5, 76.5, and 56.6 m³/min), respectively. This adjustment also allows direct comparison of the two years since additional energy required to overcome the higher ventilation rate the first year has been removed.

Figures 21 and 24 show energy input for 12-hour day periods. The data showed little dependency on the indoor-outdoor temperature differential which was highly dependent on electrical power input and water-to-air loop cooling. The water-to-water heat pump which supplied most of the equipment heating load generally cycled off for the day within an hour after the occupants arrived and increased the usage of electrical power. In these figures, the energy consumption has little relation to heat loss from the building. In addition to high values of energy consumption, the figures show considerable scatter. Although it is difficult to assess how this excess energy was dissipated, a considerable amount was absorbed into walls and furnishings and provided thermal storage within the building. Space temperatures rose during the daytime hours to allow for thermal storage, and cooling equipment also served to remove excess energy. Another method of release of this excess energy was additional infiltration and ventilation caused by occupants entering and leaving the building. No attempt was made to establish a rate for this flow although it is believed to be significant on some occasions.

The two figures for night periods, Figures 22 and 25, show considerably much less energy consumption than was expected. This observation strengthens the assumption of considerable thermal energy storage within the structure. On mild evenings, heating equipment would generally operate only for short periods following occupied days. The heaviest loads on the equipment as expected were early morning hours. These figures do show a reduced scatter for night periods when compared to day periods with a definite relationship between energy supplied and temperature differential. For both years, a considerable rise in adjusted net energy was observed for night periods with indoor-outdoor temperature differentials averaging above 40°F (22°C). This generally represents outside temperatures of less than 35°F (2°C).

Figures 23 and 26 show daily average hourly values of the adjusted net energy vs. indoor-outdoor temperature differential for 1974-75 and 1975-76, respectively. A vast reduction in scatter, as compared to the other figures, was observed. This reduction in scatter is attributed to the energy stored during day occupied periods being dissipated at night. Most of the data points lie above the minimum predicted line and below the maximum predicted line of the additional 1000 cfm (28.3 m³/min) of air infiltration. The average monthly and seasonal temperatures and rates of energy supplied are summarized in Tables 2 and 3. Table 2 summarizes the 1974-75 winter heating season and Table 3 the 1975-76 winter heating season.

Table 2. Summary of Temperatures and Energy Supplied and Withdrawn for the 1974-75 Winter Heating Season

		No. Days	ODBT °F	IDBT °F	I-O Diff. °F	Net Energy MBtu/hr	Elec- tricity MBtu/hr	Water-to-Water HP and DH MBtu/hr	W/W HP Cooling MBtu/hr	W/A HP Cooling MBtu/hr
January	Day	25	37.0	75.1	38.1	95.8	85.9	30.4	3.6	16.9
	Night	25	31.1	74.0	42.9	81.5	35.5	62.9	0.5	16.4
	24-Hour		34.1	74.6	41.0	88.7	60.7	46.7	2.1	16.7
February	Day	18	40.9	74.4	33.5	78.8	93.8	11.7	5.5	21.2
	Night	27	37.3	72.9	35.6	48.6	40.3	26.5	2.2	16.0
	24-Hour		39.1	73.7	34.6	63.7	67.1	19.1	3.9	18.6
March	Day	23	46.5	73.8	27.3	66.9	93.1	15.5	15.4	26.3
	Night	26	41.8	72.5	30.7	39.3	36.3	27.5	4.3	20.2
	24-Hour		44.2	73.2	29.0	53.1	64.7	21.5	9.9	23.3
January-February	Day	43	39.0	74.8	35.8	87.3	89.9	21.1	4.6	19.1
	Night	52	34.2	73.5	39.3	65.1	37.9	44.7	1.4	16.2
	24-Hour		36.6	74.1	37.5	76.2	63.9	32.9	3.0	17.6
January-March	Day	66	41.5	74.4	33.0	80.5	90.9	19.2	8.2	21.5
	Night	78	36.7	73.1	36.4	56.5	37.4	39.0	2.3	17.5
	24-Hour		39.1	73.8	34.7	68.5	64.2	29.1	5.3	19.5

Table 3. Summary of Temperatures and Energy Supplied and Withdrawn for the 1975-76 Winter Heating Season

		No. Days	ODBT °F	IDBT °F	I-O Diff. °F	Net Energy MBtu/hr	Elec- tricity MBtu/hr	Water-to-ater HP and DH MBtu/hr	W/W HP Cooling MBtu/hr	W/A HP Cooling MBtu/hr
December	Day	15	35.9	74.6	38.7	70.8	79.2	26.0	2.8	31.6
	Night		30.6	73.3	42.7	65.3	43.5	50.6	0	30.0
	24-Hour		33.3	74.0	40.7	68.1	61.35	38.3	1.4	30.8
January	Day	31	38.9	78.4	39.4	86.8	104.9	22.7	8.0	34.6
	Night		32.1	74.7	42.6	69.1	44.7	50.8	0.0	25.1
	24-Hour		35.5	76.6	41.0	78.0	74.8	36.8	4.0	29.9
February	Day	29	51.9	77.0	26.9	68.5	103.8	10.5	9.5	34.9
	Night		41.9	73.8	31.9	39.4	38.8	25.0	0.0	24.7
	24-Hour		46.9	75.4	29.4	54.0	71.3	17.8	4.8	29.8
March	Day	31	51.0	76.2	25.2	61.7	98.9	5.0	11.3	30.5
	Night		42.4	72.6	30.3	34.0	36.6	13.2	1.0	18.7
	24-Hour		46.7	74.4	27.8	47.9	67.8	9.1	6.2	24.6
Dec. 15 - Feb. 29	Day	75	43.3	77.1	34.4	76.5	99.3	18.6	7.5	34.1
	Night		35.6	74.1	38.5	56.9	42.2	40.8	0.0	25.9
	24-Hour		39.5	76.6	36.5	66.7	70.8	29.7	3.8	30.0
Dec. 15 - March 31	Day	106	45.6	76.8	31.7	72.2	99.2	14.7	8.6	33.1
	Night		37.6	73.9	36.1	50.2	40.5	32.7	.3	23.8
	24-Hour		41.6	75.4	33.9	61.2	69.9	23.7	4.4	28.4

3.4 Equipment Heating and Cooling as a Function of Time of Day

Preceding sections have shown 12-hour average equipment operation as a function of the indoor-outdoor temperature differential averaged over the 12-hour day and night periods. This section shows how that operation varied at hourly intervals as a function of time of day. There are 11 figures representing the period from December 15, 1975, to March 2, 1976. A one-week period is depicted in each figure beginning at 6:00 a.m. on Monday and continuing until 6:00 a.m. the following Monday. The information was plotted from the 168 hourly average values, comprising a week for each parameter. Each of these 168 points are connected to form a continuous line. These lines show hourly heating and cooling operation and the variation due to the time of day and the day of each week.

The figures are broken into two sections. Hourly values of indoor and outdoor temperatures are plotted in the upper section to show the temperature differentials at each hour. The lower section depicts the heat transfers to and from the water-to-water heat pump during heat pump heating, direct heating and heat pump cooling operation, and from the water-to-air heat pump loop during cooling. In this lower figure, the top line represents total heating by heat pump heating and direct heating. The clear area above the axis shows the energy supplied during water-to-water heat pump heating, and the dotted area above the axis shows the energy supplied during direct heating. Similarly, below the axis the lines show energy removal with the lower line portraying total cooling by the mechanical equipment. The clear area

represents the water-to-air heat pump loop energy removal and the dotted area represents water-to-water heat pump energy removal.

Each plot consists of 168 points spaced one hour apart. These points are connected by straight lines to form the weekly profile. The grid divisions are one division per six hours (four divisions per day).

Days may be clearly separated from one another by the heating and cooling spikes. The cooling spikes occur around 12:00 p.m. for a particular day and the heating spikes generally peak between 12:00 a.m. and 6:00 a.m. for occupied periods, but may continue later into the morning during weekends and holidays. The outside dry bulb, in general, reaches its maximum value during the afternoon (approximately 3:00 p.m.) and its minimum value at approximately 6:00 a.m.

The points plotted for each hour correspond to actual data with the hour defined by its starting time. In the actual data, the hourly data blocks did not necessarily start at the beginning of a clock hour but were dependent on the start-up time of the monitoring equipment. The hour defined as 6:00 a.m. may actually start after 6:00 a.m., but before the next hour, and then continued for 60 minutes.

The first figure, Figure 27, starts on Wednesday, December 17, 1975, at 6:00 a.m., with the beginning of this winter data collection period. A close examination of the indoor dry bulb revealed that the actual temperature varied throughout the day, corresponding to the peaks and troughs of the heating and cooling equipment operation.

Figure 27. Comparison of Heating and Cooling with Temperature Differentials and Time for the Week of December 15, 1975

For example, when the indoor temperature dropped, the mechanical equipment began heating operation. The temperature of the space then rose and the heating equipment cycled off. The converse was true when the space overheated. Through the use of a straight edge set vertically to the figure, this one-to-one corresponding between heating and cooling equipment operation and indoor dry-bulb temperature may be seen. Another observation, which was consistent throughout the entire period, was that the outdoor dry bulb begins to drop in the afternoon and continues to drop for approximately 16 hours; the temperature starts to rise around 7:00 a.m. and continues to do so for about eight hours. This relationship shows that the outdoor temperature rises much more sharply during the day than it falls during the night.

On Friday, the water-to-water heat pump was shut off as shown in Figure 27. No heating operation occurred between 12:00 p.m. on Friday and almost 6:00 p.m. on Saturday. The indoor temperature fell Friday night as a result. At 5:00 p.m. Friday, the indoor temperature was about 78°F (26°C). The indoor temperature fell to almost 60°F (16°C) by 8:00 a.m. the next morning. The energy stored within the building's walls and contents provided energy to the building spaces during this period and slowed the rate of temperature drop within the building spaces.

During Saturday, the water-to-air heat pump was off for several hours; this is shown by the total cooling energy dropping to a zero value. Around 5:00 p.m. Saturday, the water-to-water heat pump began heating the building. The building had been partially occupied during

the day and its interior temperature had risen above 70°F (21°C) and the thermostat reset. For the rest of the weekend, the water-to-water heat pump unit cycled on and off as needed. This period also shows some evidence of the heating and cooling systems opposing each other. The peaks of the heating and cooling energies match. This shows that heat was supplied to spaces which did not require heating and caused the cooling system to cycle on.

During the week of December 22, 1975 (Figure 28), the average outdoor dry-bulb temperature fell, and heating by the heating system increased since the building was only occupied Monday, Tuesday, and Wednesday. The water-to-water heat pump operated on Monday and a small amount on Tuesday and Wednesday. The water-to-water heat pump operation began on Wednesday around 3:00 p.m. and continued through the next day until 12:00 p.m. From 12:00 p.m. until around 12:00 a.m. Thursday, the water-to-water heat pump operated in the direct heating mode to supply energy to the building.

Tracing the time to the above plot of indoor dry bulb ~~75~~ showed that the indoor temperature fell from around 75°F (24°C) at 6:00 p.m. to 70°F (21°C) by 12:00 a.m. Direct heating from the thermal storage was only capable of supplying about 25 MBtu/hr (7.3 kW), so the temperature fell. At 12:00 a.m., the temperature had fallen to the point where the water-to-water heat pump began to operate, supplying heat at a rate of 80 MBtu/hr (23.5 kW), and the temperature rose to 73°F (23°C). After an initial period of overcoming the heat losses from the previous periods, the heat pump operated in a cyclic manner.

Figure 28. Comparison of Heating and Cooling with Temperature Differentials and Time for the Week of December 22, 1975

Friday night showed a similar period of direct heating, shifting to water-to-water heat pump heating. Direct heating Saturday night was almost nonexistent as was the case Sunday. Since the period was cold and there was little solar radiation, the thermal storage temperature fell and was not able to supply the thermal load of the building directly.

