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The Integration of Water Loop Heat Pump and Building Structural Thermal Storage Systems

**T. J. Marseille
J. S. Schliesing**

October 1991

**Prepared for the U.S. Department of Energy
under Contract DE-AC06-76RLO 1830**

**Pacific Northwest Laboratory
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FOREWORD

This report is one in a series describing research activities in support of the U.S. Department of Energy (DOE) Advanced Commercial Building Technologies Program. The goal of the program is to improve energy-related decision-making during all phases of the building life cycle by considering interactions between building systems. This whole-building approach underlies the research conducted by the program.

The objectives of the Advanced Commercial Building Technologies Program are

- to understand interactions among building components and systems
- to optimize the combined performance of these systems
- to develop, in cooperations with industry, computer-based tools to facilitate energy-efficient building design and operation.

The lead laboratory for this program is the Pacific Northwest Laboratory. Expertise and resources from industry, academia, other government entities, and other DOE laboratories are used in planning, reviewing, and conducting research activities. Cooperative efforts with other organizations are an integral component of the program.

This report documents an investigation of the interactions of a water loop heat pump system and building structural mass. The analysis of their effects on whole-building energy performance is also detailed. The authors and program managers welcome any comments about the work described in this report. Other documents issued previously from the Advanced Commercial Building Technologies Program are listed in the Bibliography in Section 10 of this report.

Michael R. Brambley
Program Manager
Commercial Building Systems
Applied Physics Center
Pacific Northwest Laboratory

Mark L. Bailey
Program Manager
Office of Building Energy Research
Building Systems and Materials Division
U.S. Department of Energy

SUMMARY

Many commercial buildings need heat in one part and, at the same time, cooling in another part. Even more common is the need for heating during one part of the day and cooling during another in the same spaces. If that energy could be shifted or stored for later use, significant energy might be saved. If a building's heating and cooling subsystems could be integrated with the building's structural mass and used to collect, store, and deliver energy, the energy might be saved cost-effectively.

To explore this opportunity, researchers at the Pacific Northwest Laboratory (PNL) examined the thermal interactions between the heating, ventilating, and air-conditioning (HVAC) system and the structure of a commercial building. Computer models were developed to simulate the interactions in an existing building located in Seattle, Washington, to determine how these building subsystems could be integrated to improve energy efficiency.

The HVAC subsystems in the existing building were modeled. These subsystems consist of decentralized water-source heat pumps (WSHP) in a closed water loop, connected to cooling towers for heat rejection during cooling mode and boilers to augment heating. An initial "base case" computer model of the Seattle building, as-built, was developed. Metered data available for the building were used to calibrate this model to ensure that the analysis would provide information that closely reflected the operation of a real building.

The HVAC system and building structure were integrated in the model using the concrete floor slabs as thermal storage media. The slabs may be actively charged during off-peak periods with the chilled water in the loop and then either actively or passively discharged into the conditioned space during peak periods. In the computer model, this concept was implemented by simulating the manifolding of an additional water loop to the main HVAC hydronic loop. Piping in the loop was embedded in the concrete floor slabs of the building. Water was routed to the loop in a controlled manner to charge or discharge thermal energy to and from the slabs. This integration strategy is referred to as the Building Structural Thermal Storage (BSTS) system.

The building model with the BSTS system was simulated and the results compared to the results of the base case building. Simulations were also performed for both the base case building model and the building model with the BSTS system using weather data for five other locations in the United States. Annual HVAC energy consumption of the total building was used to measure performance. The diurnal responses of temperature and heat transfer within the building were also monitored to gain a further understanding of the operating characteristics of the BSTS system.

Based on findings from preliminary simulations, a series of parametric simulations was performed to assess sensitivity of the BSTS system performance to different control strategies, slab configurations, and hydronic loop arrangements. A simple economic analysis of several BSTS system configurations was then performed to assess the overall economic viability of this concept. Finally, a comparison was made of this integrated technology to conventional energy-saving technologies commonly used in WSHP systems, as well as to a system using a central heating and cooling plant.

The BSTS system was shown to reduce building mechanical cooling loads for the building modeled in this study. When properly applied, the technology saves significantly more energy than is saved in identical buildings without the BSTS system; annual HVAC energy savings as high as 18% were predicted. However, in contrast to conventional thermal storage technologies, the BSTS system has a negligible impact on peak loads and thus does not reduce building peak electrical demand.

The estimated annual energy savings for the building with the BSTS system were found to be highly sensitive to climate. The BSTS system performed best in hotter, drier climates. The benefits in cold climates and hot, humid regions appear to be limited; annual energy savings of only a few percent were predicted.

Performance of the BSTS system was found to be significantly enhanced through simulation of better controls. Other parameters were also found to have an impact on the BSTS system performance, including configuration of the hydronic piping circuit and the physical characteristics and thermal properties of the slab.

Unfortunately, the cost of installing a BSTS system appears to be far too exorbitant to pay back its energy-saving potential. Too, less-expensive, conventional energy-saving technologies for WSHP applications or central systems both performed nearly as well or better than the BSTS system in simulations and should probably be considered first.

CONTENTS

FOREWORD	iii
SUMMARY	v
1.0 INTRODUCTION	1.1
2.0 BACKGROUND	2.1
2.1 SUBSYSTEM INTERACTION RESEARCH	2.1
2.2 INTEGRATION OF THE HVAC AND MASS SUBSYSTEMS	2.1
2.3 TEST BUILDING	2.6
2.4 COMPUTER SIMULATION	2.6
3.0 THE BASE CASE BUILDING MODEL	3.1
3.1 TRNSYS BUILDING SYSTEM MODEL	3.1
3.2 COMPONENT MODULES	3.5
3.2.1 Building Zone Model	3.6
3.2.2 Water-Source Heat Pump Model	3.8
3.2.3 Cooling Tower Model	3.9
3.3 BASE CASE UNCERTAINTIES AND CALIBRATION PROCEDURE	3.10
3.3.1 Unknown/Inconsistent Control or Occupant Behavior	3.11
3.3.2 Uncertainties in Equipment Operation	3.12
3.3.3 Lack of Basic Data	3.13
3.3.4 Additional Discrepancies	3.14
3.4 BASE CASE BUILDING MODEL CALIBRATION RESULTS	3.14
4.0 BUILDING STRUCTURAL THERMAL STORAGE SYSTEM MODEL	4.1
4.1 SLAB MODEL	4.1
4.2 BSTS SYSTEM MODEL	4.3
4.3 BSTS SYSTEM PERFORMANCE WITH SIMPLE CONTROL SCHEME	4.7

4.3.1	BSTS System in Seattle	4.7
4.3.2	Diurnal Response of BSTS System with Simple Control Scheme	4.8
4.3.3	Effects of Climate on Building Structural Thermal System Performance	4.17
5.0	BSTS BUILDING MODEL PERFORMANCE OPTIMIZATION STUDIES	5.1
5.1	GENERAL METHODOLOGY	5.1
5.2	BSTS SYSTEM PARAMETRIC STUDIES	5.2
5.2.1	Control Strategies	5.2
5.2.2	Slab Configuration	5.7
5.2.3	System Configuration	5.11
6.0	ECONOMIC VIABILITY ASSESSMENT	6.1
6.1	ENERGY USAGE AND DEMAND	6.1
6.2	COST ESTIMATING	6.3
6.3	POTENTIAL HIDDEN COSTS	6.9
6.4	RESULTS	6.9
7.0	COMPARISON WITH OTHER HVAC TECHNOLOGIES	7.1
7.1	BASE BUILDING WITH AIR ECONOMIZER	7.1
7.1.1	Air Economizer System Description	7.2
7.1.2	Results	7.3
7.2	BASE BUILDING WITH CONVENTIONAL THERMAL STORAGE	7.4
7.2.1	Thermal Storage Water Tank Model Description	7.5
7.2.2	Results	7.5
7.3	CENTRAL HEATING/COOLING SYSTEM	7.8
7.3.1	Model Description	7.9
7.3.2	Central System Component Sizing	7.11

7.3.3 Results	7.11
7.4 ALTERNATIVE HVAC TECHNOLOGY ECONOMICS	7.15
7.4.1 Costs	7.15
7.4.2 Results	7.16
8.0 CONCLUSIONS AND RECOMMENDATIONS	8.1
9.0 REFERENCES	9.1
10.0 BIBLIOGRAPHY OF BUILDING SYSTEM INTEGRATION RESEARCH REPORTS	10.1
APPENDIX A: METERED DATA ANALYSIS	A.1
APPENDIX B: TRANSIENT SYSTEM SIMULATION PROGRAM	B.1
APPENDIX C: TRNSYS COMPONENT MODELS	C.1
APPENDIX D: THE MEAN RADIANT TEMPERATURE MODULE	D.1
APPENDIX E: MONTHLY ENERGY CONSUMPTION FOR HVAC COMPONENTS: BASE CASE COMPARED TO BSTS SYSTEM WITH SIMPLE CONTROL	E.1
APPENDIX F: ANNUAL ENERGY CONSUMPTION FOR HVAC COMPONENTS: BASE CASE COMPARED TO BSTS WITH SIMPLE CONTROL	F.1
APPENDIX G: GENERATION OF AVERAGE-DAY WEATHER DATA	G.1
APPENDIX H: SUMMARY OF MONTHLY DEMAND PEAKS	H.1
APPENDIX I: ECONOMIC ASSESSMENT COST ESTIMATE WORKSHEET	I.1
APPENDIX J: MONTHLY ENERGY CONSUMPTION FOR AIR ECONOMIZER SYSTEM	J.1
APPENDIX K: MONTHLY ENERGY CONSUMPTION FOR WATER THERMAL STORAGE SYSTEM	K.1
APPENDIX L: MONTHLY ENERGY CONSUMPTION FOR CENTRAL SYSTEM	L.1

FIGURES

2.1	Typical Water-Source Heat Pump Subsystems	2.3
2.2	Integrated BSTS and Heat Pump Subsystems	2.5
3.1	Plan and Elevation Views of the Single-Floor Zone Arrangements Used in the Building Model	3.3
3.2	Building System TRNSYS Model Block Diagram	3.4
3.3	Comparison of Metered and Simulated Monthly Total HVAC Energy Consumption	3.16
3.4	Comparison of National Oceanic Atmospheric Administration and Typical Meteorological Year Average Monthly Temperature	3.17
3.5	Metered Compared to Simulated Monthly Heat Pump Energy Consumption	3.18
3.6	Metered Compared to Simulated Monthly Boiler Energy Consumption	3.18
3.7	Metered Compared to Simulated Monthly Cooling Tower Energy Consumption	3.19
3.8	Simulated Monthly Energy Consumption of HVAC Components	3.19
3.9	Simulated Fraction of Annual Energy Consumption by Each HVAC Component	3.20
4.1	Slab with Water Loop Circuit	4.3
4.2	Simple BSTS System Control Scheme	4.6
4.3	Total HVAC Monthly Energy Consumption for Simulated BSTS Strategies, Seattle	4.9
4.4	Heat Pump Monthly Energy Consumption for Simulated BSTS Strategies, Seattle	4.9
4.5	Boiler Monthly Energy Consumption for Simulated BSTS Strategies, Seattle	4.10
4.6	Cooling Tower Monthly Energy Consumption for Simulated BSTS Strategies, Seattle	4.10

4.7	West Zone Temperature History for the Base Building During the Fourth Week of July	4.12
4.8	West Zone Heat Transfer History for the Base Building During the Fourth Week of July	4.12
4.9	West Zone Temperature History for the BSTS Building with Simple Control During the Fourth Week in July	4.14
4.10	West Zone Heat Transfer History for the BSTS Building with Simple Control During the Fourth Week of July	4.14
4.11	West Zone Temperature History for the BSTS Building with Simple Control During the Third Week of May	4.16
4.12	West Zone Heat Transfer History for the BSTS Building with Simple Control During the Third Week of May	4.16
4.13	Total HVAC Monthly Energy Consumption for Base Case and Simulated BSTS System Strategies, Los Angeles	4.18
4.14	Total HVAC Monthly Energy Consumption for Base Case and Simulated BSTS System Strategies, El Paso	4.19
4.15	Total HVAC Monthly Energy Consumption for Base Case and Simulated BSTS System Strategies, Houston	4.19
4.16	Total HVAC Monthly Energy Consumption for Base Case and Simulated BSTS System Strategies, Washington, D.C.	4.20
4.17	Total HVAC Monthly Energy Consumption for Base Case and Simulated BSTS System Strategies, Milwaukee	4.20
4.18	Total HVAC Annual Energy Consumption for Base Case and Simulated BSTS System Strategies	4.21
5.1	West Zone Temperature History for the BSTS Building with Slab Temperature Control During the Third Week of May	5.6
5.2	West Zone Heat Transfer History the BSTS Building with Slab Temperature Control During the Third Week of May	5.6
5.3	Integrated BSTS and Heat Pump Subsystems with Core/Perimeter Piping Switchover	5.12