The week of December 29 (Figure 29) had the highest heating loads for any week of the year. In addition to low occupancy, this week had the lowest outdoor dry-bulb temperatures. During this week, direct heating accounted for very little of the heating load. At one point, the storage tank fell so low that city water had to be added to raise the tank temperature and prevent freezing of the chiller coil. The only time the water-to-water heat pump operated on cooling was two hours on Monday and a couple of hours on Tuesday. Again, there is evidence of the water-to-water heat pump heating and the water-to-air heat pump cooling opposing each other with the peaks for both heating and cooling matching each other.

Also, during this period some negative direct heating was observed by the water-to-water heat pump system. The negative direct heating occurred when the storage tank temperature fell below the room temperature and the first stage of heating was called, that of direct heating. Cool water was then circulated through the fan coil unit. When the room temperature fell further from the combination of this direct cooling and thermal losses, the heat pump switched to the heat pump heating mode. This operation did not occur very often since the storage tank temperature rarely fell below the room temperature. When it did occur, the

Figure 29. Comparison of Heating and Cooling with Temperature Differentials and Time for the Week of December 29, 1975

mode was of short duration.

Operation returned to normal the week of January 5, 1976 (Figure 30), with outdoor temperatures somewhat warmer. Electrical energy consumption was high since the building was occupied and the heating equipment operation was much reduced from the previous week. The indoor temperature plot shows a temperature drop each evening before the changeover to heat pump heating; and again, there is evidence of heating and cooling opposition. Toward the end of the week a problem developed with the recording of the indoor dry-bulb temperature. Periods in which this problem occurred the indoor temperature was set to 75°F (24°C). This, a result of a problem with channel multiplexing, occurred from time-to-time throughout the rest of the collection period, as evidenced in the figures by straight lines without variation.

During the week of January 12, 1976 (Figure 31), the energy supplied by heating equipment operation and rejected by cooling equipment operation were essentially equal on a daily basis. This week represented a week where all equipment heating could have been supplied by previous energy rejection to thermal storage. Since the building had a high rate of electrical energy consumption, reclamation of this energy was sufficient to supply all heating needs. A very high direct heating peak is shown for Wednesday of the following week (Figure 32) as a result of an elevated aboveground storage tank temperature. The aboveground tank temperature increased as a result of several days of good solar collection and reduced heating requirements.

Figure 30. Comparison of Heating and Cooling with Temperature Differentials and Time for the Week of January 5, 1976

Figure 31. Comparison of Heating and Cooling with Temperature Differentials and Time for the Week of January 12, 1976

Figure 32. Comparison of Heating and Cooling with Temperature Differentials and Time for the Week of January 19, 1976

By January 26, 1976, heat pump heating operation had become very cyclic with shorter and shorter periods of operation (Figure 33). The large slug of heat pump heating on Tuesday night was the last large period of operation during this heating season. The first week in February was the first of two mild weather weeks (Figure 34), with energy removal clearly greater than energy supplied. This trend continued for the remaining three weeks (Figures 35, 36, and 37) with heating spikes intercepting the origin more and more frequently, reflecting hours with no heating requirements.

The operation of both heating and cooling equipment was shown clearly to be dependent on both the building electrical energy usage and the indoor-outdoor temperature differential. Most heating occurred during unoccupied periods, which were predominately at night; cooling by the water-to-water heat pump began around 12:00 p.m. and continued for up to six hours during mild temperatures when the building was occupied.

3.5 Energy Delivered and Withdrawn as a Function of Time of Day

This section presents energy supplied and removed by mechanical equipment, electrical power consumption, and sensible heating from occupants. The information is presented to show total energy supplied and removed and also the method by which this energy was supplied and removed. The electrical power consumption shown in this section includes sensible gains from occupants. This inclusion was done to provide figures which were less confusing. The additional heating

Figure 33. Comparison of Heating and Cooling with Temperature Differentials and Time for the Week of January 26, 1976

Figure 34. Comparison of Heating and Cooling with Temperature Differentials and Time for the Week of February 2, 1976

Figure 35. Comparison of Heating and Cooling with Temperature Differentials and Time for the Week of February 9, 1976

Figure 36. Comparison of Heating and Cooling with Temperature Differentials and Time for the Week of February 16, 1976

Figure 37. Comparison of Heating and Cooling with Temperature Differentials and Time for the Week of February 23, 1976

energy from occupants only occurred during occupied hours and represents a small amount in comparison to electrical loads (5 to 10% or an addition of 6.75 MBtu/hr [2.0 kW], during occupied periods). To show the effect of time of day and day of week, the figure represents a one-week period as before. The beginning and the end of the week are divided as before at 6:00 a.m. Monday morning. Also, as before, each figure has four divisions per day, representing six hours with the parameters being plotted from hourly values. The temperature differential was not repeated in this section since reference may be made to the previous section if desired.

In these figures the area from the top line to the axis represents the net energy supplied to the Solar Building. The bottom line to axis shows the total energy withdrawn, by water-to-water and water-to-air loop heat pump cooling. The total area between the top and bottom lines represents all energy; e.g., total energy supplied; delivered to the building by mechanical equipment and electrical power consumption. The electrical power consumption is represented by the shaded area. Heat pump heating and direct heating are shown by the clear area below the top line and above the dotted area.

There is a clear division portrayed between days which are occupied and days which are holidays or weekends. During occupied days an electrical energy consumption spike occurred around 7:00 a.m. and continued at a high value until 6:00 p.m. On days which were holidays or weekends, this spike was at least partially occupied. These partially occupied periods are recognizable since electrical energy

usage increased during these periods although at a reduced level from fully occupied periods. Fridays show a reduction in energy consumption around 12:00 p.m. since the employees normally work 9-hour days, Monday through Thursday, and only until 12:00 p.m. on Friday.

The first figure (Figure 38) again begins on Wednesday with the beginning of the data collection for the second season. Of these five days, three days (Wednesday, Thursday, and Friday) were occupied. Saturday and Sunday were not. The figure shows resultant heating dropping during the afternoon as cooling increased although the electrical power consumption remained more constant. This increased cooling created cooling spikes below the axis, reducing net energy supplied while total energy supplied remained fairly constant. Total energy supplied is represented by the area on both sides of the axis. The effect of electrical energy consumption on heat pump heating and direct heating is clearly shown by heating equipment operation ending fairly quickly after the building became occupied and then began in the evening as the electrical energy usage was reduced.

The week of December 22, 1975 (shown in Figure 39), had three occupied days (Monday, Tuesday, and Wednesday) and four unoccupied days (Thursday through Sunday). This reduction in electrical energy usage for the four unoccupied days resulted in a reduction in the total energy supplied to the building and an increase in heating equipment operation. Heat pump cooling was also reduced. The net energy supplied again dropped off with increased cooling during the first three days. On the fourth day most energy was supplied to the building as a result of

Figure 38. Comparison of Electrical Energy Usage to Heating Energy Supplied and Cooling Energy Withdrawn, Week of December 15, 1975

Figure 39. Comparison of Electrical Energy Usage to Heating Energy Supplied and Cooling Energy Withdrawn, Week of December 22, 1975

thermal energy storage within the building structure. The water-to-air loop heat pumps were off for several hours Thursday afternoon. By the end of the week the water-to-air heat pumps removed energy at a fairly constant rate; electrical energy was used at a nearly constant rate. The water-to-water heat pump is shown to have operated in a cyclic manner by its saw-toothed shape delivering energy to the building during every hour of the weekend.

Monday, December 29, 1975 (Figure 40), showed high electrical energy use beginning at 6:00 a.m. with heating operation ceasing rather quickly. At 5:00 p.m., heating operation again began and continued until the next morning. The first three days were again occupied, and the following four were unoccupied. This week was the coldest week of the year with very high rates of water-to-water heat pump heating during the last four days. This week showed the cyclic nature of the water-to-water heat pump even during very cold periods.

The week of January 5, 1976 (Figure 41), was the first week examined with five days of occupancy in this second season. Saturday was also partially occupied. Six electrical energy consumption peaks are shown with accompanying cooling spikes in early afternoon, one for each of the occupied days. Again, cooling spikes reduced the net energy supplied; and water-to-water heat pump heating occurred mostly during unoccupied periods. Sunday also had an interesting several hours where the net energy supplied was negative since more energy was removed than was supplied for several hours.

Figure 40. Comparison of Electrical Energy Usage to Heating Energy Supplied and Cooling Energy Withdrawn, Week of December 29, 1975

Figure 41. Comparison of Electrical Energy Usage to Heating Energy Supplied and Cooling Energy Withdrawn, Week of January 5, 1976

Figure 42. Comparison of Electrical Energy Usage to Heating Energy Supplied and Cooling Energy Withdrawn, Week of January 12, 1976

Figure 43. Comparison of Electrical Energy Usage to Heating Energy Supplied and Cooling Energy Withdrawn, Week of January 19, 1976

Figure 44. Comparison of Electrical Energy Usage to Heating Energy Supplied and Cooling Energy Withdrawn, Week of January 26, 1976

Figure 45. Comparison of Electrical Energy Usage to Heating Energy Supplied and Cooling Energy Withdrawn, Week of February 2, 1976

Figure 46. Comparison of Electrical Energy Usage to Heating Energy Supplied and Cooling Energy Withdrawn, Week of February 9, 1976

Figure 47. Comparison of Electrical Energy Usage to Heating Energy Supplied and Cooling Energy Withdrawn, Week of February 16, 1976

Figure 48. Comparison of Electrical Energy Usage to Heating Energy Supplied and Cooling Energy Withdrawn, Week of February 23, 1976

The week of January 12, 1976 (Figure 42), had five occupied days with Saturday and Sunday partially occupied as shown by the electrical power consumption. Daytime cooling increased; and changes in rates of heating, cooling, and power consumption are very apparent with increased outdoor temperatures. The remaining six weeks examined show the increasing effect of rising ambient temperatures. Heat pump heating and direct heating decreased, while electrical power consumption continued to vary mostly as a function of building usage. On occasion the net energy supplied to the building became negative, representing high cooling rates in response to higher outdoor temperatures, solar radiation, and electrical power consumption. These six weeks are shown in Figures 43 through 48. All figures provide information to confirm expected trends and performance.

3.6 Energy Delivered and Withdrawn as a Function of Day of Year

To present actual energy consumption on a daily basis and to provide a base for comparison of the two winter heating seasons, this section presents daily average hourly energy transfers and temperatures. Heat pump heating and direct heating, heat pump cooling, and electrical energy usage are shown similar to that in Section 3.5 using the 24-hour average quantities for presentation rather than the one-hour averages as before. Each figure contains a month of information. Actual indoor and outdoor temperatures are shown at the top of each figure. In the lower plot the net heating energy is shown from the top line to the axis; the net electrical consumption is represented by the area from the second line to the axis, and the total cooling energy from the axis

to the lowest line. Total input energy is represented by the entire area between the top and bottom line on the plots. Total electrical consumption is shown by the dotted area, and total heat pump heating and direct heating input energy is represented by the area containing vertical lines.

The periods represented are January 1975 through April 1975, representing the 1974-1975 heating season, and December 1975 through February 1976, representing the 1975-1976 winter heating season. Observations and analysis of data collected showed many similar occurrences for each year. The accompanying seven figures, Figure 49 through Figure 55, show high values of equipment heating during cold periods and a decrease as weather becomes more mild. Heat pump cooling was relatively constant during cold periods, then increased as weather became mild. The electrical energy consumption again was shown to have little dependency on the temperature differential, but was a function of building use.

Both years portray similar performance. In December and January high equipment heating rates prevail with lower rates of equipment cooling. By February, heating by the water-to-water heat pump and direct heating balance cooling by the heat pumps; thereafter, the cooling load exceeded the heating load. The electrical power consumption was dependent on occupant use rather than on outdoor temperature, with a base load which varied around 12 kW (about 40 MBtu/hr). The peak power consumption for a particular hour was as great as 40 kW (about 140 MBtu/hr) but averaged around 28 kW (about 95 MBtu/hr) for

weekdays. As the weather became warmer, more and more of this electrical energy was removed by mechanical cooling since less was used to overcome thermal losses from the building structure.

Although data collection for the 1974-75 heating season began near the end of November, several reoccurring problems prevented complete analysis until near the end of December. For this reason, January 1975 is the first month discussed. This month began during a cold weather period having a net energy consumption of 100 MBtu/hr (29 kW) with approximately 30 MBtu/hr (9 kW) of water-to-air unit cooling on a daily basis. On the first several days heating equipment output averaged around 80 MBtu/hr (23 kW) for weekends and 40 - 50 MBtu/hr (12 - 15 kW) for occupied weekdays. By the end of the month, heat pump heating and direct heating outputs had reduced significantly. By the end of January, electrical power usage provided most of the building's heating directly. The additional heating necessary was provided by the water-to-water heat pump in the heat pump heating and direct heating modes. By the end of this month, heat rejected to the thermal storage by cooling was approximately equal to that withdrawn by the heating equipment.

In February, as weather became increasingly mild, heat pump heating and direct heating reduced to 20 - 40 MBtu/hr (6 - 12 kW) on a daily basis. Since electrical power consumption was independent of outdoor temperature, the water-to-air heat pump cooling increased as did that of water-to-water heat pump cooling. By the end of February, the energy removed by cooling exceeded that supplied by heating consistently.

Figure 49. Daily Energy Usage for January 1975

Figure 50. Daily Energy Usage for February 1975

By the beginning of March 1975, some very warm days with average temperatures around 50°F (10°C) occurred. Net heating to the building was further reduced. Cooling increased to a value approximately half of the total electrical input energy. During March and April 1975, the daily average heat pump and direct heating supplied fell to low levels as did cooling in January. The heating occurred during late night and early morning hours after the stored thermal energy within the building's structure had been dissipated. During daytime periods, the electrical energy dissipated by lights and equipment supplied all heating needs.

Data collection analyses for the second heating season began about the middle of December 1975. This study started at 6:00 a.m., December 17. Again, the electrical energy consumption rose steeply on Monday and fell steeply at the end of the week. The plot clearly shows electrical energy use dropping off on Thursday, December 25 and then rising again on Monday, December 29. This period also clearly shows that more energy was lost to the surroundings during occupied days due to the ventilation air rate being higher than the infiltration rate (except on windy days), additional heat losses due to opening of doors, to traffic in and out of building, and higher indoor temperatures resulting from high energy consumption.

January began with very cold weather. The days of January 3 through January 5 show very high heat pump heating rates. A possible reason could have been high infiltration rates due to ventilation fan left on or door not closed properly. By the middle of the month, the

Figure 51. Daily Energy Usage for March 1975

Figure 52. Daily Energy Usage for April 1975

Figure 53. Daily Energy Usage for December 1975

Figure 54. Daily Energy Usage for January 1976

heating energy supplied by the mechanical equipment matched the heat pump cooling energy withdrawn, and more of the heat generated by electrical consumption was withdrawn by cooling equipment operation.

For February, roughly half of the electrical power supplied to the building is withdrawn in cooling the spaces by the heat pumps. Throughout this month, the heat withdrawn from the building exceeded energy delivered for heating by the mechanical equipment. This building with the high internal heat gains due to electrical power consumption was heated most of the winter by using the heat rejected to the storage tank during cooling.

Figure 55. Daily Energy Usage for February 1976

4.0 BUILDING THERMAL LOAD

4.1 Thermal Losses and Methods of Calculation

This chapter describes actual energy consumption as it related to the calculated heating load for the Solar Building during the winter heating season of 1975-76. Data collection began on December 15, 1975, and continued through April 30, 1976. For winter heating and related building thermal performance, the period through the end of February was most important. Although not discussed, several areas of analysis continued through the end of March, allowing an extended period of study for some parameters. As mentioned before, during March the weather became mild; and cooling operation became much more dominant than that of heating.

The thermal losses from the Solar Building were calculated using both steady-state and transient procedures through use of the Building Environmental Analysis Program (BEAP) (1). This program was developed and written by the staff of the Department of Architectural Engineering at The Pennsylvania State University. The program performs air conditioning load designs and is able to accept various building shapes and material combinations. The program calculates instantaneous heat flows on an hourly basis. These instantaneous heat gains and losses are converted to conditioning loads by use of weighting techniques developed by the National Research Council of Canada and adopted by the American Society of Heating, Refrigeration and Air Conditioning Engineers (ASHRAE). The heat transfer coefficients required as inputs

for this method were obtained from the ASHRAE Handbook of Fundamentals (5). Building infiltration rates were inputted to this program based on infiltration studies of The Solar Building performed by Robert Doeffinger (6).

Commercial buildings typically have high internal energy consumption during occupied hours. High lighting levels, combined with office equipment including computational facilities, may produce heating far in excess of building conduction, ventilation, and infiltration losses. The Solar Building fell within this category requiring mechanical equipment cooling of spaces even during the coldest days. Internal spaces and those with computational equipment required cooling almost continuously. Since energy consumption did not necessarily reflect the energy losses from the building, special care was taken when comparing these two quantities.

The analysis of energy consumption and energy losses from the building was performed using a conservation-of-energy approach. It is known that energy input to the building must always equal energy withdrawn by mechanical equipment, plus energy lost to the surroundings, plus the change in energy stored within the building. While energy supplied as a result of electrical energy consumption and mechanical equipment operation and that withdrawn by mechanical equipment operation may be accurately assessed for any finite period of time to within the error tolerances of the measuring devices, other energy transfers had to be calculated using methods which may or may not have been based on measured flows. So while energy must be

conserved, the precise accounting for each type of energy transfer was not always possible. This was especially true for short periods of time where the storage or release of thermal energy stored in walls and furnishings accounted for significant portions of that energy.

The minimum period in which the measured and the calculated energies were expected to match was the 24-hour daily period. The data were, however, examined and energy flows predicted at hourly intervals. The shorter period was chosen to gain a better understanding of the building's operation and enable better predictions of longer periods of time.

The operating data and energy calculations in this chapter are presented in: 12-hour day periods, 6:00 a.m. to 6:00 p.m.; 12-hour night periods, 6:00 p.m. to 6:00 p.m.; and 24-hour daily periods, 6:00 a.m. to 6:00 a.m. In the final section, figures representing actual and calculated heating loads are presented for weekly and monthly intervals using hourly and daily averages, respectively.

The second year was chosen for a more detailed analysis since several problems encountered in the first heating season study were corrected by better instrumentation. These changes resulted in improved assurances that the data and assumptions were valid.

4.2 Conduction, Ventilation, and Infiltration Losses

The conduction losses were those resulting from heat conduction through the envelope of the building structure as a result of an indoor-outdoor temperature differential. This temperature differential resulted in a thermal gradient across the walls, windows, and roof structures. Although entirely a function of the temperature, solar radiation reduced these thermal losses by raising the temperature of sunlit walls, roof, and the immediately surrounding outdoor air temperature. Wind increased the heat losses by increasing the outdoor surface turbulence and thereby increasing the outdoor surface film conductance.

Airflows through the Solar Building due to infiltration and forced ventilation accounted for significant proportions of the building's total heating load. Infiltration into the building produced a direct load within the conditioned spaces. Forced ventilation, air brought into the building through the conditioning equipment, imposed a load directly on the conditioning equipment as well as imposing some load in the conditioned spaces proportional to the coil bypass factor. In Albuquerque, New Mexico, this load was entirely sensible for heating and almost entirely sensible for cooling since the area has a very dry climate.

The pressure gradients responsible for infiltration consisted of two contributing factors. The first, wind velocity, produced a buildup of pressure on the windward side and a slight draft on the leeward side.

These pressure differentials caused an increased and a decreased static pressure on opposing sides of the building. The resulting pressure created infiltration on the windward side and exfiltration on the leeward surfaces.

The second factor contributing to infiltration was produced from variations in air density caused by air of different temperatures. In winter, the colder more dense air tended to infiltrate into the building at floor level, allowing warmer and lighter air from the building's interior to escape near the top of walls and through the ceiling and roof structure. The combination of infiltration from wind velocity and temperature gradients is generally proportional to the square root of the sum of the two pressure forces acting on the building. In this case, since the building is only one story, the stack effect was minimal and the wind forces dominated. A correlation between the temperature differential and wind velocity was observed. As the temperature dropped, wind velocity generally increased and thereby caused increased infiltration.

During occupied periods, a ventilation fan operated to insure a constant supply of fresh air to all spaces. For the first year the ventilation fan was operated manually. The fan was switched on at the beginning of the work day and off as occupants left the building. The airflow rate for this season was measured at 1700 cfm ($.802 \text{ m}^3/\text{sec}$) (6), representing 2.5 air changes per hour. Three exhaust fans operated continuously, each exhausting 50 cfm ($.024 \text{ m}^3/\text{sec}$). The remaining 1550 cfm ($.732 \text{ m}^3/\text{sec}$) produced a slight positive pressure

within the building spaces which escaped by means of exfiltration through exterior surfaces. To control the rate of forced ventilation more accurately the second year, a seven-day ventilation clock was installed which turned the fan on at 7:30 a.m. and off at 5:30 p.m., Monday through Friday. In addition, the ventilation fan was replaced by a smaller unit which was then adjusted by means of a damper to allow 1000 cfm ($.472 \text{ m}^3/\text{sec}$) to enter the spaces. This 1000 cfm ($.472 \text{ m}^3/\text{sec}$) corresponded to approximately 1.5 air changes per hour.

Infiltration studies were performed on the building by Robert C. Doeffinger (6) during the summer of 1975. Using a tracer gas technique, precise measurements of building air change rates were obtained. The rates of building infiltration were based on this study. Unfortunately, these measurements were made only during a period of low wind speed and relatively small temperature differentials. Extrapolation of this data was necessary for winter periods containing large temperature gradients and higher wind speeds. These predicted infiltration rates, based on wind speed alone, were then used in predicting building thermal loads. The rates were found to be very acceptable and correlated well within uncertainty of other areas.

The test data indicated that infiltration rates between 0.5 and 2.5 air changes per hour could be expected, representing 334 to 1670 cfm ($.158$ to $.788 \text{ m}^3/\text{sec}$), respectively. The procedure to calculate infiltration used 500 cfm ($.236 \text{ m}^3/\text{sec}$) of infiltration at a base wind velocity of 5 mph. This base value was then extrapolated proportional to the square root of the wind speed.

Calculation of the heating load was performed through the use of the Building Environmental Analysis Program, Version 4, (BEAP) (1). Use of this program required all wall structures, windows and roofs, and their orientations to be specified as input parameters. These inputs in the form of heat transmission coefficients were converted into transfer functions by the program. Building construction information used in these calculations is contained in Appendix A.

Heat losses and gains are calculated by the BEAP program at hourly intervals and then converted to an hourly load. To perform these calculations, outdoor temperature, horizontal solar radiation, and wind speed were inputted hourly. The indoor temperature was fixed at 75°F (24°C). Solar radiation on walls, through windows, and on the roof was calculated based on the horizontal solar radiation. The program calculated the azimuth and altitude angles and the resulting angle of incidence for each surface based on the horizontal solar radiation, clock hour, and day of the year. Heat losses and gains were then calculated using sol-air temperatures as described in the ASHRAE Handbook of Fundamentals (5). The outdoor temperature and horizontal solar radiation used were obtained from collection devices at the building site. The wind velocities used as inputs were obtained from the National Oceanographic and Atmospheric Administration as recorded at the Albuquerque Airport Weather Station (7). These wind velocities changed on a daily basis and represented averages for the particular day. Direction was set from the predominant direction northwest. Relative humidity and atmospheric pressure were set at constant values throughout the period.