5.4	Integrated BSTS and Heat Pump Subsystems with Series Piping of Zone Components	5.15
6.1	United States Census Regions	6.8
7.1	Central System Chilled Water and Condenser Water Loop	7.10
7.2	Central System Hot Water Loop	7.12
7.3	Central System Air Handling Unit	7.13

TABLES

5.1	Comparison of Total Annual HVAC Energy Consumption for Base Building, BSTS Building with Simple Control, and BSTS Building with Slab Temperature Control	5.5
5.2	Comparison of Total Annual HVAC Energy Consumption for Base Building, BSTS Building with Slab Temperature Control, and BSTS Building with Optimized Slab Temperature Control	5.8
5.3	Comparison on Total Annual HVAC Energy Consumption for Base Building, BSTS Building with Slab Temperature Control and BSTS Building with Switchover Hydronic Piping	5.14
5.4	Comparison of Total Annual HVAC Energy Consumption for Base Building, BSTS Building with Slab Temperature Control, and BSTS Building with Series Hydronic Piping	5.16
6.1	Summary of Total Annual Energy Consumption for BSTS Strategies	6.2
6.2	Peak Summer/Winter Electrical Demand for BSTS Strategies	6.2
6.3	Capital Equipment Requirements for BSTS System	6.4
6.4	Incremental Installed Cost of BSTS System	6.7
6.5	Financial Assumptions of Incremental Life-Cycle Cost Comparisons of BSTS System	6.7
6.6	Commercial Average Electricity Prices for 1990	6.9
7.1	Comparison of Total Annual HVAC Energy Consumption for the Base Building, BSTS Building with Slab Temperature Control, and Base Building with Air Economizer	7.3
7.2	Water Thermal Storage Tank Volumes	7.6
7.3	Comparison of Total Annual HVAC Energy Consumption for Base Building, BSTS Building with Slab Temperature Control, and Base Building with Water Thermal Storage	7.6
7.4	Total Annual Boiler Energy Consumption of Base Building with Water Thermal Storage	7.7

7.5	Comparison of Monthly Peak Demand for Base Building and Base Building with Water Thermal Storage	7.8
7.6	Comparison of Total Annual HVAC Energy Consumption for Base Building, BSTS Building with Slab Temperature Control, and Central System	7.13
7.7	Incremental Installed Costs of Alternative HVAC Technologies	7.16

1.0 INTRODUCTION

Previous research on building energy consumption has focused primarily on improving the performance of building subsystems operating independently rather than interactively. From this perspective, many of the potential energy-conserving improvements in individual subsystems have been identified. To now develop an understanding of how these independent building subsystems interact and may be integrated is clearly one of the key steps to further energy conservation. Yet, because of the complex and dynamic nature of the energy delivery process in commercial buildings, developing this understanding is also one of the most challenging steps.

In many commercial office buildings, heat is needed in one area or zone at the same time that cooling is needed in another. Even more common is the need for heating during one part of the day and cooling during another in the same spaces. If energy could be shifted between zones or stored for later use, a significant amount of it might be saved. It was also hypothesized that integrating a building's heating and cooling subsystems with the building's structural mass to collect, store, and deliver energy could be done cost-effectively to obtain these potential benefits.

To test this hypothesis, researchers at the Pacific Northwest Laboratory (PNL)^(a) analyzed the integration of heating, ventilating, and air-conditioning (HVAC) systems with the mass of a building. This analysis, performed for the U.S. Department of Energy (DOE) Commercial Building System Integration Research Program, included development of a numerical simulation model--the Building Structural Thermal Storage (BSTS) system--and modeling of a specific existing building into which the BSTS system was integrated. A series of parametric studies was used in an effort to optimize physical configuration and control strategies for the BSTS system, and life-cycle costs for this system were compared to the "base case" building without the BSTS system.

(a) Operated by Battelle Memorial Institute for the U.S. Department of Energy under Contract DE-AC06-76RLO 1830.

In this report, the PNL analysis is described in detail. The document is intended to serve as the final report for this research; initial findings were previously reported by Marseille et al. (1989).

Background information critical to this analysis appears in Section 2.0. A description of the modeling methodology used in developing the base case building model is given in Section 3.0, as are the results of calibrating this model to the metered data from the test building. Development and initial testing of the BSTS building model are presented in Section 4.0, along with a description of results from simulations made using this model in different climatic regions. Performance optimization studies of the BSTS system are discussed in Section 5.0, followed by a description of a simple economic analysis of the technology in Section 6.0. A comparison of the BSTS system to other HVAC technologies is discussed in Section 7.0. Final conclusions and recommendations are presented in Section 8.0. Detailed supporting information is provided in Appendixes A through L.

2.0 BACKGROUND

This section provides background information critical in understanding the analysis described in this report. The nature and purpose of building subsystems interaction research, of which the work reported here is a part, is described in Section 2.1. A general discussion of thermal structural storage in buildings, water-source heat pump (WSHP) subsystems, and the way these two building subsystems can be integrated is given in Section 2.2. Background on the existing building modeled in this study is given next in Section 2.3. Finally, the appropriateness of using a computer simulation code as the primary analytical tool for this study is discussed in Section 2.4.

2.1 SUBSYSTEM INTERACTION RESEARCH

This investigation is a part of subsystems interaction research, which is one element of the commercial portion of the Building System Integration (BSI) program, a multiyear research endeavor. Subsystems interaction research will develop an understanding of how various materials, components, equipment, and subsystems interact or can be made to interact, and how these interactions affect energy consumption. Examples of subsystems in commercial buildings are the building structure and envelope, the HVAC energy conversion equipment, the HVAC distribution subsystems, and the lighting subsystems. Collectively, they provide enclosure and division of space, temperature and humidity control, ventilation, internal heating/cooling and electrical load control, and illumination.

2.2 INTEGRATION OF THE HVAC AND MASS SUBSYSTEMS

Energy storage in building structures occurs passively in all buildings to some extent and has been shown to have nontrivial energy impacts (Eto 1985). When HVAC equipment is shut off or set back during unoccupied periods, the effect of the building mass often results in long temperature recovery time periods. To reduce the peak during morning temperature recovery in the heating season, some building owners leave the lighting on overnight to help maintain the space temperature. For the cooling mode, work by Lawrence

Berkeley Laboratory (LBL) researchers indicates that Monday morning recovery from the passive heat addition to the building mass over a summer weekend negates part or all of the savings gained by the system shutdown (Eto 1985). Purging a building on Monday mornings with cool outdoor air has been shown to be one remedy to the problem of summer weekend recovery. However, the efficiency of precooling the building mass using outside air that is distributed via the ventilation system is low.

On the other hand, active thermal storage in building structures is a relatively new concept. An LBL case study of a Swedish commercial building (Andersson et al. 1979) discussed the use of a massive floor structure for storing heat, via an air distribution subsystem, using hollow-core concrete slabs. More recently, researchers have focused on the impact of "precooling" the building at night and on weekends to reduce daytime cooling loads by using supply air that has either been cooled mechanically with a water chiller or using an air or water economizer cycle (Rudd, Mitchell, and Klein 1990; Braun 1990). However, using a water distribution subsystem to charge and discharge the energy stored in the structure appears to have the potential for better control in most applications, because water has a higher heat capacitance and heat transfer rate than air and thus allows a more thorough and rapid purging of stored heat.

The HVAC subsystems used in this study consist of decentralized water-source heat pumps in a closed water loop. This technology is not new, but it has been recently gaining popularity among designers because of occupant satisfaction (due to excellent zone temperature control) and because of its flexibility in both new design and retrofit applications. A schematic of a typical WSHP subsystem design is shown in Figure 2.1.(a)

Water-source heat pump subsystems with thermal storage have been studied only sporadically. A Canadian study (Rogers 1980) indicates that adding storage to a water loop heat pump subsystem can substantially improve

(a) This system corresponds to the base building simulation model described in Section 3.0.

FIGURE 2.1. Typical Water-Source Heat Pump Subsystems

subsystem performance. In practice, thermal storage can be added by increasing the heat capacitance of the water loop by increasing the amount of water or, in principle, by coupling the loop to the building thermal mass. The structural thermal mass can be used for storage in a water loop heat pump subsystem because of the inherent dynamics of the system configuration. Time lags in heat transfer into and out of thermal mass can be used beneficially. Because heat pumps can either extract heat from the water loop or reject heat to it, all of the heat capacitance of the distribution piping and fluid, as well as the heat capacity of any storage tank or structure that is thermally coupled to the system, is available for storage.

The drawing in Figure 2.2 shows the water loop for the WSHP and BSTS system configuration initially modeled in this study.^(a) A secondary water loop connects the conventional WSHP water loop to pipes embedded within concrete floor slabs. Heat is transferred to and from the structure and the water loop via these pipes.

Practical considerations associated with using the floor slab as the storage medium involve material compatibilities, component connections, and configuration of the piping in the water loop. However, embedding water pipes within concrete is a proven concept; thus, many of these considerations have already been addressed in past research. Practical experience has been gained in solar-charged as well as conventional radiant slab heating applications (*Solar Heating and Cooling* 1979).

The cost of these applications is reasonable in comparison to other HVAC subsystems. Established design criteria for the application include the following: pipes embedded in floors must be leakproof; long continuous lengths and good flexibility for bending are necessary to minimize the use of any connections and elbows; the pipe must be inert to corrosion, scaling, and electrolysis; and excellent long-term pressure rating and easy installation are needed.

Another important practical consideration for the design and operation of the BSTS system is the effect of structure thermal storage on occupant comfort, which, in turn, affects control strategies. For example, comfort requirements will set limits on the allowable temperature at the surface of the concrete structural elements and, ultimately, on the allowable temperature swings in the water loop, a key determinant of the amount and rate at which thermal energy can be delivered to the slabs. Also, slab temperatures should be moderated to avoid possible condensation during high wet-bulb temperature conditions.

(a) Figure 2.2 is presented here to illustrate conceptually the integrated design. More detailed descriptions in Section 4.0 will also refer to this figure, because it portrays the actual building subsystems that were simulated.

FIGURE 2.2. Integrated BSTS and Heat Pump Subsystems

2.3 TEST BUILDING

The simulation model is based on an existing six-story office building located in Seattle that was monitored by Seattle City Light (SCL) between 1983 and 1985 (Cleary and Crimmin 1986). Metered data collected by SCL for the existing building are discussed in Appendix A.

The building used for this study is typically occupied 50 hours a week, from 8 a.m. to 6 p.m., Monday through Friday; the average occupancy is 400 people. Built in 1976, the building consists of two rectangular sections, one four stories tall and the other six. The structure was built on a concrete slab with precast concrete walls. Total floor area is approximately 89,550 ft². Forty-seven percent of the gross wall area is glass.

Heating and cooling are provided by 97 water-source heat pumps that are connected to a common water loop. Two electric boilers provide the auxiliary heat for the water loop, and an open cooling tower is used to reject excess heat during cooling. Ventilation air is provided by a constant-volume supply/exhaust system such that about 20% of the supply air to each zone is outside air. The ventilation system uses a water-to-air heat recovery system between the supply and exhaust air streams and a resistance duct heater to temper outside air. The system operates for 12 hours on weekdays only. There is no economizer (100% outside air) mode in this system; this eliminates the need for larger ducts and fans and for fan capacity modulation. Lighting is provided predominantly by fluorescent lamps with some incandescent task lighting.

2.4 COMPUTER SIMULATION

For this study, PNL developed a computer model for analyzing the integrated BSTS system using the Transient System Simulation (TRNSYS) program (Solar Energy Laboratory 1988) (see Appendix B). For a building-level analysis such as this project, computer simulation is the most cost-effective and appropriate analysis tool for initial studies. It can readily provide the researcher with much insight and understanding of the concept by allowing many variations to be examined with relative ease.

Although field monitoring can supply some actual building operation data that are unavailable in simulations, the costs generally preclude full-scale experiments. In addition, because independent variables such as weather have strong effects, consistent comparisons and assessments of alternative subsystems within the building are difficult, if not impossible. In contrast, parametric sensitivity analyses may be readily conducted wherein most variables are held constant while others are varied through a range of physically realistic values.

When using computer simulations as a research tool, care must be taken to appreciate the inherent limitations of mathematical modeling. A model of a complex system can rarely perform exactly as the real system does in every detail. Computer simulation should thus be used to determine the relative contribution of individual variables to the specific phenomenon under observation.

3.0 THE BASE CASE BUILDING MODEL

As a first stage in developing the BSTS system model, PNL researchers used TRNSYS to simulate the existing test building (termed the base case building). To verify that the actual building was being realistically modeled, the input was calibrated against metered data. Using the calibrated base case building model as a starting point, the BSTS system was then added to the model. The simulated results of the base case model with the BSTS system provided a realistic estimate of the impact of integrating the HVAC subsystems and the building structural mass.

This section focuses on the base case building model. First, the approach to modeling the building system is discussed in Section 3.1. Key component module subroutines used in the building model, some of which were developed specifically for this project, are then described in Section 3.2. A discussion of modeling uncertainties and the methodology used to calibrate the base case model to achieve close agreement with metered data is covered in Section 3.3. Finally, results of the base case building model calibration are discussed in Section 3.4.

3.1 TRNSYS BUILDING SYSTEM MODEL

TRNSYS can mathematically represent building envelope thermal properties, HVAC subsystems performance, occupant heat and moisture generation, light and equipment convective and radiative heat gains, solar radiation, heat conduction to and from the environment, and other properties for an entire building. However, because large TRNSYS models require great computer speed and memory, it was not practical to simulate the entire building in this study. For this reason, the building model was simplified in several ways.

The first such simplification was to model the entire multistory base case building as a single representative floor with envelope characteristics based on the actual building monitored by SCL. This is equivalent to assuming that the heating or cooling loads on all floors are equal, an assumption justified for commercial buildings where the space usage on each floor is roughly identical and the building envelope is homogeneous. When each floor

has approximately the same load profile, there is no net heat transfer between floors. Furthermore, although the first floor differs from the others in that it is thermally coupled to the ground via its slab, this effect was assumed to be small. This assumption is valid unless the building has large conditioned spaces underground. And too, in most commercial buildings, edge heat losses from the first-floor slab to the ground are small compared to other losses because the area adjacent to the perimeter of a large slab is much smaller than the total floor area. The roof area, on the other hand, does exchange significant heat with the environment and was included in the model. Because the base building has six stories, one-sixth of the roof was modeled and the heat conductance was coupled to the single floor, which represents the entire building.

The next simplifying aggregation was to assume that the single floor can be modeled as a core zone, four perimeter zones, and a common ceiling plenum (see Figure 3.1). In the model, the core and perimeter zones are each conditioned by a single large heat pump that represents the performance of a number of smaller units. In the actual building monitored by SCL, each floor is served by 16 heat pumps, ranging in capacity from 1 to 3 tons of cooling. In the model, each floor is served by five units, one for each zone (excluding the plenum), sized according to the amount of space served. The efficiency of the aggregated heat pumps is the same as that of individual smaller units in the actual building. Boiler, cooling tower, and other component capacities were scaled by one-sixth as well to maintain the proper fluid temperatures and flow rates. The energy consumptions of these components were each multiplied by six to obtain whole-building totals comparable to the actual six-floor building.

Figure 3.2 shows the information flows in the TRNSYS base case building model. Hourly values of temperature, solar radiation, humidity, and wind speed from Typical Meteorological Year (TMY) weather data (National Climatic Center 1981) were used to determine loads on the building zones. Internal heat gains from people, lights, and equipment are established by 24-hour weekend and weekday profiles sequenced into a repeating weekly profile. Using the weather, internal gains, and the supply air temperature and mass flow

* building model roof area (one-sixth of existing building roof area) not shown

FIGURE 3.1. Plan and Elevation Views of the Single-Floor (One-Sixth Building) Zone Arrangements Used in the Building Model

FIGURE 3.2. Building System TRNSYS Model Block Diagram

rate, an energy balance was calculated on each zone. From the energy balance, a zone temperature was calculated. This zone temperature, in turn, triggers the room thermostat that commands the heat pump operations. Cooling tower or boiler operation occurs when the water loop temperature exceeds or falls below setpoints of 85°F or 60°F, respectively, as the heat pumps exchange heat with the water loop. Electric energy input to the boiler, cooling tower, heat pumps, fans, pump, and duct heater are calculated and output.

Outside air is controlled by a time clock subroutine such that, during weekday occupied hours, 20% of the total supply air to the zones is tempered fresh air. The outside air is preconditioned by an electric resistance heater and a glycol runaround loop, which exchanges heat with exhaust air that comes from the plenum. The outside air temperature and flow rate are passed to the plenum zone model where the mixed air temperature is calculated. Plenum air temperature and return air temperature are input to the heat pumps where heat pump fan and compressor power are calculated.

TRNSYS calculates air and water temperatures that depend on control signals based on these same temperatures. Because this information flow is recursive, TRNSYS iterates through these calculations during each time step until the temperatures calculated in successive iterations converge to within a specified tolerance (a tolerance of 0.2% was used for this study). Similar iterative calculations are performed throughout the simulation model.

3.2 COMPONENT MODULES

In TRNSYS a subsystem model is made up of a number of different component models. Many of the individual component models used in this study are available with TRNSYS. Details of these components may be found in the manual (Solar Energy Laboratory 1988) and will not be repeated here. However, the existing TRNSYS building zone model will be summarized in terms of the approach it uses to model the zone heat exchanges (Section 3.2.1).

Five new system component models were also developed by PNL researchers specifically for the base case and BSTS building models: cooling tower, heat pump, water loop boiler, water loop thermostat (used to control cooling tower and boiler operations), and floor slab used in the BSTS system. Two of these

components, the boiler and thermostat, are extended versions of existing TRNSYS modules. They were modified so that boiler heat output during the simulation would be linearly proportional to inlet water loop temperature over a user-specified range above the heating setpoint. The thermostat module was also modified for use in the BSTS system control scheme to activate the cooling tower freecooling cycle (see Section 4.2). The heat pump, cooling tower, and floor slab models are, on the other hand, entirely new. The simulation approaches of the heat pump and cooling tower are described in Sections 3.2.2 and 3.2.3, respectively. The floor slab model, used to simulate thermal charging and discharging of the floor using water flowing through embedded pipes within the slab, is described in Section 4.1. Appendix C includes the computer code for all five of these component models.

3.2.1 Building Zone Model

The component model used to simulate the thermal behavior of the building was the TRNSYS Detailed Zone Model (TRNSYS Type 19). The zone model uses temperature level control (discussed in Appendix B). The zone temperature and humidity are functions of the ambient conditions, the heating and cooling equipment inputs, and any other thermal inputs (e.g., the passive thermal discharge of the thermal storage slab). This model simulates the zone sub-components such as walls, windows, flat roof, doors, and floor. A multizone building such as the one analyzed for this report is simulated by applying this model to each zone.

An energy balance is formulated for each subcomponent within each zone to account for the conduction, convection, and radiation heat transfers. These equations are described in the TRNSYS manual (Solar Energy Laboratory 1988). Conduction through walls is modeled using the American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. (ASHRAE) transfer function approach (ASHRAE 1988). Convection heat exchange from walls is determined using constant values for convection heat transfer coefficients and the difference between the surface temperature and the room temperature.

Solar radiation passing through windows is assumed to be isotropic and to strike diffusely, reflecting gray (i.e., having radiative properties that do not vary with wavelength) interior surfaces within the zone. TRNSYS

computes overall exchange factors for solar radiation at the beginning of a simulation using surface reflectivities input by the user and geometric view factors. A simplified method for determining geometric view factors that was available in TRNSYS was used for the base building that computes these factors from area ratios.^(a) In TRNSYS all surfaces are assumed to be black (i.e., perfect absorbers) to radiation from lights, people, and equipment, as well as long-wave radiation from other surfaces. Radiative gains from people are assumed to be 70% of their total sensible energy output; the remaining 30% is assumed to be convected directly to the indoor air. Radiation heat transfer from lights, people, and equipment is apportioned evenly over all zone interior surface areas. Long-wave radiation heat transfer between interior surfaces is modeled using constant, linearized radiation heat transfer coefficients. These radiative coefficients are calculated for pairs of surfaces in the zone using average initial surface temperatures and the black body view factors.

An energy balance for each subcomponent and the room air is set up in matrix format. For the long-wave radiative and convective exchanges of each interior surface, an equivalent zone temperature is used. This equivalent zone temperature is analogous to the sol-air^(b) temperature used for the exterior surfaces to account for both solar and convection heat transfers. The surface temperatures are calculated at each time step by solving the simultaneous equations using direct matrix-inversion (Solar Energy Laboratory 1988).

-
- (a) This simplified approach for computing geometric view factors was used because of uncertainties in the actual geometric configuration of the interior space in the base building. Sensitivity studies of the base case building model were performed, however, that compared results obtained using this approximation to those obtained when approximated user-input view factors in the southern exposure zone were used. The differences were found to be insignificant.
- (b) Sol-air temperature is the inside air temperature that, in the absence of long-wave surface radiative exchange, results in the same heat transfer as actually occurs.

3.2.2 Water-Source Heat Pump Model

The heat pump component model developed for this study was not a detailed equipment model that simulates the individual energy conversion components, such as the compressor and the heat exchangers. Rather, regression on the manufacturer's rating data was performed to obtain coefficients for equations then used to complete performance. For example, to compute the heat pumps total capacity when operating in cooling mode, a capacity factor C_T is computed by

$$C_T = (A_1 + A_2 \cdot \dot{m}_w + A_3 \cdot T_w) \cdot (A_4 + A_5 T_{DB} + A_6 T_{WB}) \cdot (A_7 + A_8 \cdot \dot{m}_A) \quad (3.1)$$

where $A_1 - A_8$ = coefficients from regression

$\dot{m}_w$ = water flow rate (gal/min)

$\dot{m}_A$ = air flow rate (ft³/min)

T_{DB} = entering air dry-bulb temperature (°F)

T_w = entering water temperature (°F)

T_{WB} = entering air wet-bulb temperature (°F).

C_T is then multiplied by the nominal heat pump capacity. Next, a time constant is applied to this product (C_{TN}) to approximate transient performance, i.e.,

$$C_{TOT} = C_{TN} - (C_{TN} - C_{TO}) e^{-\frac{\Delta t}{\tau}} \quad (3.2)$$

where C_{TOT} = total capacity at end of time step (Btu/h)

C_{TN} = new time step capacity (Btu/h)

C_{TO} = old time step capacity (Btu/h)

Δt = time step increment (s)

τ = time constant.

With expressions like these, the model calculates the steady-state sensible and (for cooling mode) latent heat capacities and power inputs of the

heat pump at given entering water and air conditions. The amount of heat rejected or extracted from the water loop is determined by an energy balance on the refrigeration loop. Finally, the exit temperature and humidity of the air and the exit temperature of the water are calculated based on the heat addition or extraction.

3.2.3 Cooling Tower Model

A cooling tower model is used to calculate the exiting water temperature at a given set of conditions (i.e., inlet air and water temperature and flow rates). The most commonly used cooling tower model is the Merkel model (ASHRAE 1988). It involves a numerical integration of the heat transfer rate across the two flow streams through the tower. To predict the outlet states of the flow streams, an iterative process is required. In order to reduce the execution time of the model, a simpler approach developed by Whillier (Whillier 1976) was used that follows closely the algorithm used by Braun, Mitchell, and Klein (1987).

The Whillier model was established based on the assertion that the performance of a cooling tower may be characterized by a single linear function that correlates its effectiveness, E ,^(a) to its capacity factor, R :

$$E = F/[F + (1 - F)R] \quad (3.3)$$

where F , a relative factor of merit for the cooling tower, is determined from experimental results or manufacturer's data, while R is directly computed given air and water inlet conditions. After using the above relation to determine E , the exit state of the water for any off-design conditions is easily determined by multiplying E by the minimum of either 1) the amount of heat/mass transfer required to bring the air to saturation or 2) the amount of heat transfer required to reduce the water temperature to the air entering

(a) Cooling tower effectiveness is defined as the ratio of the actual to the maximum possible heat transfer from the water to the air stream. The capacity factor is the ratio of the minimum to maximum thermal capacities of the air and water streams.

wet-bulb temperature. This product is then multiplied by a time constant (similar treatment to Equation [3.2]) to obtain the tower's total capacity.

The results calculated by the Whillier model have been compared with the Merkel model. The accuracies of the two models have been shown to be essentially the same, and they both compare favorably to experimental data (Braun, Mitchell, and Klein 1987). Whillier recommends that, for a typical commercial cooling tower, an F of 0.65 be used. Because of a lack of specific data for the actual cooling tower in the existing building, this value of F was chosen for the model.