4.3 Thermal Storage

During occupied hours, the high electrical power consumption rate resulted in the heating of spaces in excess of that needed to overcome the thermal losses. Energy dissipated as heat from lights and equipment increased the building space temperature. This increase in space temperature allowed storage of thermal energy in walls, floors, and furnishings. Even after mechanical equipment cooling removed energy, a significant amount of this stored energy remained. The release of this stored energy reduced the heating equipment load below that expected in evening hours.

An estimate of the stored energy within the building was made using 50 lb/ft² (244 kg/m²) to describe the internal mass. The internal capacitance was assumed to be 0.22 Btu/lb °F (0.22 kCal/kg °C) and the total floor area was approximately 5000 ft² (465 m²). A one degree change in building temperature resulted in 55 MBtu (13.9 kCal) of heat energy.

$$50 \text{ lb/ft}^2 \times 5000 \text{ ft}^2 \times 0.22 \text{ Btu/lb} = 50,000 \text{ Btu/}^\circ\text{F} = 16.1 \text{ kWh/}^\circ\text{F} \\ = 25,000 \text{ kCal/}^\circ\text{C} \text{ (the } 50 \text{ lb/ft}^2 \text{ includes walls, floors, and furnishings).}$$

With a five degree temperature rise during the day, 275 MBtu (80.6 kWh) could be stored, and a five degree temperature swing was not uncommon. If this energy was dissipated evenly over a 12-hour night period, it would represent a reduction in required heating of 22.9 MBtu/hr (6.7 kW). To reduce the effect of storage and release of thermal energy

within the structure, the daily period was divided in early morning before occupants entered the building. By beginning the daily period at 6:00 a.m., the majority of the previous day's internally stored energy was dissipated. Examination of data showed this to be true for periods in which the thermostat setting for consecutive days was constant and weather patterns similar. Daily periods, usually Friday, when an uncontrolled thermostat setback was encountered, the building thermal storage did have an effect on the following day's mechanical equipment operation. Changing weather patterns similarly could effect the internal storage of energy by changing the exterior wall temperatures and rates of energy gains and losses from the structure. For most time periods the decision to divide the daily period at 6:00 a.m. reduced problems associated with internal energy storage significantly. This was especially advantageous since the BEAP program did not allow inputting a continuously varying indoor temperature or continually varying electrical power as the building actually experienced.

4.4 Steady-State Load Calculations

Steady-state load calculations were performed with indoor-outdoor temperature differentials from 10°F to 70°F (5.6°C to 38.9°C) in ten-degree steps. For each calculation the outdoor temperature and indoor temperature was fixed, and the BEAP program was allowed to run to obtain a steady-state load profile. Excluding infiltration and forced ventilation and without solar effects, the calculations resulted in a load described by:

$$0.936 \text{ MBtu/hr} \times \Delta T \quad (0.274 \text{ kW} \times \Delta T)$$

where ΔT is the indoor-outdoor temperature differential in °F. For these calculations the indoor temperature was fixed at 75°F (24°C). Next, the calculations were again performed using design solar radiation data for January 21 and the solar heat gain factors for a 32° north latitude from the ASHRAE Handbook of Fundamentals (5) inputted at hourly intervals. The solar radiation reduced the daily load by 118.8 MBtu/day (34.8 kWh/day) or 9.9 Btu/hr (2.9 kW) when averaged over a 12-hour daytime period. The equation for this load was:

$$0.936 \text{ MBtu/hr} \times \Delta T - 9.9 \text{ MBtu/hr} \quad (0.274 \text{ kW} \times \Delta T - 2.9 \text{ kW})$$

The steady-state load without solar effects resulted in a 0.0 MBtu/hr (0.0 kW) intercept at 0°F (0°C) temperature differential. The steady-state load with solar effects resulted in an 0.0 MBtu/hr (0.0 kW) intercept at an indoor-outdoor temperature differential of 10.5°F (5.8°C). At a 70°F (38.9°C) temperature differential the conductive thermal load was 65.5 MBtu/hr (19.1 kW), and 55.6 MBtu/hr (16.3 kW), respectively. Steady-state night periods resulted in heat loads described by the first equation, and steady-state day periods resulted in a heat load described by the second equation. The steady-state conductive losses for the 24-hour daily period resulted in a load described by the average of the two previous equations:

$$0.936 \text{ MBtu/hr} \times \Delta T - 4.95 \text{ MBtu/hr} \quad (0.274 \text{ kW} \times \Delta T - 1.45 \text{ kW})$$

and had a 0.0 MBtu/hr (0.0 kW) intercept at 5.3°F (2.9°C) and a 60.6 MBtu/hr (17.8 kW) intercept at an indoor-outdoor temperature differential of 70°F (38.9°C).

The above conduction loads were calculated with a wind speed of 10 mph. The changes in thermal conduction through walls, windows, and roofs, as a result of varying wind speeds and a sinusoidal varying temperature differential as opposed to a constant daily temperature, were relatively small. By increasing the wind speed to 20 mph, the conduction thermal load was increased approximately 6%. A daily sinusoidal varying outdoor temperature decreased night thermal load by approximately 5% and increased daytime load about 10%, resulting in an increased daily load of approximately 5%. Their combined effect averaged under 10% in these tests. The simplification of a 10 mph wind speed and a steady-state temperature is used for presentation. It should be remembered that increasing wind speed will increase the conductive load slightly, while the effect of a sinusoidal daily temperature will decrease the nighttime load and increase the daytime load. The 24-hour daily load will also increase slightly.

A second set of steady-state computer calculations was performed to calculate the total building thermal load; i.e., the heating load resulting from the sum of the conduction and convection losses; with convection losses being the sum of the thermal losses resulting from ventilation and infiltration. These computer calculations used the same parameters as before, but also included infiltration and ventilation for speeds of 10 mph and 20 mph. For wind speeds of 12 mph or less, the ventilation fan operation during occupied hours was able to keep the building slightly pressurized. For occupied periods with wind speeds greater than 12 mph, infiltration produced an additional load on the building. During periods when the ventilation fan was

off, convection losses were a function of infiltration alone.

The total steady-state heating load for occupied and unoccupied day periods (the ventilation fan operated 10 hours for occupied periods), night periods, and 24-hour daily periods (with occupied and unoccupied 12-hour day periods) for wind speeds of 10 mph and 20 mph are presented in Table 4. This information is presented for occupied periods in Figure 56 and is used in discussions of the balance-point temperature.

4.5 Balance-Point Temperatures

As expected, the building balance-point temperature was highly dependent on building use; i.e., whether or not the building was occupied. Balance-point temperatures were identified for 12-hour occupied day periods, 12-hour unoccupied day periods, and 12-hour night periods for an indoor temperature of 75°F (24°C). The balance-point temperature was also dependent upon the method by which it was defined. If all electrical energy usage was allowed to offset the heating load, one set of balance point temperatures would be defined. By this method, cooling operation by the water-to-air heat pump units would not be allowed. Excess heating of the computer facilities could then occur. Allowing the water-to-air heat pump units to operate, the operation removed energy and reduced the energy available for heating. This second method defined the balance-point temperature as the temperature at which water-to-water heat pump operation was expected to begin. For completeness, results of both methods are shown

Table 4. Steady-State Load Equations

	10 mph wind speed	20 mph wind speed
12-hour day periods		
occupied	$2.18 \text{ MBtu/hr} \times \Delta T - 9.9 \text{ MBtu/hr}$	$2.82 \text{ MBtu/hr} \times \Delta T - 9.9 \text{ MBtu/hr}$
unoccupied	$1.84 \text{ MBtu/hr} \times \Delta T - 9.9 \text{ MBtu/hr}$	$2.82 \text{ MBtu/hr} \times \Delta T - 9.9 \text{ MBtu/hr}$
12-hour night periods		
unoccupied	$1.84 \text{ MBtu/hr} \times \Delta T$	$2.82 \text{ MBtu/hr} \times \Delta T$
24-hour daily periods		
with occupied days	$2.12 \text{ MBtu/hr} \times \Delta T - 4.95 \text{ MBtu/hr}$	$2.82 \text{ MBtu/hr} \times \Delta T - 4.95 \text{ MBtu/hr}$
with unoccupied days	$1.84 \text{ MBtu/hr} \times \Delta T - 4.95 \text{ MBtu/hr}$	$2.82 \text{ MBtu/hr} \times \Delta T - 4.95 \text{ MBtu/hr}$

The ΔT in the above equations represents the indoor-outdoor temperature differential in °F.

Figure 56. Steady-State Thermal Load as a Function of the Outdoor Temperature

in Table 5, although the one disallowing water-to-air cooling operation is preferred.

The heating load lines varied for occupied and unoccupied periods since the ventilation fan operated providing a continuous supply of outside air during occupied periods. Wind velocities also affected infiltration; this infiltration resulted in changes in the heating load line which, again, affected the balance-point temperature differential. To simplify the variables, presentation is based on the steady-state loads presented in the previous section and seasonal average values of electrical energy usage and water-to-air heat pump cooling. The figures in this section are based on Figure 56 and are divided into night, day, and 24-hour daily periods.

Figure 57 shows the daytime steady-state losses for day periods for conductive thermal losses and total thermal losses for a 10 mph wind with ventilation fan off, 10 mph wind with ventilation fan on, and for a 20 mph wind. At a 20 mph wind the ventilation fan had little effect; and therefore, no difference between occupied and unoccupied periods are shown. The total average electrical consumption is 116 MBtuh (34 kW). If all this electrical energy were allowed to be dissipated to overcome thermal losses, the balance-point temperature would be 17°F (-8°C) for a 10 mph wind and 30°F (-1°C) for 20 mph wind. Allowing the water-to-air heat pump to operate on cooling results in a rise of the balance-point temperatures to 33°F (1°C) and 42°F (6°C). For unoccupied periods, similar balance point temperatures may be located.

Table 5. Balance-Point Temperatures

	<u>Total Electrical Energy Usage</u>				<u>Net Electrical Energy Usage*</u>			
	<u>Occupied</u>		<u>Unoccupied</u>		<u>Occupied</u>		<u>Unoccupied</u>	
	<u>10 mph wind</u>	<u>20 mph wind</u>	<u>10 mph wind</u>	<u>20 mph wind</u>	<u>10 mph wind</u>	<u>20 mph wind</u>	<u>10 mph wind</u>	<u>20 mph wind</u>
Day	17°F	30°F	47°F	57°F	33°F	42°F	62°F	66°F
Night	--	--	52°F	60°F	--	--	67°F	70°F
24 Hours	33°F	45°F	50°F	58°F	48°F	56°F	64°F	68°F

* Net Electrical Energy Usage = Total Electrical Energy Usage - Water-to-Air Heat Pump Cooling

Figure 57. Steady-State Thermal Load with Corresponding Balance-Point Temperatures for Day Periods

The night period has just one set of balance-point temperatures (Figure 58) since the building was not normally occupied at night. Daily values of the balance-point temperature differential are shown in Figure 59. All of these balance-points have been summarized in Table 5. The two most important balance-points are for 24-hour periods with 10 mph winds and defined by the total electrical energy usage. The outdoor balance-point temperatures with total electrical energy usage offsetting thermal losses and 10 mph winds are 33°F (1°C) for occupied 24-hour periods and 50°F (10°C) for unoccupied 24-hour periods.

4.6 Transient Load Calculations

4.6.1 Hourly Comparisons of Net Energy Consumption and Calculated Thermal Load

The BEAP program discussed in Section 2 of this chapter was also used to calculate the hourly heat losses for the 1975-1976 heating season from actual recorded data. The results of these calculations are presented here. The first part of this section compares the hourly net energy consumption of The Solar Building to the hourly calculated load for an 11-week period.

The second sub-section compares the mechanical equipment operation to the calculated load for a one-week period during this heating season. The final sub-section presents the net energy consumption and calculated load using daily averages of the hourly rates for the 11-week period.