3.3 BASE CASE UNCERTAINTIES AND CALIBRATION PROCEDURE

In developing the overall building simulation model, physical parameters were required for each of the building components. Parameters are variables that are held constant during the simulation in contrast to inputs that may vary with time. Examples of parameters include the physical dimensions and configuration of the building spaces, zone temperature setpoints, operation schedules, material properties, equipment capacities, lighting and equipment distributions, and rated efficiencies. Inputs include temperatures, flow rates, humidity, and control signals. Inputs are passed from one component subroutine to another and are calculated based on the outputs from other components, subroutine algorithms, and parameter values. Thus, changing the value of a parameter changes the outputs from the component subroutine, which, in turn, changes the inputs to other components. This ultimately affects the results of the simulation, namely, the equipment energy consumption.

Many parameters in the test building were not known or were not well understood. For those parameters, a sensitivity analysis was performed to establish the model's responsiveness to the parameter value within a physically realistic range. If the model was insensitive, then a high degree of uncertainty was considered permissible, and a reasonable value within the range was selected.

If the model was sensitive to an unknown or uncertain parameter, trial and error simulations were used to estimate an appropriate and/or typical value. Reasonableness was checked by comparing the simulation outputs such as

zone or water loop temperature fluctuations to what is accepted as being typical for the system being modeled, and by comparing total energy consumption to the SCL metered data.

Finally, many parameters (such as the manual operation of outside air fans and re-insulation of the roof) were changed seasonally in the actual building, either intentionally or unintentionally. These changes could have been modeled but were not because of the considerable difficulty in doing so.

Because the objective of this study was to determine the effect of the BSTS on overall energy consumption--not individual parameter quantities--these parameters were exploited (within reasonable bounds) during calibration of the base building model so the best agreement with metered data was obtained. This process entailed an iterative series of parametric simulation runs in which "sensitive" parameters for which actual values were uncertain or unobtainable were varied through trial and error. Final values of these parameters were established when simulated monthly HVAC energy consumption was found to be within 10% of the metered data.

Model deficiencies resulting from uncertainty in values of building parameters for the actual building were uncovered during the calibration. These deficiencies significantly affected the simulation results. Discussed in Sections 3.3.1 through 3.3.4, they include

- unknown or inconsistent control or occupant behavior
- uncertainties in equipment operation
- lack of basic data.

3.3.1 Unknown/Inconsistent Control or Occupant Behavior

Exact thermostat setpoints for the different zones were unknown. They appeared to be in the range of 70 to 80°F, but were not uniform either with time or location. As there are thermostat controls located in the occupied areas of the actual building, it is likely that they were adjusted by the occupants. Constant setpoints of 78°F for cooling and 71°F for heating were chosen during the calibration of the base case.

Both occupant use of operable window shading (venetian blinds) and actual cloud cover were unknown, resulting in uncertainties related to solar gains. The calibration of the test building suggested that overall window shading factors were quite low.

Occupancy levels are not known and are subject to changes, resulting in variable internal loads. Sporadic occupancy on Saturdays was not modeled. Constant occupancy during normal building occupied hours was assumed.

The actual monthly lighting and equipment loads varied by a factor of two during 1985, as indicated by metered data. This is probably related to changes in building tenancy, because the metered data indicates that electricity consumption rose roughly 25% in July. Constant lighting and equipment loads were used, the values of which were determined through model calibration. Distributions of these loads in the model were also determined through calibration.

3.3.2 Uncertainties in Equipment Operation

Night and weekend thermostat setbacks are inconsistent. The heat pumps are supposed to be shut off each night by disconnecting power to the thermostats, but the January metered data, for example, reveals nighttime heat pump operation that is inconsistent with this strategy. For the model, 24-hour heating operation with a 5°F night setback during nights and weekends was used. The cooling was shut off on weekends.

Constant boiler and cooling tower setpoints of 60°F and 85°F, respectively, were used in the model, but may deviate in the actual building due to alteration or drift.

It is not certain from the metered data whether heat pump fans are on all day or cycle with compressors. Because these subsystems are most commonly controlled so heat pump fans are operational whenever the outdoor ventilation fans operate, this operation schedule was used in the model.

The metered data shows seasonal control strategy changes, most notably in the supply and exhaust fan operation of the ventilation system. Although this could have been simulated using individual monthly runs, it was overburdening for annual runs; therefore, these changes were not modeled.

An outside air fraction of 20% was used in the model, but it is not clear how the use of indirect ducting through the plenum affects this in the actual building. (The flow rate through a fan is a function of the power input and the static pressure. Because a plenum has an uncertain static pressure, the outdoor air fraction is affected.)

3.3.3 Lack of Basic Data

The effective quantity of thermal mass in the zones was unknown. It depends on not only the quantity of heat storage material in the zone (such as furniture), but also its composition, color, and placement, as well as the rate of heat transfer to and from the zone air. This important parameter, which tempers the heating and cooling loads, was estimated using iterative calibration of the test building.

The rates of infiltration, exfiltration, and interzonal air exchange were not known. Constant rates of infiltration and exfiltration were assumed; the interzonal air exchange was not explicitly modeled.

The effective thermal mass in the water loop, which also includes the thermal mass of the piping, heat exchanger coils, and cooling tower sump water, was not known. This parameter was estimated by calculating the total water volume in the loop and comparing it with metered water loop temperature responsiveness.

The effectiveness of the glycol loop outside air heat recovery heat exchanger was unknown because the entering and exiting air temperatures were not metered. The value used in the model (0.64 overall) is believed to be representative.

Thermal storage effects in glazings (if any) and light fixtures were assumed to be zero. There was no definitive data on this phenomenon nor on the apportioning of lighting heat gains between the conditioned zones and the plenum and of convective versus radiant heat transfer. The type of glass used in the windows was unknown; thus, properties of the glass (e.g., transmittance for diffuse solar radiation, inside film coefficient, window loss coefficient) were considered parametric variables during the calibration of the base case.

Because actual physical floor configurations were unavailable, the heat gains and interzonal air exchange between the core and perimeter were modeled indirectly. The model simplified the interaction by assuming that a continuous interior wall separated the core from the perimeter zones. The conductance and other properties of the wall were then established during the calibration of the base case. Because of uncertainties in the actual physical floor configuration and the (somewhat) arbitrary way in which the model was divided into zones, the modeling of radiation heat transfer between interior surfaces had to be simplified. For example, geometric view factors between interior surfaces were computed based on area ratios (see Section 3.3.2).

3.3.4 Additional Discrepancies

Replacement of building roof insulation during 1985 added significant thermal resistance to the roof. The thermal resistance of the original insulation was poor and its actual value unknown partly because it reportedly was partially water-saturated. In the simulation model, it was assumed that the overall resistance of the roof was constant throughout the year. The value of this constant resistance was set during calibration.

There was no accounting for holidays in the model. TRNSYS has this simulation capability but it is cumbersome to use. It is uncertain whether occupancy is actually negligible on holidays and likely that many occupants take additional vacation time near holidays.

The use of TMY data for Seattle-Tacoma Airport, rather than local data for 1985 (the latter was not readily available), resulted in somewhat erroneous instantaneous conductance and solar loads. As shown in Section 4.1, in the long run these errors tend to cancel out unless a seasonal or annual bias occurs, which results in significant variations in local average temperatures during the metering period from the typical average.

3.4 BASE CASE BUILDING MODEL CALIBRATION RESULTS

An important step in this investigation was the calibration of the base case building model with metered data (see Section 3.3), because it helps verify that the actual building was realistically modeled. This, in turn,

provides confidence that the simulated results of the building model with the BSTS system added (see Section 5.0) provides a reasonable estimate of the impact of integrating the HVAC system and the building structural mass.

The simulated monthly overall HVAC energy consumption results are compared with the available corresponding metered data in Figure 3.3. These comparisons are the most reasonable that could be obtained, given the limitations of the data and the computer model (see Section 3.3). Monthly simulated energy consumption trends match fairly closely to metered performance. A significant observed discrepancy in November, unresolved by the calibration, appears to primarily be attributable to the difference between the TMY weather data^(a) used in the simulation and the actual weather at the building site during 1985. As indicated in Figure 3.4, the TMY average monthly temperature was approximately 11°F higher in November than National Oceanic and Atmospheric Administration (NOAA) temperature data for 1985 (measured at the Seattle-Tacoma airport). This difference correlates with the discrepancy between metered and simulated building energy consumption. Differences in energy consumption in other months may be attributed primarily to the many assumptions made in development of the model. No attempt was made to use NOAA weather data for 1985 because solar radiation data is not included and the merging of TMY data with NOAA data would produce numerous hours when solar radiation and temperatures were inconsistent.

In Figures 3.5 through 3.7, the simulated monthly energy consumption of the major HVAC components is compared to the metered data. Although differences are larger than 10% for many of the months, the operational trends for each component compare well, as does the overall annual energy consumption.

Figures 3.8 and 3.9 summarize the simulated monthly and annual energy consumption of the HVAC components. These also indicate that the energy consumption of the HVAC components is distributed in a manner characteristic of this type of commercial building, both on a monthly and annual basis.

(a) The TMY weather data represents actual data from "typical" months, and includes values of temperatures, solar radiation, and wind speed in a year.

FIGURE 3.3. Comparison of Metered and Simulated Monthly Total HVAC Energy Consumption

Overall, the calibration approach provided a model that compares reasonably well with metered data. Major discrepancies appear to be linked with the differences between the weather data used in the simulation and the actual weather conditions in 1985 at the building site and the many uncertainties in the building model. Because trends in energy consumption match fairly well, these discrepancies should not significantly impact the usefulness of the model as a tool for studying the integration of the HVAC subsystems and the building structural mass.

FIGURE 3.4. Comparison of National Oceanic Atmospheric Administration and Typical Meteorological Year Average Monthly Temperatures

FIGURE 3.5. Metered Compared to Simulated Monthly Heat Pump Energy Consumption

FIGURE 3.6. Metered Compared to Simulated Monthly Boiler Energy Consumption

FIGURE 3.7. Metered Compared to Simulated Monthly Cooling Tower Energy Consumption

FIGURE 3.8. Simulated Monthly Energy Consumption of HVAC Components

FIGURE 3.9. Simulated Fraction of Annual Energy Consumption by Each HVAC Component

4.0 BUILDING STRUCTURAL THERMAL STORAGE (BSTS) SYSTEM MODEL

The development and initial simulation results of the "prototype" BSTS system model are described here. This model was used to demonstrate functionality, to obtain an initial assessment of the energy-saving potential of the HVAC/structure integration, and to begin to gain a broader understanding of the operational characteristics of the BSTS concept. Knowledge gained from this work was used to help focus the optimization studies described in Section 5.0 that look at improved BSTS control strategies, slab thermal response optimization, and alternative hydronic water loop configurations.

The numerical model developed by PNL researchers to simulate the floor slabs used in the BSTS system is discussed in Section 4.1. A description of the BSTS system configuration and the initial "simple" control strategy is given in Section 4.2. Simulation results for Seattle using the simple control strategy are presented and interpreted in Section 4.3, along with results of simulations made using TMY weather data from five other cities: El Paso, Houston, Los Angeles, Milwaukee, and Washington, D.C.

4.1 SLAB MODEL

A numerical heat transfer model was developed by PNL researchers to simulate the thermal charging and discharging of the hot or cold water flowing through 1-in. inside diameter polybutylene pipes^(a) embedded in the building floor slabs. The model solves the two-dimensional unsteady conduction equation in cartesian coordinates:

$$\frac{1}{\alpha} \frac{\partial T}{\partial t} = \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \quad (4.1)$$

where T = temperature
 t = time
 α = thermal diffusivity.

(a) Reports on existing radiant slab heating applications using polybutylene confirm the practicality of this material (Shell Chemical 1984).

Explicit formulas for the temperature field were derived from this relation using the central difference approximation for partial derivatives. These formulas are applied in the model at each node in a user-specified grid that includes portions of the slab, pipe wall, and water.

The model assumes uniform surface temperature at the top and bottom of the slab, consistent with the assumptions of the TRNSYS zone model. The temperature gradient along the direction of the water flow is not directly modeled. Instead, an iterative procedure is used to determine an average temperature (average of slab inlet and outlet temperature) of the water. It is then assumed that the water throughout the pipes is at this average temperature. This assumption is justified because these pipes are typically laid in multiple parallel circuits at flow rates high enough to ensure that water of about the same temperature is available at every point of the grid (Shell Chemical 1984). Hence, the entire pipe system is made up of sets of identical circuits. Figure 4.1 shows a sample pipe pattern for a single circuit. Because of the symmetry of the pipe field, the simulation of only the smallest common element (half of a pipe cross section and its associated slab) is required, simplifying the formulation of the model. Figure 4.1 also shows the portion of the slab being modeled.

At each simulation time step, surface temperatures calculated from the Zone Model are provided as inputs to the slab model. The slab model then calculates the amount of heat transferred from the fluid to the slab, and that transferred from the slab surfaces to the conditioned space (above) and plenum space (below). These heat transfer rates are a function of the fluid temperature and flow rate through the embedded pipes, as well as of pipe diameter and spacing, slab thickness, and surface temperatures. In addition to the heat transfer rates, the cross-sectional temperature distribution of the slab, hence the stored energy, is also calculated from the model. At the next time step, the heat transfers at the surfaces are input back to the zone model, the zone model computes surface temperatures of the surfaces enclosing the zone, and the simulation continues.

For the simulations reported in this study, 6 in. of concrete having a density of 145 lbm/ft^3 , a thermal conductivity of $1.0 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$, and a

FIGURE 4.1. Slab with Water Loop Circuit

specific heat capacity of $0.21 \text{ Btu/lbm}\cdot^{\circ}\text{F}$ were modeled. This heavyweight concrete material was selected somewhat arbitrarily because of its high thermal storage capacity.^(a) The surface conductances selected for the slabs were $0.72 \text{ Btu/h}\cdot\text{ft}^2\cdot^{\circ}\text{F}$ for the slab top and $1.00 \text{ Btu/h}\cdot\text{ft}^2\cdot^{\circ}\text{F}$ for the bottom. These values are representative of typical carpet installations.

4.2 BSTS SYSTEM MODEL

In incorporating the BSTS system model into the building model, the existing base building subsystems and structure were modified as little as possible for the following reasons:

-
- (a) Optimization studies that looked in more detail at the sensitivity of BSTS performance to changes in slab characteristics are described in Section 5.2.2.

- to maintain the integrity of the base building HVAC system design - This was a "design" decision, as it was felt a real building owner would want the capability of running the HVAC system alone.
- simplicity in modeling
- interpretation of results - Tracing the cause and effect of various system parameter modifications becomes more difficult with each change in the basic model.

Thus, the existing HVAC subsystems and controls were left essentially intact. The thermostats were identical to those of the base case for each of the controlled zones. The HVAC water loop flow rate was the same as in the base case, as were boiler, cooling tower, and heat pump capacities. The primary change to the HVAC water loop was the additional secondary piping loop, which was modeled to route water through the concrete floor of the model (see Figure 2.2).

Included in the actual model are five separate slabs, one for each of the four perimeter zones and one for the core. As is the case with the HVAC system heat pumps, matching slab models to individual zones allowed the control of the BSTS system to account for differing needs, based on location within the building. The slab water loop parallel circuits were arranged to maintain turbulent water flow (approximately 3 ft/s) at a reasonable pressure drop. The total water flow available to each slab equaled the flow rate of the heat pump associated with the same zone as that slab.^(a) Inlet water temperature to the slabs was based on the overall mixed temperature of water leaving the five zone heat pumps.

As mentioned earlier, initial BSTS system simulations used a control scheme that was simple, in terms of both simplicity in modeling and the expected difficulty of implementation in an actual building. The simple control model uses five slab water flow controllers that sense the space temperature in each of the five zones. These temperatures are compared to the water loop temperatures to determine if water flow control valves to the slabs

(a) Because heat pumps in all zones were nominally sized at 370 ft²/ton, 3 gpm/ton, water flow to the slab was uniform over the entire floor area.

should be open or closed and if the slab water pump should be on or off. The setpoints for night and weekend temperature setbacks on the zone heat pump thermostats are identical to those of the base building model for winter heating conditions. During cooling-load-dominated months, a cooling tower "free cooling" cycle is used. This cycle turns on the cooling tower whenever ambient wet-bulb temperatures are low enough so that the tower can produce cool water (i.e., relative to the zone temperatures) for use by the BSTS system. This free cooling feature was not used on weekends, because the mechanical cooling system was shut off.

The slab water flow controllers, like the zone thermostats, have both heating and cooling setpoints. These setpoints were established for the initial simulations so charging of the slabs with heat or coolth would occur before the room thermostats signaled the heat pumps to start. For example, during occupied hours the slab water flow controller setpoint for cooling was set at 77°F, while zone thermostats (as established by the base case simulation) were set at 78°F. When a zone temperature rises to 77°F, the controller checks the loop temperature to see if water below 77°F is available to help cool the zone's slab. If water temperatures are below 77°F, the controller opens the flow control valve and the slab water pump is started, routing water to that zone's slab. The block diagram in Figure 4.2 summarizes the simple BSTS system control scheme.

The establishment of appropriate slab flow valve controller setpoints for the simple control scheme is essential for the design. First, to provide any benefit at all, the setpoints must lie within the deadband of the zone thermostats. A slab water flow controller cooling setpoint for a zone that is higher than that zone's heat pump thermostat cooling setpoint would be useless, because the zone heat pump would start first, adding heat to the hydronic loop, potentially raising water temperatures to above zone temperatures, making them unsuitable for use by the slab.

Another concern with the simple control strategy relates to the thermal capacitance of the slab. The simple control considers the slab as an essentially instantaneous heat exchanger for cooling or heating the zone. However, the slab's response is actually slowed by its large mass and thermal

FIGURE 4.2. Simple BSTS System Control Scheme

properties. Thus, although the controller directs cool water to the slab to help cool the space before the thermostat starts the heat pump, by the time the slab can significantly help to cool the space, the zone temperature may have increased enough so that the heat pump comes on anyway. Or, the zone temperature may drop of its own accord as outside air temperatures drop or interior loads decrease, so that the coolth transferred to the slab is no longer needed, or perhaps even serves to overcool the space so heating is required because of BSTS system operation.

Finally, with the thermal storage slab coupled to the conditioned space, the allowable range of slab temperatures was dictated by occupant thermal comfort considerations. Hence, the surface temperatures played an important role in the feasibility of the integration control strategy. Two comfort criteria related to surface temperatures were identified (Olesen et al. 1989):

- The mean radiant temperature (MRT) can be as important as air temperature in affecting thermal comfort and is a direct function of both the floor slab and wall surface temperatures. Because of this, an additional TRNSYS component was developed by PNL researchers to monitor MRT in zones during simulation of the building model. This module, along with results for some test simulations, is described in Appendix D.
- For people wearing appropriate indoor footwear, the floor temperature should be maintained between 65 and 84°F.

4.3 BSTS SYSTEM PERFORMANCE WITH SIMPLE CONTROL SCHEME

4.3.1 BSTS System in Seattle

The BSTS system model with simple control was extensively tested using Seattle weather data to provide a basis for determining whether to continue simulations in other locations. Trends in energy consumption of the BSTS system operation were found at this stage, which included partial-year simulations and full annual simulations. The partial-year simulations used portions of the January and July months, with which trends in peak heating and cooling operation were found. In addition to confirming functionality of the BSTS system and providing some insight into the optimal application of the simple control scheme for the annual simulations, these simulations were also

used to identify important parameters and system characteristics that could be addressed during the more detailed optimization studies that followed later (see Section 5.0).

Simulation results indicated noticeable energy-conserving patterns for the BSTS system in Seattle. The total predicted annual HVAC energy consumption for the BSTS building was 572,470 kWh, as compared to 612,459 kWh for the base case, a savings of over 7%.

Comparisons of predicted monthly total energy consumption for the base case building and the building with the BSTS system are shown in Figures 4.3 through 4.6. On a monthly basis (Figure 4.3), HVAC energy savings were found in every month of the year except May. Heat pump energy consumption was slightly increased from November through March, but significantly decreased in May through September (Figure 4.4). The boiler used less energy in each of the winter months, but was also now needed in each of the summer months, whereas for the base case, summer boiler energy consumption had been zero (Figure 4.5). Cooling tower energy consumption for the BSTS system is much higher than the base case for most months of operation, as shown in Figure 4.6. However, as was illustrated in Figures 3.8 and 3.9 for the base case building, cooling tower energy consumption is normally a small part of the total HVAC energy consumption. And, as is described further in Section 4.3.2, it was found that the free cooling cycle of the cooling tower provided an economical cool water source for the BSTS slabs during cooling-load-dominated months with low ambient wet-bulb temperatures. Thus, increases in cooling tower energy consumption patterns are acceptable.

4.3.2 Diurnal Response of BSTS System with Simple Control Scheme

Although annual and monthly energy consumptions were used for comparison of building performance in this work, important insight was gained into the operation of the BSTS system by also studying temperature and heat transfer profiles. West zone temperature and heat transfer rate histories are graphically portrayed in this section for both the base and BSTS building models.

FIGURE 4.3. Total HVAC Monthly Energy Consumption for Simulated BSTS Strategies, Seattle

FIGURE 4.4. Heat Pump Monthly Energy Consumption for Simulated BSTS Strategies, Seattle

FIGURE 4.5. Boiler Monthly Energy Consumption for Simulated BSTS Strategies, Seattle

FIGURE 4.6. Cooling Tower Monthly Energy Consumption for Simulated BSTS Strategies, Seattle

In each temperature history figure, the applicable zone temperature (T_{west}), bulk temperature of the slab adjacent to the zone^(a) (T_{slab}), outdoor air ambient (T_{amb}), and water loop temperature leaving the heat pumps (i.e., available to the slabs) (T_{wtr}) are presented. In the figures illustrating the heat transfer rate histories, the heat transfer rates from the top of the slab to the zone (Q_{top}), the bottom of the slab to the plenum (Q_{bom}), and from the slab to the water loop piping (Q_{pipe}) are presented.

The west zone temperature and heat transfer histories for the base building during the fourth week of July (Sunday - Saturday) are shown in Figures 4.7 and 4.8, respectively. Outside ambient temperature ranges have daily, periodic swings, with highs between 80°F and 85°F during daylight hours and lows between 55°F and 60°F at night. Zone temperatures hover at the zone heat pump thermostat cooling setpoint of 78°F during the weekdays. Heat pump operation during the weekdays is evident because the zone temperature has fluctuated throughout this portion of the week. Zone temperatures sharply increase during the day on Sunday and Saturday when mechanical cooling is turned off.

The base building west zone slab bulk temperature has periodic daily swings, rising during the day and decreasing at night. As the working week progresses, its mean daily temperature slowly drops toward the zone temperature. However, even on Friday its temperature is still a few degrees warmer than the zone. Hence, the slab imposes a net cooling load on the space^(b) (see also Figure 4.8). During the weekend when mechanical cooling is shut off, slab temperatures increase during the day, but lag behind zone temperatures because of the thermal capacitance of the slab.

-
- (a) The bulk slab temperature is computed in subroutine THESIS (see Appendix C) by taking the arithmetic average of each discretely defined node in the computational model.
- (b) Note that the instantaneous heat transfer does go negative during early occupied hours each day, indicating that heat is being transferred from the zone to the slab. This occurs because slab surface temperatures fall during the night and then lag behind zone temperature increases during the day.

FIGURE 4.7. West Zone Temperature History for the Base Building During the Fourth Week of July (Sunday Through Saturday)

FIGURE 4.8. West Zone Heat Transfer History for the Base Building During the Fourth Week of July (Sunday Through Saturday)

Water loop temperatures leaving the heat pumps in the base building reflect the cooling-dominated operation of the heat pumps, rising to 100°F when heat pumps are rejecting heat to the circuit but always staying above 80°F (recall that the cooling tower is set to maintain 85°F loop water temperatures in the base building).

In striking contrast to the base building are the temperature and heat transfer responses of the BSTS building west zone, shown in Figures 4.9 and 4.10. Operation of the BSTS system is evidenced by the large positive heat transfer rate from the slab to the slab water piping that occurs during each weekday night (Figure 4.10). Loop water temperatures at night become cool (approximately 65°F) due to the tower free cooling cycle, which operates quite effectively in the relatively cool Seattle nights. The cooled water in the loop is used at night to cool the slab about 4 to 5°F nightly. The slab bulk temperature is now always lower than the west zone temperature during the week, and thus can be seen to reduce the zone's net cooling load. Zone temperatures drop below the heat pump thermostat cooling setpoint at night due to the cooler slab during the week, reducing the overall requirements for mechanical cooling during this cooling-dominated period.

One limitation of the BSTS simple control strategy that was observed in Figure 4.10 was a significant negative heat transfer from the slab to the slab water piping during the morning hours (around 7:00 A.M.). This occurs because, as the building starts up, an increasing cooling load is met by heat pump operation, which gradually raises loop water temperatures. During the transitory period when water temperatures are above the slab temperatures but still cooler than the zone temperature, the BSTS controller routes water to the west zone slab, partially rewarming the slab. This negates some of the benefit obtained through slab cooling at night. Eventually, loop temperatures rise above the west zone temperature, the BSTS controller shuts off water flow to the west slab, and Q_{pipe} drops to zero.