Figure 58. Steady-State Thermal Load with Corresponding Balance-Point Temperatures for Night Periods

Figure 59. Steady-State Thermal Load with Corresponding Balance-Point Temperatures for Daily Periods

These daily averages are presented in monthly intervals.

This analysis was performed for the 11-week period from December 15, 1975, through February 29, 1976. The calculated load predicted by the BEAP program using actual weather conditions was plotted at hourly intervals. The 11 figures (Figure 60 through Figure 70) in this section each represent a one-week period of 168 hours. The figures represent hourly values beginning at 6:00 a.m. Monday and continuing to 6:00 a.m. the following Monday. To make the comparison of calculated heating load to net energy consumption, each figure contains two plots. The upper plot shows net energy consumption of the Solar Building versus time, and the lower plot shows calculated heating load of the Solar Building versus time. Again, the net energy consumption was defined as the sum of electrical energy consumption plus mechanical equipment heating minus mechanical equipment cooling. The scales have been set equal in both plots to provide easy comparison between the two plots. Time divisions have been provided every six hours with 6:00 a.m. being defined on the plots with a corresponding letter representing the day of the week. The area below the curves has been shaded to increase clarity of the figures. As before, spaces in which the net energy consumption line is broken represent gaps in the data collection. The shaded area continues through these gaps, joining the data breaks with the most likely energy consumption rates.

The lower plot has also been divided into total heating load and conductive energy load alone. The area between the

axis and the top curve represents the total energy load. The area between the axis and the lower curve represents the building conduction load, and the area between the top and lower curve represents that which was calculated as a result of ventilation and infiltration losses.

These sets of net energy consumption and calculated heating load are not necessarily directly comparable on a one-to-one correspondence. The upper plot includes energy delivered to the building to provide the building occupants with lighting and with office equipment and computer operation as well as mechanical equipment heating to overcome heating losses. The energy consumed to provide the high lighting levels and the office equipment and computer operation almost always exceeded heat losses from the structure during occupied periods. Mechanical cooling reduced this excess energy significantly although a considerable amount of energy was stored within the building structure during occupied periods, then released later during an unoccupied period. The calculated heat load plot calculates energy usage to overcome heat losses only. In calculating the heating load the BEAP program also maintained a constant indoor space temperature of 75°F (24°C) while the indoor temperature of the actual building varied around this setpoint.

Since the high internal loads resulted in more energy being supplied than was needed to overcome heat losses, this subsection shows evidence that the internal thermal storage during occupied day periods reduced

energy consumption of corresponding night period as expected. Thus, these figures show the relationship between energy usage and heating load. Although not expected to normally balance at hourly intervals, for longer periods of time the two values should be similar.

Again, the net energy consumption was defined as the total energy delivered to the building minus that removed by cooling equipment. The large spikes during weekdays represent electrical power consumption. These peaks tend to fall off in the afternoon, not from reductions in total energy usage, but as a result of increased cooling by the mechanical equipment. So, while the total building energy consumption remained fairly constant in the afternoon, increased cooling resulted in decreased net energy consumption. The jagged configurations during nights and weekends were a result of the cyclic nature of mechanical equipment heating.

Examination of the calculated energy load plots reveal the envelope loads without ventilation and infiltration varied sinusoidally, with peaks around 6:00 a.m. and troughs around 3:00 p.m. As expected, these peaks and troughs correspond to minimum and maximum outdoor temperatures and to the effect of solar radiation. The total heat loss was more erratic. The combined effect of changing wind speeds, the ventilation fan starting at 7:30 a.m. and shutting off at 5:30 p.m., and the varying outdoor air temperature were responsible for this behavior.

Figure 60 begins this series of figures with the plots of net energy usage and calculated heating load for the week of December 15, 1975. This particular figure begins at 6:00 a.m. on Wednesday, December 17, with the beginning of the field test period. Wednesday, Thursday, and Friday were occupied days with high rates of internal energy consumption and with the ventilation fan operating from 7:30 a.m. to 5:30 p.m. The net energy usage figure shows the effect of the ventilation fan starting in the morning, together with occupants arriving and turning on lights and office equipment. On each of these three days large spikes occurred at 7:00 a.m. and continued for several hours until the mechanical equipment heating cycled off and cooling operation increased. A trough is found at 12:00 a.m., representing the reduction in energy use during the lunch hour. At 6:00 p.m. energy consumption dropped very steeply as internal energy use was reduced as the occupants leave and the ventilation fan cycled off. The same trend with the ventilation fan was observed with the calculated heating load. At 7:00 a.m., Wednesday, Thursday, and Friday, a significant increase in the heating load was observed as the ventilation fan began operation for the day. The heating load then decreased in a smooth manner until about 3:00 p.m. when the load began to increase. At 6:00 p.m., the heating load dropped as the ventilation fan shut off. A change in the profile of the curve was next noted at 1:00 a.m. when the wind speed changed for the next day. At 7:00 a.m. the cycle starts to repeat itself as the ventilation fan again started its daily cycle. Saturday and Sunday were cold and cloudy so the load stayed fairly constant throughout this period.

Figure 60. Comparison of Net Energy Usage with the Calculated Heating Load, Week of December 15, 1975

The week of December 22, 1975 (Figure 61) was occupied Monday, Tuesday, and Wednesday as shown by the high daytime energy consumption. Thursday and Friday were Christmas holidays and, therefore, did not have the high daytime energy consumption rates and were, therefore, similar to Saturday and Sunday with the exception that the ventilation fan operated. The erratic behavior of the net energy consumption curve is the result of mechanical equipment cycling. The general shape of this curve, however, is similar to that of the calculated load during unoccupied periods.

The highest energy usage was in the early morning since heat pump heating reached its maximum in early morning. As occupants arrived, electrical consumption increased creating an even higher spike. The first drop, within several hours, represented the heating equipment cycling off. The subsequent drops represent cooling equipment operation increases. By 6:00 p.m. the electrical power consumption dropped and heating equipment operation began. Wednesday night shows increased heating equipment operation; and since the rest of the week the building was unoccupied due to the Christmas holiday, the lowest net energy usage shifts to daytime periods as expected.

The wind high velocity, limited solar radiation, and varying temperatures were responsible for the irregular behavior of the calculated heating load. The actual minimum energy usage on Saturday agreed well with that predicted by the BEAP program.

Figure 61. Comparison of Net Energy Usage with the Calculated Heating Load, Week of December 22, 1975

The week of December 29 (Figure 62) was the coldest week of the heating season. Again, Monday, Tuesday, and Wednesday were occupied; Thursday through Sunday were not. The peaks and troughs during the unoccupied days corresponded well. The ventilation fan operation was particularly apparent Friday morning. The very low outdoor temperatures resulted in increased mechanical equipment operation of over 50,000 Btu/hr (14.6 kW) from 6:00 a.m. to 7:00 a.m. Ignoring the cyclic effect of the net energy consumption, the shape of the two plots are very similar although actual energy consumption was much greater than that calculated for Saturday and Sunday.

The week of January 5, 1976 (Figure 63) represents the first five-day work week examined; even Saturday saw partial building use. High rates of electrical energy usage are shown for Monday through Friday with a lesser amount on Saturday. The effect of building energy storage was again illustrated by actual energy usage lowest in the evenings when the calculated load was actually increasing. On Saturday afternoon, the calculated thermal load dropped to approximately 30,000 Btu/hr (8.8 kW), while the net building energy usage fell to a negative value with increased mechanical cooling and previously stored internal energy being released.

For the next three weeks of January (Figures 64, 65, and 66), the net energy consumption and calculated load followed very similar trends. Early weekday morning energy use, as a result of the ventilation fan starting to operate and occupants turning on lights and office equipment, continued to be very high although the duration was reduced as

Figure 62. Comparison of Net Energy Usage with the Calculated Heating Load, Week of December 29, 1975

Figure 63. Comparison of Net Energy Usage with the Calculated Heating Load, Week of January 5, 1976

Figure 64. Comparison of Net Energy Usage with the Calculated Heating Load for the Week of January 12, 1976

Figure 65. Comparison of Net Energy Usage with the Calculated Heating Load, Week of January 19, 1976

Net Energy Consumption for the Solar Building

Calculated Heating Load of the Solar Building

Figure 66. Comparison of Net Energy Usage with the Calculated Heating Load, Week of January 26, 1976

the heating equipment cycled off earlier in the morning. Changes in the weather were very apparent in both plots with agreement in shapes and trends especially on weekends being excellent.

By the week of February 2 (Figure 67) mechanical equipment heating had become very cyclic with larger amounts of net energy usage being the result of lighting and office equipment operation. The net energy usage and calculated load opposed each other more often as large amounts of energy use and associated thermal storage reduced nighttime mechanical equipment heating operation. Generally, only during unoccupied periods were the shapes of the two plots similar.

During the remainder of the examination period as weather became milder, the net energy consumption continued to be largest during occupied periods and then fell off steeply at night (Figures 68, 69, and 70). Generally, by 10:00 a.m. the cooling equipment had begun to operate, reducing this net energy consumption; agreement between the two plots then increased.

Hourly comparisons between the net energy consumption and calculated heating load were considered very good. The BEAP program used a constant indoor temperature of 75°F (24°C), which disallowed the internal storage of energy and calculated a heating load based on energy losses from the structure without regard to cyclic building thermal storage. The average building temperature generally varied approximately five degrees through a normal day cycle. Considering the resulting storage and release

Figure 67. Comparison of Net Energy Usage with the Calculated Heating Load, Week of February 2, 1976

Figure 68. Comparison of Net Energy Usage with the Calculated Heating Load, Week of February 9, 1976

Figure 69. Comparison of Net Energy Usage with the Calculated Heating Load, Week of February 16, 1976

Figure 70. Comparison of Net Energy Usage with the Calculated Heating Load, Week of February 23, 1976

of internally stored energy and the high internal electrical energy usage, the calculated load correlated well with actual energy use.

4.6.2 Hourly Comparisons of Mechanical Equipment Heating and the Calculated Heating Load

This subsection compares the hourly values of heat pump and direct heating to the calculated heating load for the week of December 22, 1975. Figure 71 is arranged similar to those of the preceding subsection. Only one week is presented since this information may be easily extrapolated to the other weeks examined. This particular week was chosen since it represented a week with low temperatures with several occupied weekdays, several unoccupied weekdays, and a weekend period.

Monday, Tuesday, and Wednesday were occupied. The high rate of internal electrical energy usage reduced the mechanical equipment operation substantially. For two hours on Wednesday the heating equipment did not operate at all. Heating operation Monday and Tuesday nights was very low although the calculated heating load was substantial. This shows that the internally stored energy being released during night periods following occupied days substantially reduced mechanical equipment operation.

Wednesday night, around 10:00 p.m., operation of the heating equipment began. Until this time, internally stored energy release entirely maintained the building's temperature. Heating operation then continued until 12:00 p.m. Thursday. Operation of the heating system was

Figure 71. Comparison of Heat Pump and Direct Heating with the Calculated Heating Load, Week of December 22, 1975

very cyclic with one hour at a high rate followed by one hour at a reduced rate. It was this type of operation which was responsible for the short spikes often apparent in the net energy consumption plots. After 12:00 p.m., the mechanical equipment heating operation increased gradually. In the previous days this operation was not necessary since the high electrical energy usage resulted in more heat dissipation than was necessary to overcome thermal losses. Again around 10:00 p.m. mechanical equipment heating operation began. Notice that more mechanical equipment heating was necessary although the calculated loads were very similar for the two days. This was due to excess thermally stored energy in the building becoming depleted. The profile of Friday night through Saturday at 12:00 p.m. is similar. The building required mechanical equipment heating operation Saturday afternoon. The weekend being cold and cloudy resulted in mechanical equipment operation during every hour.

This figure shows little relationship between mechanical equipment heating and the indoor-outdoor temperature during the occupied days. Referring back to Figure 61, the shape of the net energy consumption plot and the shape of the heat pump and direct heating plot are very similar during unoccupied periods although they share little agreement during the occupied periods. Mechanical equipment heat pump and direct heating supply the majority of the heating load only during unoccupied periods.