A second, more serious, limitation of the BSTS simple control strategy was BSTS slab overcooling during intermediate months (see also Section 4.2). The following describes the simulated west zone temperature and heat transfer

FIGURE 4.9. West Zone Temperature History for the BSTS Building with Simple Control During the Fourth Week of July (Sunday Through Saturday)

FIGURE 4.10. West Zone Heat Transfer History for the BSTS Building with Simple Control During the Fourth Week of July (Sunday Through Saturday)

responses for the third week of May (Tuesday through Monday), a week during which a significant amount of slab overcooling was found to occur.

West zone temperatures during this week are shown in Figure 4.11 to fluctuate between the heat pump cooling thermostat setpoint of 78°F during the day and the heat pump night setback heating temperature setpoint of 66°F at night, tracking the large daily ambient temperature swings. Both heat pump heating (at nighttime) and cooling (during daytime) occur Tuesday through Thursday. Ambient temperatures gradually increase over the week, raising the total daily cooling load while reducing heating; by Friday, the amount of heat pump daytime cooling operation has increased noticeably, and no nighttime heat pump heating occurs.

Along with this ambient temperature effect, the zone load is also impacted by the BSTS system. On Tuesday and Wednesday (the first two days shown in Figure 4.11), BSTS operation is continuous during most of the daytime hours after west zone temperatures rise above the BSTS setpoint temperature. Water loop temperatures are maintained by the cooling tower's free cooling cycle well below the west zone temperature, but fluctuate between heating and cooling the slab (Figure 4.12) in a manner similar to those observed in Figure 4.10 for July. Because the BSTS system is able to maintain slab temperatures at approximately 67°F throughout Tuesday and Wednesday, cooling loads are reduced. However, this also causes zone temperatures to drop more quickly at night.

Thus, the question arises as to whether it is better--from an energy consumption standpoint--to cool the slab continuously whenever the zone temperature exceeds the BSTS setpoint, to maximize the reduction in cooling load, or to find a way to limit the slab overcooling this causes, to avoid later heat pump heating. Because this dilemma was caused by the simple control's inability to account for the slab thermal capacitance, a modified BSTS control strategy was developed during parametric simulations (see Section 5.2.1) that accounts for slab bulk temperature when making logic decisions.

FIGURE 4.11. West Zone Temperature History for the BSTS Building with Simple Control During the Third Week of May (Tuesday Through Monday)

FIGURE 4.12. West Zone Heat Transfer History for the BSTS Building with Simple Control During the Third Week of May (Tuesday Through Monday)

4.3.3 Effects of Climate on Building Structural Thermal Storage System Performance

Additional simulations modeling the same building were performed using TMY weather data from five other cities: Los Angeles, Houston, El Paso, Washington, D.C., and Milwaukee. These cities were chosen because of their unique climatic conditions compared to the temperate climate found in Seattle. Specifically, Milwaukee has a predominantly cold climate, El Paso and Los Angeles are hot and dry, and Houston is hot and humid, while Washington, D.C., is hot and humid in summer and cold in winter. Together, these cities typify fairly well the range of climatic conditions found across the United States. Thus, performing the simulations in each of these locations provided quantitative insight into where a BSTS system can be most effectively applied.

The approach used for assessing the potential effectiveness of the BSTS system in the five different locations was as follows. First, the Seattle base case building model was simulated for an entire year in a new location with no other changes. The capacities for the boiler, cooling tower, and heat pumps were then adjusted if necessary so that zone temperature setpoints could be maintained during peak load conditions, and the simulation was rerun. Next, the BSTS system used for the Seattle simulation (reported in Section 4.2) was incorporated into this adjusted base case model for the new location. Simulations were then performed in the new location for short winter and summer peak load periods to establish BSTS simple control temperature setpoints. Finally, an annual simulation of the building with this modified BSTS system was performed, and these results were compared to the base case building annual simulation results in the new location.

Figures 4.13 through 4.17 show comparisons of the total simulated HVAC monthly energy consumption for the base building in each of the five new locations to that of the same building with the BSTS system. In Figure 4.18, the total HVAC annual energy consumption is compared for all of the locations. Additional comparisons are shown in Appendixes E and F for individual HVAC subsystem components.

The energy-conserving effects of the BSTS system were found to be most pronounced in the hot, dry climates. El Paso and Los Angeles benefitted the

most, with annual HVAC energy savings of approximately 15% and 11%. A substantial fraction of these energy savings occurred during the May through September cooling period. Although also a hot climate, Houston did not fare as well because the higher wet-bulb temperatures found in Houston limited the ability of the cooling tower to reject heat; thus, water loop temperatures were higher, and the performance of the BSTS system was decreased. Energy savings in Washington, D.C., were low because of cold winter conditions and high summer wet-bulb temperatures. The building with the BSTS system showed very small savings in the cold climate of Milwaukee, achieving on the order of 1.5% annual HVAC energy savings. The BSTS system was found to actually increase energy consumption in winter months for Milwaukee and Washington, D.C. This increase occurs because, although the BSTS system reduces the core cooling loads, this heat source is also no longer available to offset perimeter heating loads. Thus, the boiler, a less energy-efficient heat source than the core heat pumps, was required more frequently.

FIGURE 4.13. Total HVAC Monthly Energy Consumption for Base Case and Simulated BSTS System Strategies, Los Angeles

FIGURE 4.14. Total HVAC Monthly Energy Consumption for Base Case and Simulated BSTS System Strategies, El Paso

FIGURE 4.15. Total HVAC Monthly Energy Consumption for Base Case and Simulated BSTS System Strategies, Houston

FIGURE 4.16. Total HVAC Monthly Energy Consumption for Base Case and Simulated BSTS System Strategies, Washington, D.C.

FIGURE 4.17. Total HVAC Monthly Energy Consumption for Base Case and Simulated BSTS System Strategies, Milwaukee

FIGURE 4.18. Total HVAC Annual Energy Consumption for Base Case and Simulated BSTS System Strategies

5.0 BSTS BUILDING MODEL PERFORMANCE OPTIMIZATION STUDIES

The simulation results described in Sections 4.3 for the BSTS system with simple control suggest that energy can be saved by integrating the structural mass and a WSHP system in a typical office building. This section highlights followup research work performed to obtain a better understanding of the operation of the integrated systems and the effect of key parameters on performance, and to estimate the maximum energy savings that can be obtained through optimization. Section 5.1 describes the general methodology used for the optimization studies. Section 5.2 describes the specific parametric studies actually performed.

5.1 GENERAL METHODOLOGY

Various strategies were tested using a series of parametric simulations, beginning with the building model with the BSTS system and simple control described in Section 4.2. The general procedure was similar to that employed during calibration of the base case building: key parameters of the BSTS system were individually modified in each climate zone model and tested in short "screening" simulations. Results of these simulations were then analyzed to gauge the impact of the parameter on energy consumption and system response. Finally, a series of simulations to find the most favorable configuration was performed.

Because simulations of the BSTS system were computationally highly intensive, it was impractical to perform full-year runs of the model for all of the initial screening simulations. Therefore, typical weather days for each month of the year in the six climatic regions were developed (see Appendix G) from the TMY data, and the "screening" parametric simulations were limited to these twelve days of the year. When results (in terms of energy consumption) differed significantly from similar "average" day simulations of the BSTS system model with simple control (described in Section 4.0), additional year-long simulations were performed to more accurately quantify the impact.

5.2 BSTS SYSTEM PARAMETRIC STUDIES

The key parameters of the BSTS system were divided into three categories for the parametric studies: BSTS control modifications (Section 5.2.1), slab configuration modifications (Section 5.2.2), and water loop configuration modifications (Section 5.2.3).

5.2.1 Control Strategies

The BSTS control strategy parametric studies included optimization of BSTS simple control, development of BSTS slab temperature control, and optimization of BSTS slab temperature control through predictive methods to determine temperature setpoints. These efforts are described in the following sections.

5.2.1.1 BSTS Simple Control Optimization

As was described in Section 4.3, the integration of the BSTS system with a WSHP system reduces the overall cooling load requirements in the test building model, thus substantially reducing annual heat pump operation. To implement the BSTS system, a simple control scheme was developed by which water from the primary loop is routed to a slab when 1) its adjacent zone temperature reaches or exceeds either the high- or low-temperature limit of the BSTS control deadband, and 2) the water temperature is either higher or lower than the zone air temperature, as applicable. This deadband lies within the heat pump thermostat deadband so that use of the BSTS system will precede operation of the heat pumps.

Observations of the diurnal response in Seattle (see Section 4.3.2) indicate that the BSTS system could more effectively be controlled if heating and cooling setpoints were varied on a monthly, weekly, or daily basis, based on the expected load patterns for the building. Therefore, initial attempts to improve performance of the BSTS system focused on manually readjusting these setpoints in an intuitive/iterative manner to obtain improved system performance for a few selected average-weather days of the year in the different cities. For example, for an average July day in El Paso, a hot day is forecast, yet nighttime wet-bulb temperatures are low. Precooling the building in this instance by using the cooling tower free-cooling cycle to

produce cool water for the slabs would reduce heat pump cooling operation during the day without a significant danger of overcooling.

Results of these initial parametric studies indicated additional energy-saving potential in this strategy, but also strongly pointed out a primary limitation of the simple control. That is, by trying to control the slab like a simple heat exchanger between the water loop and the zone, the control ignores the time-delayed response of the BSTS system caused by the thermal capacitance of the slab. As hypothesized (Section 4.2) and later confirmed (Section 4.3.2), this limitation often resulted in overcooling of slabs. Overcooling occurs when slab bulk temperatures are reduced such that, although the slab first helps reduce zone cooling load, mechanical heating is needed later to help offset what eventually becomes a heating load. A clear example of this in the earlier simulations occurred in Seattle in May (Section 4.3.2). Although the BSTS system reduces heat pump cooling loads by 50%, the overcooled slabs 1) increase the demand for heating and 2) shift the building's heating/cooling requirements such that boiler operation is required to maintain the minimum allowed loop temperatures, because less heat is now rejected into the circuit.

5.2.1.2 BSTS Slab Temperature Control

To remove the energy penalty associated with slab overcooling phenomena, an optimal control strategy was developed in which the slab temperature rather than the zone temperature became the control parameter. For this strategy, when water is available at the correct temperature in the primary loop, it is routed to the slabs to help maintain the slab bulk temperature at a predetermined setpoint. For example, in times of high cooling loads, the setpoint would be set automatically to near the zone thermostat heating setpoint; when high heating loads are expected, the setpoint is reset to be near the zone cooling setpoint. By controlling the slab bulk temperature to normally stay within the zone thermostat deadband, the BSTS system cannot accidentally overcool/heat (i.e., overcharge) the slab. On days with high cooling loads, this restriction can be relaxed to maximize cooling potential. It is also important to reemphasize that the slab bulk temperature control works to maintain a given setpoint, but is not always successful, because primary loop

water temperatures may be inappropriate. For example, during periods of extreme cold, all heat pumps draw heat from the water loop, causing temperatures to drop well below desired zone conditions.

Coupled with the slab bulk temperature control would again be the cooling tower free-cooling control. During cooling-dominated days, the free-cooling controls would turn on the cooling tower whenever the outside air wet-bulb temperature is low enough that water can be cooled by the tower to below slab bulk temperatures--provided that the loop water temperatures are not already low enough that boiler operation could result if the water is still further cooled (although, of course, the boiler low water temperature setpoint could be dropped in summer months to avoid this).

Note that overheating of the slab is not a problem with either the BSTS slab temperature control or the BSTS simple control for two reasons. First, there is no free-heating source (i.e., the electric boiler coefficient of performance (COP) = 1.0). Second, during intermediate seasons where zones in the building are in both heating and cooling modes, it is far more beneficial to optimize BSTS operation to reduce cooling loads. Active control of water loop temperatures is thus limited to 1) maintaining the 60/85°F limits common to normal WSHP systems and 2) a free-cooling low water temperature cutoff.

To assess the effectiveness of the slab temperature control strategy, parametric simulations of the slab temperature control with free-cooling strategy were made using the average-day weather data for the six cities. These simulations made it possible to establish functionality of the strategy and determine some rough, seasonally optimized control setpoints. Full-year simulations were then performed. Annual energy consumption for each city using these seasonally optimized control setpoints are compared to the BSTS system with simple control in Table 5.1. Improvements in performance using the BSTS slab temperature are observable, but are quite small--on the order of 1%.

The diurnal response characteristics of the BSTS building with slab temperature control contrast with those of the BSTS building with simple control. Shown in Figures 5.1 and 5.2 are the west zone temperature and heat transfer histories for the third week of May (Tuesday through Monday) in

TABLE 5.1. Comparison of Total Annual HVAC Energy Consumption for Base Building (Model 1), BSTS Building with Simple Control (Model 2), and BSTS Building with Slab Temperature Control (Model 3)

<u>Climate Zone</u>	Building HVAC Energy Consumption, kWh		
	<u>Model 1</u>	<u>Model 2</u>	<u>Model 3</u>
El Paso	594,327	506,814	501,208
Houston	744,874	716,832	710,217
Los Angeles	463,337	413,072	411,265
Milwaukee	1,018,760	1,003,522	994,797
Seattle	612,459	572,470	556,584
Washington, D.C.	756,298	730,847	715,847

Seattle for the BSTS building with slab temperature control. These correspond in simulation time and location to Figures 4.11 and 4.12 for the BSTS system with simple control. With slab temperature control, heat transfer from the slab to the slab piping (Figure 5.2) is always positive during times when the BSTS circuit is functional. Consequently, the inefficiency associated with accidental partial rewarming of the cooled slab is avoided. Cooling of the slab by the BSTS system is limited so that the slab bulk temperature never drops below the zone thermostat heating setpoint of 71°F. Thus, the slab helps maintain zone temperatures above the night setback level during the nighttime. Nighttime operation of the heat pumps is avoided Tuesday through Thursday; with the simple control (Figure 4.11), it was required. On the other hand, because slab temperatures are not cooled further, they have less potential for offsetting the cooling load; thus, the west zone temperatures rise to the cooling setpoint earlier each morning and drop off later in the afternoon. Although this indicates that more heat pump operation is occurring than was the case with simple control, the total predicted overall HVAC energy consumption in May is reduced by nearly 17% because operation of the electric boiler is minimized.

5.2.1.3 Predictive Control of Slab Temperature Setpoint

Although the slab temperature control strategy showed little energy savings over the simple control strategy, it did show better operational

FIGURE 5.1. West Zone Temperature History for the BSTS Building with Slab Temperature Control During the Third Week of May (Tuesday Through Monday)

FIGURE 5.2. West Zone Heat Transfer History for the BSTS Building with Slab Temperature Control During the Third Week of May (Tuesday Through Monday)

characteristics, so it was used for an additional parametric study of control. In that study, the potential impact of a predictive BSTS slab temperature control algorithm on energy consumption was considered. Such a control would adjust the slab temperature control setpoint based on expected loads in the space. With such a predictive control device, precooling of slabs during the night and mornings could be maximized without concern about slab overcooling. This differentiates it from the nonpredictive slab temperature control, where slab setpoints vary only with season, and these are maintained within the zone thermostat setpoint deadband.

Because development of a TRNSYS control component to implement predictive control was beyond the scope of this work, optimal daily setpoints for average-day months were iteratively determined from additional parametric simulations, using knowledge about that day's expected weather to make "smart" (i.e., by way of changes in the input) control decisions. These so-called optimized results were compared to average-day runs using the nonpredictive model with the seasonal setpoints previously chosen for the annual simulations (Section 5.2.1.2). In almost all cases, improved BSTS system performance was obtained using setpoints optimized for a given day. By assuming that the relative percentage of energy savings for each average day could be obtained for that entire month with a true predictive control, annual energy consumption of the "optimized BSTS system" in each of the six cities can be estimated.

These estimated annual energy consumption values are compared to the nonoptimized BSTS slab temperature control model results and the base building in Table 5.2. In El Paso, Houston, Los Angeles, and Washington, D.C., these results show approximately a 2% improvement over the nonpredictive slab temperature control. These findings suggest that a control algorithm that controls BSTS system setpoints by predicting future building loads (given forecasted weather data for 24 hours in advance) would be valuable.

5.2.2 Slab Configuration

The motivation behind the slab parametric studies was to find those properties and characteristics that maximize the use of thermal storage and

TABLE 5.2. Comparison of Total Annual HVAC Energy Consumption for Base Building (Model 1), BSTS Building with Slab Temperature Control (Model 3), and BSTS Building with Optimized Slab Temperature Control (Model 4)

<u>Climate Zone</u>	Building HVAC Energy Consumption, kWh		
	<u>Model 1</u>	<u>Model 3</u>	<u>Model 4</u>
El Paso	594,327	501,208	487,718
Houston	744,874	710,217	695,121
Los Angeles	463,337	411,265	404,403
Milwaukee	1,018,760	994,797	993,619
Seattle	612,459	556,584	550,695
Washington, D.C.	756,298	715,847	700,184

exchange in a way that produces the most energy-conserving design. This involves finding the best tradeoff between the following essentially competing objectives:

- to use the slabs as heat exchangers for transferring thermal energy quickly between the hydronic piping loop and adjacent zones - Speed of this exchange is important because the heat transfer should be accomplished before zone temperatures rise/fall enough to require the heat pumps to operate.
- to use the slabs' thermal capacitance to delay the transfer of thermal energy between the hydronic loop and the zone, so that it can be stored when it is available and released at some later time when it is needed - Maximizing storage in a given mass of slab also becomes important, given this objective.

Combinations of some of the slab's thermal properties provide information about how the slab will behave as an energy storage/transfer device. These include

- the thermal diffusivity, α - This is an indicator of the speed at which a temperature profile across the slab's cross section can develop.
- α and the slab thickness - Both of these properties directly affect the time required for a change in slab temperature.
- the ratio L^2/α , where L is the slab thickness - This indicates the time required for a wall to reach a steady-state temperature in

response to a ramp change in boundary temperature. The greater the value of L^2/α , the greater the time required.

- the product of the density and the specific heat - An indicator of the thermal storage capacity per unit volume of mass.

Within the context of the complex conditions that actually exist within a building, two primary factors must be considered to maximize thermal storage benefits:

- variation in ambient environmental conditions within a 24-h daily cycle - In cases where utilization of ambient conditions at night (say, via the free-cooling cycle of the tower) would be beneficial when used during the day, a 12-h lag in response is nearly optimal.
- extent of variation in conditions between zones (e.g., core and perimeter) that occurs, such that energy in one zone could be used beneficially if transferred via the water hydronic loop to the slabs in other zones.

Over a year, both factors could come into play, in addition to large amounts of time when the BSTS system cannot be used at all. Slab parameters should be selected to not have a negative impact during these times. Thus, the optimum slab configuration is determined by a complicated tradeoff.

5.2.2.1 Simulation Descriptions

The impacts on BSTS system performance of a number of slab parameters were examined through an extensive simulation study. Parameters that were varied include

- location - Six cities/climates were again simulated.
- slab thickness - Simulations were completed in which slab thickness was varied from 6 to 10 in.
- thermal diffusivity - Simulations were performed in which α was increased by a factor of 10 over the value used for the standard concrete material by modifying the thermal conductivity from 1.0 Btu/h·ft·°F to 10.0 Btu/h·ft·°F.
- surface conductance - Slab top and bottom surface conductances were increased by a factor of 20 (same as floor tile) to increase heat exchange between the slabs and the zone and plenum air. Additional parametric simulations were also completed in which the surface conductances were set to zero to observe storage behavior when the slabs were isolated from the zone spaces.

- thermal storage capacity per unit volume - This was increased by using higher values for the slab density and specific heat. This incidentally also decreased α .

For the simulations, the building model with the BSTS slab temperature control with seasonal setpoints was used as the benchmark against which comparisons were made.

5.2.2.2 Results

Increasing the slab thickness from 6 in. to 10 in. had, overall, a negligible impact on building energy use. The only change of any significance occurred during average-day simulations of colder winter months. In the coldest climate (Milwaukee), the 10-in. slabs decreased the daily predicted energy consumption by 3.3% from runs with the standard 6-in. slab when using January average weather data. However, because the BSTS system is not used to any great extent during the winter months, this result is likely simply the effect of passive thermal storage on temperature recovery from night setback of zone thermostats.

Sensitivity runs where slab thermal conductivity was increased by a factor of 10 also showed little change when compared to simulations using the unchanged conductivity.

Of more interest was the impact of modifying slab surface conductances on energy consumption. During summer months when BSTS operation normally offers substantial performance improvements, the predicted total daily HVAC energy consumption increased dramatically over the benchmark average-day BSTS simulations to which they were compared. In El Paso, daily predicted energy consumption increased by nearly 46%! This suggests that the coolth that is stored during the night and early morning when tower free cooling is being operated is released before it is needed by the zones. Thus, heat pump operation is needed earlier during the day when floor tile is used to cover the slab. This clearly points to the importance of having some thermal resistance on the slab surface to delay/decrease the discharge of coolth from the slab (i.e., the transfer of heat to the slab).

Sensitivity to increases in slab density and specific heat were observed during the winter. For example, in Seattle and Milwaukee, daily predicted

HVAC energy consumption dropped by 16% and 18%, respectively. However, because BSTS system operation is minimal during winter, these changes are again likely to be due to passive thermal mass effects (storage).

In conclusion, slab modifications overall had far less impact on simulation results than had been surmised. This may have been because those impacts that do occur over a daily cycle could be both positive and negative, offsetting each other and leading to an overall small effect on energy consumption. One exception to this was surface conductance, which strongly impacted summer BSTS system performance. Delaying/decreasing heat transfer between the zone and the slab by having some insulation (e.g., carpeting) on these surfaces improves performance when compared to a floor covered only with tile.

5.2.3 System Configuration

Two alternative hydronic loop arrangements were modeled as part of the optimization studies. The first, summarized in Section 5.2.3.1, was performed to see if BSTS system performance could be improved through a different water loop arrangement that might better take advantage of periods when the core zone is dominated by cooling loads while the perimeter zones need heating. The second, summarized in Section 5.2.3.2, was performed because an economic analysis (described in Section 6.0) indicated that this arrangement could significantly reduce the estimated incremental costs of installing the BSTS system in the test building.

5.2.3.1 Core and Perimeter Switchover Piping

In the first alternative modeled (Figure 5.3), water piping leaving the five zone heat pumps is arranged so that water leaving the larger core zone heat pump is available for use only by the perimeter BSTS slab water circuits. Likewise, water leaving the perimeter heat pumps is available for use only by the core zone slab water circuit. This configuration was simulated to see if, during periods in which the perimeter is undergoing a net heating load and the core a cooling load, heat pump operation would result in more favorable temperatures for use by the now split BSTS circuits to help cool core slabs and heat perimeter slabs. In contrast to the initial hydronic loop arrangement

FIGURE 5.3. Integrated BSTS and Heat Pump Subsystems with Core/Perimeter Piping Switchover

studied (Figure 2.2), this configuration physically would require two independent secondary water circuits and two secondary pumps instead of one, as indicated in Figure 5.3.

For the simulations, the slab temperature control with seasonal set-points (as described in Section 5.2.1.2) was employed. Thus, annual simulations results for the core/perimeter switchover hydronic circuit were compared to the simulations of the standard hydronic loop in which this control was also employed. In Table 5.3, simulation results expressed as annual HVAC energy consumption are compared to the results obtained with the nonoptimized BSTS slab temperature control model and the base building. In each climate simulated except for Los Angeles, an extremely small performance improvement (0.1% to 0.2%) occurs; Los Angeles shows a slight decrease (0.1%) in performance. Because the apparent benefit of this switchover was not only of second-order importance but also significantly increases capital costs because of the additional BSTS pumps required (see Section 6.0), this option was not analyzed further.

5.2.3.2 Series Heat Pump and Slab Piping Arrangement

The second alternative hydronic piping loop studied, shown in Figure 5.4, consists of five identical parallel water lines (one per zone), in which each zone's heat pump and BSTS slab water circuit are in series. A control valve and bypass line are also included so use of the BSTS slab water circuit will be made by the slab temperature controller. This configuration was chosen for economic reasons; it requires significantly less additional water piping for the BSTS system and no additional BSTS secondary loop pump. From a performance standpoint, however, this configuration was not expected to save as much energy as the normal BSTS configuration; the increased pumping head necessary to run water through the floor slabs is required 100% of the operating time instead of only when the BSTS secondary circuit is activated by the controller.