4.6.3 Daily Comparisons of Net Energy Consumption and Calculated Heating Load

This comparison, unlike the preceding two subsections, presents 24-hour hourly average calculated heating loads and net energy usage. In addition, Figures 72, 73, and 74, which accompany this subsection, include the daily average indoor and outdoor temperature with the resulting temperature differential being shaded. These temperatures are shown in an upper plot. Calculated heating load and net energy usage are shown together on the lower plot to provide easy comparison of their values. The area between the calculated heating loads and the axis is shaded to provide clarity to the figure. The area between a second curve and the axis represents the net energy usage. The net energy usage curve has a clear area between it and the calculated heating load when its value exceeded the calculated load and is shaded on both sides when its value was below the calculated load.

These three figures represent a 75-day period from December 17, 1975, through February 29, 1976. Each individual figure contains a month's information. Some of the daily discrepancies are attributed to building thermal storage. Building use and weather changes may have resulted in a net gain or loss of internal energy storage from one day to the next. A second area of disagreement may have been fluctuations in indoor temperatures increasing and decreasing energy losses. The indoor temperature swing may also have resulted in increased losses as compared to average temperature similar to those observed for fluctuations in outdoor temperature.

Figure 72. Comparison of Net Energy Usage with Calculated Heating Load with Corresponding Indoor and Outdoor Temperatures, December 1975

Figure 73. Comparison of Net Energy Usage with Calculated Heating Load with Corresponding Indoor and Outdoor Temperatures, January 1976

Figure 74. Comparison of Net Energy Usage with Calculated Heating Load with Corresponding Indoor and Outdoor Temperatures, February 1976

The largest area of possible disagreement between actual and calculated energies was probably caused by calculated rates of infiltration and ventilation not precisely matching those which actually occurred. The opening of exterior doors as a result of traffic into and out of the building, for example, would increase the infiltration load beyond that calculated. Also, while infiltration was based on wind speed, wind direction was always assumed to be from the predominate direction, northwest. And finally, the wind speeds used in the BEAP program calculations were daily average wind speeds and not instantaneous wind speeds.

Another area of possible disagreement could have resulted from wall, roof, and window conductances used to predict conduction being idealized values. A last area of error was associated with collecting the actual data. Discrepancies between actual energy flows and those recorded were limited to the error tolerances of the measuring devices. Data collection equipment failures were also suspect for discrepancies in the beginning of January.

Considering these above areas of possible error and also that the comparison was made between a calculated building heating load and net building energy usage, long-term agreement was excellent. When viewed over the entire 75-day period, the calculated load was 89.8% of the actual net energy consumption. Observations of the figures show that, on some days, the actual net energy usage and calculated heating load agree completely.

Throughout the month of December the net energy usage and the calculated heating load matched very closely. The monthly averages of actual and calculated loads were 69 MBtu/hr (20.2 kW) and 69.5 MBtu/hr (20.4 kW), or within 1%.

The first week of January, the coldest week of the year, had an average temperature around 20°F (-7°C) for the first several days of the month. These first days were unoccupied, and the actual and calculated load matched exactly. Then for the next three days, the net energy consumption went to abnormally high values. This increase in net energy consumption occurred as the outdoor temperature rose and the temperature differential decreased. This problem in the data was never identified although probably resulted from some type of data-recording failure. A second possible explanation could have been abnormally high rates of infiltration except that the wind velocity was not extreme; and even if the ventilation fan was left on 24 hours a day during this problem period, the resulting heating load would not have been as high as was recorded. For the remainder of the month of January the actual and calculated loads followed the same trends with reasonable agreement. As a result of those three days early in the month, the monthly average hourly rates of net energy usage and calculated heating load had the largest discrepancy of the three-month period. The resulting hourly averages were 77 MBtu/hr (22.6 kW) and 65 MBtu/hr (19.0 kW), respectively. The percent agreement for the month was 84.4%.

During the month of February, the average hourly net energy usage was 53 MBtu/hr (15.5 kW) while the average hourly calculated heating load was 48 MBtu/hr (14.1 kW). This may easily be due to excess daytime heat being dissipated by more frequent opening of exterior doors as daytime temperatures rose. On the days between February 14 and 21, high wind velocities recorded at the airport account for the spikes in the calculated heating load calculations.

5.0 CONCLUSIONS AND RECOMMENDATIONS

5.1 Conclusions

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 heating energy which had to be supplied by mechanical equipment.

After the indoor space temperature rose through the dead band, 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 efficiently rejected to water storage to maintain proper occupied space temperatures and efficiently 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. In addition, the water-to-water heat pump operated very efficiently with an average coefficient of performance of 4.2 (8).

Throughout the winter heating season most of the energy being 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 years, 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 requirement 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 Environmental 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.

Table 6 summarizes actual energy usage and calculated loads on a monthly basis for the 1975-76 heating season. Both the steady-state and transient loads are shown. Both the calculated and measured loads were approached separately with no adjustment of either the net energy usage or calculated heating load to bring the two rates into better agreement. It is believed this method will provide more insight into predicting thermal loads for other structures. For the 75-day period examined, the steady-state load predicted a load which was 93.4% of that measured. The transient load calculations predicted a load 89.8% of that measured. Considering experimental error, additional losses due to inefficiencies, and additional infiltration which could not be measured, these results are considered excellent. This study provides strong evidence that the BEAP program is responsive and accurate in predicting heating and cooling loads on a building structure.

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,

Table 6. Summary of Monthly and Seasonal Net Energy Usage and Calculated Loads

Period	Net Energy Usage ¹ 10 ³ Btu/hr	Calculated Load ²		Calculated Load ÷ Net Energy Usage	
		Steady-State 10 ³ Btu/hr	Transient 10 ³ Btu/hr	Steady-State %	Transient %
December (15 days)	69.0	71.3	69.5	103.3	100.7
January (31 days)	77.0	71.3	65.0	92.6	84.4
February (29 days)	53.0	46.3	48.0	87.4	90.6
Dec. - Feb. (75 days)	66.0	61.6	59.3	93.4	89.8

1 Hourly average measured rate of energy supplied to building.

2 Hourly average calculated rate of energy supplied to building using the Building Environmental Analysis Program. Steady-state load calculated using average weather conditions for period indicated. Transient load calculations use actual hourly weather data.

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 predict infiltration rates, which included the measurement of infiltration using tracer gas techniques, variations in building use, indoor-outdoor temperature differentials, wind speed, and wind direction did affect its magnitude. 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 several miles apart. 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 (24°C), 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 Temperature</u>	
Day (occupied)	17°F	(-8°C)
Day (unoccupied)	47°F	(8°C)
Night (unoccupied)	52°F	(11°C)
Daily (24 hour with occupied day)	38°F	(1°C)
Daily (24 hour with unoccupied day)	50°F	(10°C)

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 with nights below freezing and typical daytime temperatures between 40°F and 50°F (4°C to 10°C). 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 efficient 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.

5.2 Recommendations

The lighting system was responsible for most 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 for heat pump heating and 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 is 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 all climatic regions of the United States.

APPENDIX A: BUILDING CONSTRUCTION INFORMATION

The following tables describe the exterior wall construction. Wall and roof types are described with the exterior area and orientation in Table 7. Individual wall and roof assemblies are described in Table 8. Exterior wall types and surface areas are shown for each individual office in Table 9.

Table 7. Solar Building Exterior Wall Construction

Surface (Wall or Roof)	Assembly	Slope	Orientation	Assembly Area	Surface Area
North	BRWL	90	0	43	1138
	STWL	90	0	770	
	MULL	90	0	70	
	SNGL	90	0	254	
North-Sloped	ROOF	16	0	2501	2561
	SKLT	16	0	60	
East	BRWL	90	90	399	450
	MTDR	90	90	20	
	BLND	90	90	31	
South	STWL	90	180	223	243
	MTDR	90	180	20	
South-Sloped	BLND	66	180	28	778
	COLL	66	180	750	
West	STWL	90	270	185	286
	BRWL	90	270	90	
	MTDR	90	270	20	
	SNGL	90	270	11	
Flat	ROOF	0	0	1695	1716
	SKLT	00	0	21	

Table 8. Solar Building Wall and Roof Assemblies

Roof (ROOF)

Material	Thickness (in.)	Total Resistance °F hr ft ² /Btu	Density (lb/ft. ³)	Specific Heat (lb °F)
Built-up	0.375	0.33	70.0	0.35
Plywood	0.625	1.28	34.0	0.29
Air Space	4.000	0.85	0.075	0.24
Insulation	4.000	11.00	4.0	0.18
Gyp. Bd.	0.500	0.43	50.0	0.26
Outside Air		0.17		
Inside Air		0.61		

Total U-Value = 0.068 at 15 mph wind

Outside Absorption to Solar 0.7

Brick Wall (BRWL)

Material	Thickness (in.)	Total Resistance °F hr ft ² /Btu	Density (lb/ft. ³)	Specific Heat (lb °F)
Face Brick	4.0	.44	130.0	0.20
Air Space	1.0	0.96	0.075	0.24
Com. Brick	4.0	.80	120.0	0.20
Gyp. Bd.	0.5	0.43	50.0	0.26
Outside Air		0.17		
Inside Air		0.68		

Total U-Value = 0.29 at 15 mph wind

Outside Absorption to Solar 0.8

Table 8. Solar Building Wall and Roof Assemblies (continued)

Stucco Wall (STWL)

Material	Thickness (in.)	Total Resistance °F hr ft ² /Btu	Density (lb/ft. ³)	Specific Heat (lb °F)
Outside Air		0.17		
Stucco	1.0	0.20	116.0	0.18
Sheating	0.5	1.32	18.0	0.31
Insulation	4.0	11.00	4.0	0.18
Gyp. Bd.	0.5	0.45	50.0	0.26
Inside Air		0.68		

U-Value = 0.072 at 15 mph wind

Outside Absorption to Solar 0.5

Metal Door (MTDR)

Material	Thickness (in.)	Total Resistance °F hr ft ² /Btu	Density (lb/ft. ³)	Specific Heat (lb °F)
Outside Air		0.17		
Metal Door	1.75	0.91	2.0	0.17
Inside Air		0.68		

U-Value = 0.57 at 15 mph wind

Outside Absorption to Solar 0.5

Mullions (MULL)

Material	Thickness (in.)	Total Resistance °F hr ft ² /Btu	Density (lb/ft. ³)	Specific Heat (lb °F)
Outside Air		0.17		
Wood	4.5	4.35	32.0	0.33
Inside Air		0.68		

U-Value = 0.19 at 15 mph wind

Outside Absorption to Solar 0.5

Table 9. Solar Building Room Construction

Space	Assembly	Surface	Area
Office #1	STWL	North	66.0
	SNGL	North	22.5
	ROOF	N-Slope	180.0
	MULL	North	5.5
Office #2	STWL	North	66.0
	SNGL	North	21.7
	ROOF	N-Slope	180.0
	MULL	North	6.3
Office #3	STWL	North	70.0
	SNGL	North	21.7
	ROOF	N-Slope	134.0
	MULL	North	6.3
Office #4	STWL	North	66.5
	SNGL	North	22.5
	BRWL	East	99.0
	ROOF	N-Slope	173.0
	MULL	North	5.5
Office #5	ROOF	Flat	54.0
	ROOF	N-Slope	36.0
Office #6	ROOF	Flat	54.0
	ROOF	N-Slope	36.0
Office #7	ROOF	Flat	118.0
Office #8	ROOF	Flat	242.0
	SKLT	Flat	7.0
Drafting Room (extension)	STWL	North	171.0
	SNGL	North	55.0
	STWL	West	165.0
	STWL	South	223.0
	MTDR	South	20.0
	ROOF	Flat	651.0
	MULL	North	18.0

Table 9. Solar Building Room Construction (continued)

Space	Assembly	Surface	Area
Drafting Room	STWL	North	231.0
	SNGL	North	77.5
	BRWL	West	90.0
	ROOF	N-Slope	987.0
	SKLT	N-Slope	21.3
	MULL	North	20.4
	SOLAR	S-Slope	788.0
Computer Room	STWL	North	100.0
	SNGL	North	33.7
	ROOF	N-Slope	425.0
	SKLT	N-Slope	10.7
	MULL	North	8.7
Conference Room	BRWL	East	300.0
	ROOF	N-Slope	138.0
	SKLT	N-Slope	10.7
	BLND	S-Slope	27.5
	BRWL	North	43.0
Receptionist	ROOF	N-Slope	30.0
	ROOF	Flat	350.0
	SKLT	Flat	7.0
	BLND	East	31.0
	MTDR	East	20.0
Print Room	ROOF	Flat	167.0
Corridor	ROOF	N-Slope	182.0
	SKLT	N-Slope	10.7
Hall	ROOF	Flat	59.0
	SKLT	Flat	7.0
	MTDR	West	20.0
	SNGL	West	11.6

APPENDIX B: INSTRUMENTATION, DATA ACQUISITION, AND DATA VALIDATION

B-1 Instrumentation

The building instrumentation included both automatic and manual measurement devices. Those devices which were considered automatic used sensing devices coupled with the building's IBM System 7 data acquisition computer. These measurements were recorded at six-minute intervals. The manual measurements were those which relied on personnel making visual or manual observations at specific times.

During the first data acquisition period, a total of 72 data points were recorded automatically for each six-minute monitoring period. These represented 40 analog inputs and 32 digital inputs. Analog inputs recorded were: date, time, mechanical equipment water temperatures, ambient and indoor temperatures, solar insolation, wind velocity and direction, and several reference points. The digital inputs consisted of: electrical readings, operational modes, and door-opening sensors. For the second heating season the total number of data points increased to 165. A large number of the additional points resulted from temperature probes inserted into each of the three tanks. These storage tank probes formed an array at different depths and were used to measure for possible stratification (9, 10).

All continuously monitored air and water temperatures were made with high-precision thermistors which were composite pairs encapsulated in epoxy to form a probe. The linearized thermistor pairs had a maximum

tolerance of $\pm 0.66^{\circ}\text{F}$ (0.37°C), but were selected in pairs which agreed within 0.33°F (0.18°C). Attempts were made to maintain this increased accuracy by checking data output and by periodically testing thermistor pairs utilizing a constant temperature bath. The water flow thermistors were placed in wells protruding at least two inches into the flow stream and were used to determine inlet and outlet temperatures to and from various parts of the mechanical equipment. These thermal wells, at the inlets and outlets of each piece of mechanical equipment, made possible the calculation of energy flows and energy balances throughout the system. The locations of measuring devices are shown schematically in Figure 4. Indoor temperature sensors recorded building temperatures at five locations. The one outdoor temperature sensor was placed in the weather station located on the roof of the building. In addition, two dew point sensors, one located in the drafting room and one in the weather station, measured indoor and outdoor dew points. Air temperature sensors were located in the airstreams of the air-handling unit and in each of the water-to-air units. Due to variations in air velocity and temperature gradients in these streams, the recordings were used qualitatively only.

Incident solar fluxes were recorded by means of three pyranometers located on the building's roof. Two were oriented horizontally, of which one measured total horizontal insolation and the other covered by a shadow band received only diffuse radiation on that horizontal surface. The third pyranometer oriented 55° (.96 radians) from the horizontal measured total insolation incident on the collector surface.

Digital input data were recorded by digital counters. Electrical energy consumption was recorded by an electrical impulse produced by the revolving wheel on the watt-hour meters. These electrical readings were made on 19 watt-hour meters sensing: total three-phase power, total single-phase power, power for each heat pump compressor and fan, power for each water pump, the evaporative water cooler power, and power for other equipment such as computers, which produced high internal loads. Exterior doors and some valves were equipped with positional sensors. The position of the valves indicated water flow configurations used to predict mode of operation of the water-to-water heat pump. The door sensors indicated additional ventilation to the building when open. The data acquisition system was programmed to sense a positioned device in each watt-hour meter, once each second, in order to detect their positions.

The manual measurements included water and airflow rates on mechanical equipment, and ventilation and infiltration rates.

Water flow rates were measured through the water-to-water heat pump evaporator, the condenser, the air-handling unit, the water side of the solar loop heat exchanger, the evaporative water cooler, each of the five water-to-air heat pumps, and the water-to-air loop. A flow meter, accurate to within 1/2 gal/min ($0.00003 \text{ m}^3/\text{s}$), was used to record all rates except that of the water-to-air heat pump for which an integrating type meter was used. Visual observation was employed; and water flow rates were found to be nearly constant, although minor fluctuations occurred even during steady-state operation.

Airflows were measured using a pitot tube. Measurements were obtained for airflows through the ventilation fan, through each of the water-to-air heat pump units and through the main water-to-water heat pump. This method produced acceptable results with only the ventilation air fan. Monitoring of airflow rates through mechanical equipment produced serious discrepancies. Due to this lack in uniformity, these observations of flows and temperatures were used only to validate operation of mechanical equipment and not to evaluate energy flows.

The building's air change rate was measured using a tracer gas technique (6). The tests were performed for both occupied and unoccupied periods. Occupied periods were those where the building received positive ventilation from the fresh air fan. Only infiltration occurred during unoccupied periods. The results were obtained during the summer with the temperature differences and wind forces minimal. For this reason, expected trends in infiltration rates due to wind velocities and large temperature gradients had to be extrapolated.

B-2 Data Acquisition

Data acquisition was performed by the IBM System 7 computer at the building site. The system is an interrupt driven computing system designed to collect data directly from instrumentation. The System 7 was equipped with 12,000 words of memory, 64 analog input channels, and 32 digital input channels. A binary synchronous communication mode permitted the IBM System 7 to communicate with the IBM 370/168 at The

Pennsylvania State University Computation Center. The IBM System 7 was chosen as the data acquisition system since it had the ability to read both analog and digital inputs, it was compatible with the IBM 370/168, and three IBM field developed programs for data collection were available which suited the needs of the project (10).

In addition to the three readily available programs, a fourth program was written by the departmental staff. This data acquisition program was designed to fit the particular needs of the project (11).

B-3 Data Validation

There were several procedures developed to validate data collection and its processing. These were: on-site inspection and testing, the review of raw collected data in both six-minute and hourly form, and redundancy checks of energy flows and temperature differentials made possible through the use of computational programs.

Regular on-site observations were made during system operation to check system functions, system response, and instrumentation. The on-site inspections involved visual inspections of the system which could detect gross malfunctions. This phase was most important during periods in which instrumentation and data acquisition would undergo changes such as the beginning of the collection periods. Certain checks were also performed to validate the fact that the instrumentation would respond to system changes. Finally, site monitoring was used. Since the System 7 had no provision for on-site monitoring,

information stored first had to be sent to The Pennsylvania State University for conversion to engineering units. A suspected problem would require days or perhaps a week before data could be reviewed. To increase the on-site capabilities and to reduce this lag time, a Kaye Instrument Model 8000 data logger was borrowed from the University of New Mexico, Department of Mechanical Engineering (10, 12). This instrument had the ability to sample instrumentation at regular intervals and could output measurements on a paper tape. This device was used to check temperature thermistors and power-recording instruments on a regular basis. While primarily used for validation and diagnostic analysis, it proved very worthwhile in analysis system component transients by outputting instantaneous data.

Once the automatically recorded data from the IBM System 7 monitoring unit was shipped to The Pennsylvania State University, printed data were obtained. The data were printed initially in hour averages, providing an overview of the performance. If for some reason data problems were suspected, the data were then printed at six-minute intervals. The six-minute and hourly averages provided the basis for data validation and review throughout the research. Hourly data were also sent to the staff in Albuquerque for their review; six-minute data were supplied to them upon request. As the project proceeded, analyses and reduction programs were implemented. The form in which the data was displayed was such that problems in the data would be as readily recognizable as possible.

Several methods were developed for examining the data for consistency and accuracy. One of the most widely used procedures was that of redundancy checks. During steady-state operation in a particular mode, sensors located at inlets and outlets of components could be checked. For example, during heat pump heating, the tank leaving water temperature sensed at the tank outlet could be compared with the evaporator inlet. Since the two components were joined directly, the temperature of water at the two points should be within tolerances as the system approached steady state, assuming negligible heat transfer from the pipe to surroundings. The redundancy water temperature checks for the water-to-water heat pump were as follows:

For heat pump heating:

- Tank 1 leaving and evaporator entering.
- Condenser leaving and air-handling unit entering.
- Air-handling unit leaving and condenser entering.

For heat pump cooling:

- Tank 1 leaving and condenser entering.
- Evaporator leaving and air-handling unit entering.
- Air-handling unit leaving and evaporator entering.

For direct heating:

- 1974-75 Tank 1 leaving and air-handling unit entering.
- 1975-76 Tanks 2 and 3 leaving and air handling unit entering.

For dead band:

- Evaporator leaving and air-handling unit entering.
- Air-handling unit entering and air-handling unit leaving.
- Air-handling unit leaving and evaporator entering.
- Evaporator entering and evaporator leaving.
- Tank 1 leaving and condenser entering.

A second method, one which utilized related rates and temperature differences, compared corresponding thermal energy flows through the use of conservation of energy principles. The energy flows for equipment were calculated using measured temperatures, flow rates, and electrical energy. This procedure was used for the water-to-water heat pump and the water-to-air heat pump loop.

For the water-to-water heat pump several energy balances could be performed for both heating and cooling. The water flow between the condenser and the air-handling unit formed a closed loop on heating. Energy production in the condenser should have, therefore, equalled energy dissipation at the air-handling unit for steady-state operation. The energy withdrawn from the water at the chiller plus the compressor electrical power should also have agreed with energy produced at the condenser side of the heat pump. Hourly values of these energy flow rates have been plotted (12). The first plot, Figure 75, compares condenser heat output to the input into the air-handling unit. The second plot, Figure 76, compares the chiller heat absorbed plus compressor power to the condenser heat output. The plots represent the heat pump operation over a 40-day period in 1976. The principal reason for the point scatter was the transient behavior of the system. Three factors contributed to this transient behavior. The first was mode switching which exposed components to different temperatures and water flow rates. The second was the cyclic nature of the heat pump compressor when exposed to the elevated tank temperatures present in the system. A third factor on the chiller-condenser plot caused most points to lie above the line. This discrepancy was due to the assump-

Figure 75. Water-to-Water Heat Pump Heating Condenser vs. Air-Handling Unit Energy Balance (11)

Figure 76. Water-to-Water Heat Pump Heating Chiller vs. Condenser Energy Balance (11)

tion that all compressor electrical energy was dissipated to the flow stream. In reality, some of this energy dissipated as heat was conducted to the surrounding from the compressor.

For periods when the water-to-water heat pump was in the cooling mode, similar energy balances were performed. The evaporator and the air-handling unit were now joined to form a closed loop and, therefore, should have had equal exchange rates. The heat exchange rate from the condenser should also have equalled the heat exchange rate from the evaporator after allowing for the heat dissipated by the compressor. The energy balances between the chiller heat absorbed and the air-handling unit input were generally in good agreement. The energy balance between the condenser heat output minus compressor power and the chiller heat absorbed has considerable scatter. This scatter is attributed to transients within the system as the mode switches to and from heat pump cooling. Figures 77 and 78 show the energy balances recorded as hourly averages for the chiller vs. air-handling unit and condenser vs. the chiller, respectively.

Energy balances were made on the water-to-air heat pump loop by comparison of the thermal energy output of the entire loop to the sum of the outputs from each of the individual water-to-air heat pump within the loop. The total thermal energy for the entire loop was calculated using the temperature difference across the loop and the water flow rate through the loop. The individual units were summed using the water flow rates through and temperature differences across each water-to-air unit. Since the temperature difference across the loop

Figure 77. Water-to-Water Heat Pump Cooling Chiller vs. Air-Handling Unit Energy Balance (11)

Figure 78. Water-to-water Heat Pump Cooling Condenser vs. Chiller Energy Balance (11)

was usually three degrees or less, the best expected accuracy of this loop was $\pm 13\%$. Figure 79 shows a plot of the energy balance. Deviations from the line indicate the effect of transients and measurement errors. Although the deviation is significant, it was considered acceptable considering the small temperature difference across the loop.

Examination of the 7 1/2 ton water chiller comprised the largest single area of component examination. To predict performance and operating characteristics, a sub-program was developed to determine the mode in which the heat pump was operating. Since the heat pump mode did not change with the start and end of a six-minute period, mode switching occurred at random from the standpoint of building data acquisition. Programs, therefore, must not only have had the ability to choose the correct mode when the system was operating within a mode for an entire sample period, but must also have had the ability to choose the better mode for those sampling periods that contain transitions from one mode to the next.

Tables 10 and 11 show the best expected accuracy for the water-to-water heat pump and water-to-air heat pump loop. It must be remembered that these numbers represent an optimum for both years. The numbers are based on steady-state operation of systems with all instruments within the tolerance noted.

Figure 79. Water-to-Air Heat Pump Loop Energy Balance (11)

Table 10. Mechanical Equipment Flow Rates, Average Temperature Differentials, and Steady-State Error for the 1974-75 Winter Heating Season

		Flow Rate			Temp.		Heat Flow	Error
		gpm	m	%	T	%	Q	%
Heat Pump Heating	Air-Handling Unit	20.8 $\pm$ 0.5	10410 $\pm$ 250	2.4	10.0 $\pm$ 0.33	3.3	104,100 $\pm$ 5,935	5.7
	Condenser	20.8 $\pm$ 0.5	10410 $\pm$ 250	2.4	10.0 $\pm$ 0.33	3.3	104,100 $\pm$ 5,935	5.7
	Chiller	22.60	11311 $\pm$ 250	2.2	7.0 $\pm$ 0.33	4.7	79,117 $\pm$ 5,483	6.9
Heat Pump Cooling	Air-Handling Unit	23.50	11762 $\pm$ 250	2.1	6.7 $\pm$ 0.33	4.9	78,805 $\pm$ 5,556	7.1
	Condenser	18.60	9309 $\pm$ 250	2.7	11.0 $\pm$ 0.33	3.0	102,400 $\pm$ 5,822	5.7
	Chiller	23.50	11762 $\pm$ 250	2.1	6.7 $\pm$ 0.33	4.9	78,805 $\pm$ 5,556	7.1
Direct Heat	Air-Handling Unit	16.0 (15.70)	8008 $\pm$ 250	3.1	5.0 $\pm$ 0.33	6.6	40,040 $\pm$ 3,893	9.7
Loop	Loop	17.50	8759 $\pm$ 250	2.9	3.0 $\pm$ 0.33	11.0	26,277 $\pm$ 3,640	13.9

Table 11. Mechanical Equipment Flow Rates, Average Temperature Differentials, and Steady-State Error for the 1975-76 Winter Heating Season

		Flow Rate			Temp.		Heat Flow	Error
		gpm	m	%	T	%	Q	%
Heat Pump Heating	Air-Handling Unit	20.40 $\pm$ 0.5	10210 $\pm$ 250	2.4	10.0 $\pm$ 0.33	3.3	102,100 $\pm$ 5,869	5.7
	Condenser	20.40 $\pm$ 0.5	10210 $\pm$ 250	2.4	10.0 $\pm$ 0.33	3.3	102,100 $\pm$ 5,869	5.7
	Chiller	18.50 $\pm$ 0.5	9259 $\pm$ 250	2.7	8.3 $\pm$ 0.33	4.0	76,850 $\pm$ 5,130	6.7
Heat Pump Cooling	Air-Handling Unit	23.30 $\pm$ 0.5	11662 $\pm$ 250	2.1	6.5 $\pm$ 0.33	5.1	75,803 $\pm$ 5,473	7.2
	Condenser	18.20 $\pm$ 0.5	9109 $\pm$ 250	2.7	11.0 $\pm$ 0.33	3.0	100,200 $\pm$ 5,756	5.7
	Chiller	23.30 $\pm$ 0.5	11662 $\pm$ 250	2.1	6.5 $\pm$ 0.33	5.1	75,803 $\pm$ 5,473	7.2
Direct Heat	Air-Handling Unit	23.30 $\pm$ 0.5	11161 $\pm$ 250	2.2	4.0 $\pm$ 0.33	8.3	44,644 $\pm$ 4,683	10.5
Loop	Loop	21.54 $\pm$ 0.5	10781 $\pm$ 250	2.3	3.0 $\pm$ 0.33	11.0	32,343 $\pm$ 4,308	13.3

APPENDIX C: DATA REDUCTION AND EVALUATION

C-1 Data Reduction Computer Programs

Much of this research is computer oriented. With the vast amount of data, approximately eight million data samples over the two heating seasons, representing information from 341 days, computational reduction and analysis provided a means of studying and researching data. Two large computer programs were developed and written by the project staff, one to handle the data for each winter period. The two programs were developed to be entire packages to handle all reduction and analysis programs in subroutines. Only those parts necessary for the building and equipment performance will be discussed here. Discussions are limited to program purposes, logic being discussed only where necessary. Additional information is supplied in particular sections using the sub-program. Further documentation may be found in an unpublished working paper within the Department of Architectural Engineering entitled, "Data Reduction and Evaluation Methods".

C-2 Operational Mode Determination

Three areas of complication arose in the determination of the water-to-water heat pump operational mode. First, the modes did not change as a six-minute sampling period changed. Second, the valve switching and pump changes did not always occur simultaneous, resulting in overlaps in mode sensor time, in some cases more than 40 seconds per six-minute sample. And third, the inherent transients that occur

as piping and flow rate changes lead to temperature discrepancies at the beginning of periods. A particular mode was chosen for each six-minute sampling period. No attempt was made to divide a sampling period into two modes. Programs were designed, however, to minimize discrepancies inherent in mode changing by choosing the most dominant mode for a particular sample.

Two methods were developed to select the mode for a six-minute sample. The first method (Figure 80) is based on compressor operation and air-handling unit temperature differentials. Upon investigation of data, isolated problems arose when Method 1 did not choose the best mode. Assessing the rate of failure, or even those periods where problems might arise, was very difficult. A second method of mode detection was then developed. This method involved the mode sensors as well as comparing temperatures and power consumption of the compressor. Mode detectors had been previously discounted due to overlaps and periods in which some or all had failed. Through careful manipulation of constraints, a system of logic checks was devised to overcome the problems (Figures 81 and 82). The two methods allowed validation of each other since discrepancies could be more easily identified. Comparing the outputs for discrepancies yielded fewer than previously expected with the second method being slightly better. Although variations did occur in short periods (a few sample periods or a few hours) when results were compared over the entire heating seasons, they produced essentially the same results. Both methods produce results well with the uncertainties of other areas.

Figure 80. Water-to-Water Heat Pump Mode Selection, Method 1, 1974-75 and 1975-76 Winter Heating Seasons

Switching near the beginning or end of periods, together with a typical operation, caused the problems with method 1. Much of the problems were due to transients within the system. Method 2 was found to fail during test periods when piping arrangements were changed, an example being direct heating from the underground tank during the second season. Both mode selections are incorporated into the computer program for each year. The user was able to select whichever method he desires. The second method of mode selection was used throughout this thesis.

During the first winter, that of 1974-75, the mode sensors first began to work properly, day 353, hour 2036. After that time the sensors worked properly with a few exceptions. The mode detection logic of method 2 allowed for these exceptions and was able to correctly identify periods throughout the collection period. Examination of sample periods yielded the three major modes and six transitional modes. The problem lay in making determination of mode when switching occurred. The mode determination logic normally selected the mode with the longest time increment. If there was compressor power, preference was given to heat pump operation. The relative amounts of each mode were compared, and compressor power was checked. The logic checked temperature differences to determine mode when sensors failed. Certain minimum temperature rises or drops were required for heating or cooling operation.

The dead band between direct heating and heat pump cooling may have had, on occasion, either a net heating or cooling effect over the

six-minute sample. A heating effect would have been caused by the switching from direct heating to the dead band; the cooling effect would have been caused by previous heat pump cooling. To account for this delivered energy, the air-handling unit temperature differential was checked; if positive, direct heating was assumed and if negative, heat pump cooling was assumed. Another complicated factor arose; this occurred when the thermostat called for direct heating with low direct heating tank temperatures. In cases when the storage tank was below 70°F (21°C), cooling rather than heating could occur.

After the mode was selected, a minimum temperature differential was required for calculations to proceed. A one degree drop or rise was chosen as this minimum value. Examination of data showed this to be realistic.

During the second winter, 1975-76, a fourth mode sensor was added to the data collected. This checked for heat pump operation. The resulting logic for mode determination differed slightly to reflect this addition. Time increments, compressor power, and certain temperature checks were consistent with the previous year.

C-3 Heat Pump Heating and Cooling Computer Program

This program outputs performance data for periods which are selected by the mode sensors as either heat pump heating or heat pump cooling. Each six-minute sample is analyzed individually, then arranged in hour blocks. The program contains all necessary informa-

tion to perform calculations and print results. The program was developed for several purposes. First, it provided a method for checking system performance by producing results that could be easily checked with collected data. Since output was printed in six-minute blocks, a one-to-one correspondence could be obtained for verification. The second purpose of the program was to pass the results to other sub-programs including Water-to-Water Heating Pump Coefficient of Performance, Total Building Energy, and System Performance Factor.

C-4 Direct Heating Computer Program

This program arrangement and purposes are similar to the Heat Pump Heating and Cooling program. The program performs analysis on those periods selected by the mode sensors as direct heating analysis, and verification again may be performed with a one-to-one correspondence; and calculations may then be passed to other analysis programs.

C-5 Water-to-Water Heat Pump Coefficient of Performance Computer Program

This program provides the basis of performance analysis for the 7 1/2 ton water-to-water heat pump. The analysis is broken into two sections for any run. The first contains periods of heat pump heating; the second periods of heat pump cooling. For each six-minute period in which the heat pump operated, the analysis was performed.

Operation was also summarized for periods which were selected as heat pump heating (or heat pump cooling) for 10 consecutive periods

(one hour). In addition to printing six-minute data blocks, the program also produced summaries of operation.

The output in both six-minute and hourly periods of operation included entering and leaving temperatures, temperature differentials, compressor power and energy flows, and coefficients of performance for each component; i.e., chiller, compressor, and air-handling unit.

C-6 Total Building Energy Computer Program

The Total Building Energy program accounts for all energy input into the building including electricity, output from heat pumps, occupants, and ventilation or infiltration. A net energy is summed representing the total energy supplied to the spaces minus energy removed. The indoor and outdoor temperatures are printed along with the temperature differential across the building envelope. Energy dissipated by the evaporative water cooler is included to provide a greater understanding of the system, but is not used in any calculations. Incident solar radiation is also included to compare energy use with outdoor weather conditions.

The output is printed on an hourly basis and summarized in 12-hour increments. The day period started at the first hour block of collected data after 6:00 a.m. (600 hours), and the night period started the first hour after 6:00 p.m. (1800 hours). This program provides the basis for building performance.

C-7 Building Performance Factor Computer Program

The Building Performance Factor program produces daily summaries of energy delivered by the air-handling unit by heat pump heating and direct heating. The daily periods correspond to the daily periods selected by the Total Building Energy program.

For each monitored day, the energy delivered by heat pump operation and direct heating was calculated and printed, as were Tank 1 average daily temperature, air-handling unit power, compressor power, pump power, system performance and equipment performance. In addition, a summary of the entire heating season was printed.

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