As was the case with the simulations described in Section 5.2.3.1, the slab temperature control with seasonal setpoints was used for this model, so comparisons are again made with simulation results for the standard hydronic loop in which this control was used. Predicted annual HVAC energy consumption

TABLE 5.3. Comparison of Total Annual HVAC Energy Consumption for Base Building (Model 1), BSTS Building with Slab Temperature Control (Model 3), and BSTS Building with Switchover Hydronic Piping

<u>Climate Zone</u>	<u>Building Total Annual HVAC Energy Consumption, kWh</u>		
	<u>Model 1</u>	<u>Model 3</u>	<u>Switchover</u>
El Paso	594,327	501,208	499,697
Houston	744,874	710,217	709,490
Los Angeles	463,337	411,265	411,769
Milwaukee	1,018,760	994,797	993,834
Seattle	612,459	556,584	555,895
Washington, D.C.	756,298	715,847	714,568

for this model in each of the six climate zones is shown in Table 5.4. As expected, the higher pumping requirements of the series hydronic loop noticeably increased energy consumption. Because of its favorable climate, the El Paso model is able to use the BSTS system extensively. Consequently, simulations of this model showed the smallest increase in pump (and, hence, total) energy consumption. Interestingly, when the energy penalty for this additional pumping head is subtracted from the total HVAC energy consumption, the differences between the two sets of simulations are again of only second-order importance in each climate zone. Because of this, the physical option of using separate BSTS pumps for each zone was considered one way to avoid this penalty, where each pump is individually activated when the BSTS controller signals it. However, although this avoids the additional pumping energy penalties, these extra pumps increased installation costs sufficiently to discourage selection of this configuration. Thus, for the economic analyses described in Section 6.0, a single, larger primary pump was used.

FIGURE 5.4. Integrated BSTS and Heat Pump Subsystems with Series Piping of Zone Components

TABLE 5.4. Comparison of Total Annual HVAC Energy Consumption for Base Building (Model 1), BSTS Building with Slab Temperature Control (Model 3), and BSTS Building with Series Hydronic Piping (Model 5)

<u>Climate Zone</u>	Building Total Annual HVAC Energy Consumption, kWh		
	<u>Model 1</u>	<u>Model 2</u>	<u>Model 5</u>
El Paso	594,327	501,208	590,961
Houston	744,874	710,217	726,597
Los Angeles	463,337	411,265	428,094
Milwaukee	1,018,760	994,797	1,014,939
Seattle	612,459	556,584	575,894
Washington, D.C.	756,298	715,847	732,978

6.0 ECONOMIC VIABILITY ASSESSMENT

Because cost-effectiveness is of critical importance in the final evaluation of any new technology, several configurations of the BSTS system were analyzed using a simple, incremental life-cycle cost analysis. The economic figure-of-merit was the discounted payback period. This approach is commonly used by actual building owners and designers; thus, the overall acceptability of results from such an analysis are likely to be high.

6.1 ENERGY USAGE AND DEMAND

Summarized in Table 6.1 is the predicted total annual energy consumption for the following five building models when simulated in each of the six cities:

- Model 1 -- base (non-BSTS) building (Section 3.4)
- Model 2 -- BSTS system with simple control (Section 4.3)
- Model 3 -- BSTS system with slab temperature control and seasonal (winter/summer) slab temperature control setpoints (Section 5.2.1)
- Model 4 -- BSTS system with slab temperature control, adjusted using monthly average-day optimized control setpoint simulation results as reducing factor (Section 5.2.1)
- Model 5 -- BSTS system with slab temperature control and seasonal control setpoints, but with "series" heat pump/slab water circuit arrangements for each zone (Section 5.2.3).

The model number designations are used here to simplify references within this section.

Electrical demand charges have become an increasingly important part of electric rate structures. Therefore, peak monthly demand charges for each city for the five models were computed by a PNL-developed TRNSYS component module (TYPE 61) during the simulations. A copy of the FORTRAN source code for this module is included in Appendix C.

The peak summer/winter month seasonal demand charges obtained using TYPE 61 are summarized in Table 6.2 for each of the five building models. Complete tables of month-by-month peak demands for each of Models 1, 2, 3,

TABLE 6.1. Summary of Total Annual Energy Consumption (kWh) for BSTS Strategies

<u>Climate Zone</u>	<u>Building Model</u>				
	<u>1</u>	<u>2</u>	<u>3</u>	<u>4</u>	<u>5</u>
El Paso	594,327	506,814	501,208	487,718	509,961
Houston	744,874	716,832	710,217	695,121	726,597
Los Angeles	463,337	413,072	411,265	404,403	428,094
Milwaukee	1,018,760	1,003,522	994,797	993,619	1,014,939
Seattle	612,459	572,470	556,584	550,695	575,894
Washington, D.C.	756,298	730,532	715,847	700,184	732,978

TABLE 6.2. Peak Summer/Winter Electrical Demand (kW) for BSTS Strategies

<u>Climate Zone</u>	<u>Building Model</u>							
	<u>1</u>		<u>2</u>		<u>3</u>		<u>5</u>	
	<u>Summer/Winter</u>		<u>Summer/Winter</u>		<u>Summer/Winter</u>		<u>Summer/Winter</u>	
El Paso	321	733	321	732	320	733	319	733
Houston	418	647	417	649	417	647	417	646
Los Angeles	319	307	321	709	318	530	318	530
Milwaukee	337	856	612	856	337	855	336	855
Seattle	321	739	632	816	315	734	315	734
Washington, D.C.	382	776	382	776	383	776	382	780

* Model 4 was assumed identical to Model 3.

and 5 and each climate are given in Appendix H. Differences in peak demand among the models appear insignificant at first glance, a result that is not surprising when one considers the observed response characteristics of the building. For example, peak load periods in winter, i.e., morning warmup, require operation of all heat pumps and the electric boiler, regardless of the BSTS system. Likewise, although weekday morning heat pump operation is often reduced in summer by BSTS system operation, late afternoon peak cooling loads

are large enough that operation of all zone heat pumps at some point to maintain zone temperatures is still required.

However, one significant exception to this uniformity in seasonal peak-demand loads occurs in Los Angeles, where winter peak loads are influenced by the electric boiler operation when the BSTS system is used, yet no boiler operation occurs in the base building. To understand this, we must first recognize that cooling/heating loads are nearly balanced, an ideal situation for a normal WSHP system, because heat rejected by zones in the cooling mode (typically the core) can be used to heat zones in the heating mode. When a good balance between the two exists, neither the boiler nor the cooling tower needs to operate. Reduction of the net cooling load by the BSTS system in this scenario can upset this balance, resulting in less heat pump rejected heat being available in the water loop to help satisfy heating demand. Thus, water loop temperatures drop far enough that boiler operation is required. In the case of the BSTS system with simple control, the winter demand peak is 709 kW in Los Angeles because of boiler operation; for the base building, it is only 307 kW.

The month-by-month peak demands (Appendix H) indicate that this is a fairly common problem in mild weather months when heating and cooling loads are nearly balanced, particularly for the BSTS system with simple control (Model 2). For the mild Seattle climate, this problem even carries over to summer, where the summer peak of 321 kW for the base building jumps to 632 kW. Thus, the BSTS system, though a thermal storage system, is not useful for demand reduction and, unless properly controlled, increases demand during milder months because electric boiler operation is required.

6.2 COST ESTIMATING

Additional capital equipment required for each of the BSTS system configurations described above (as compared to the non-BSTS base building) is listed in Table 6.3. As shown schematically in Figure 2.2, the BSTS system water piping for Models 2 through 4 consists of a secondary piping loop (assumed one per floor) with a separate pump that uses as supply water the

TABLE 6.3. Capital Equipment Requirements for BSTS System

	<u>Building Model</u>				
	<u>2</u>	<u>3</u>	<u>4</u>	<u>5</u>	<u>6</u>
Water Side					
Secondary Loop (Schedule 40 steel)	X	X	X		
Pipe Fittings	X	X	X	X	X
Valves (include control-type)	X	X	X	X	X
Pumps	X	X	X		
Controls					
Slab Resistance Temperature Detectors (RTDs)		X	X	X	x(a)
Pump Controls	X	X	X	X	
Outdoor Air Sensors				X	
Water Loop Temp Sensors			X		
Cooling Tower Free Cooling	X	X	X	X	X
Personal Computer/Software	X	X	X	X	X
BSTS Circuit					
Plastic Tubing	X	X	X	X	X
Manifolds	X	X	X	X	X

(a) Required only for one floor of building.

return from the heat pipe primary loop main on the same floor. The secondary water piping loop main is assumed to be constructed of Schedule 40 black steel piping. Depending on a control signal to a three-way control valve, this main feeds water through either a bypass loop or to manifolds that direct the water to the 1/2-in. plastic tubing loops embedded in the floor slabs.

Approximately 90,000 ft² of tubing are required for the entire six-story building. Separate, parallel BSTS water piping loops were used for each building zone.

The alternative hydronic loop piping system design (Model 5) uses considerably less steel piping than the BSTS system Models 2 through 4, because zone heat pumps and slabs are connected in series with a bypass. Thus, no secondary piping circuit mains are required, but piping from individual heat pumps to the floor is needed. No BSTS pumps are needed for this configuration, but, as described in Section 5.2.3.2, an energy penalty in pumping was included in the energy analysis because of the higher total primary loop pressure drop.

The BSTS slab temperature control used in Model 3 includes the following on each floor of the building:

- resistance temperature detectors (RTDs) embedded throughout each zone's floor slabs to monitor slab bulk temperatures, from which an average for each zone is computed and then used by the controller to make logic decisions
- primary loop water temperature sensor to measure the temperature of water leaving heat pumps
- BSTS pump controller
- cooling tower fan and water bypass control valve for free-cooling cycle
- three-way control valve for the slab piping circuit in each zone and the BSTS water secondary loop
- monitors of cooling tower water temperatures, flows, and fan power, and controllers for the tower fan and bypass
- central control station (i.e., personal computer) and software.

The BSTS simple control (Model 2) requires no RTDs, but instead monitors zone temperatures. The more sophisticated BSTS slab temperature control with predictive capabilities (Model 4) would require the bulleted items above as well as monitors to track the temperatures of water entering and leaving the primary loop to the zone heat pumps and to measure outdoor air dry-bulb and wet-bulb temperatures.

For the economic analysis, an additional case that was not simulated was also costed. This model--designated Model 6--was developed to obtain some indication of the lowest-cost BSTS system that might still provide reasonable energy savings. Water piping for this case is in series between zone heat pumps and slabs (i.e., the same as Model 5), but it is assumed for the economic analysis that the six floors in the test building experience essentially identical load patterns. Thus, slab RTDs are needed only for a single floor of the building, and control decisions made for this floor are repeated on all six floors. Energy usage is therefore assumed to be the same as that in Model 5.

A summary of cost estimates including labor and equipment is shown in Table 6.4. Appendix I includes a copy of the worksheet with a breakdown of costs used in developing these estimates. Water piping estimates were obtained from an industry-acknowledged mechanical estimator's rule-of-thumb estimating guide (Konkel 1987) and adjusted to account for inflation and geographic variations in field labor and material costs. Incremental cost estimates for the purchase and installation of the slab water circuit hardware were obtained from a supplier of radiant floor heating systems^(a) and were not adjusted for geographic location. Control points, which include monitoring locations and controllers, were costed at \$500 per point monitored/controlled, an estimate that was based on discussions with field-experienced PNL staff. Maintenance costs of 4% of total capital equipment costs were assessed annually.

Life-cycle cost parameters used for the analysis are summarized in Table 6.5. Each of the four building ownership types (owner-occupied, state and local government, federal government, and speculative developer) was evaluated in turn. To provide the most information about the BSTS system's potential applications, average fuel prices and escalation rates for four geographic census regions (Lippiatt and Ruegg 1990) were used in the analysis instead of specific data from local power companies in the six cities. This

(a) Telephone conversation with Chuck Duman of DuAlaska, Inc., Seattle, Washington, December 4, 1989.

TABLE 6.4. Incremental Installed Cost of BSTS System (1990 Dollars)

Climate Zone	Building Model				
	2	3	4	5	6
El Paso	\$169,000	\$235,000	\$266,000	\$163,000	\$108,000
Houston	\$183,000	\$250,000	\$280,000	\$168,000	\$113,000
Los Angeles	\$225,000	\$290,000	\$321,000	\$182,000	\$127,000
Milwaukee	\$203,000	\$269,000	\$299,000	\$174,000	\$119,000
Seattle	\$210,000	\$275,000	\$307,000	\$178,000	\$122,000
Washington, D.C.	\$187,000	\$253,000	\$284,000	\$168,946	\$114,000

TABLE 6.5. Financial Assumptions of Incremental Life-Cycle Cost Comparisons of BSTS System

Parameter	Building Ownership Type			
	Owner-Occupied	State and Local Government	Federal Government	Speculative Developer
Discount Rate	13.4%	12.4%	12.4%	13.4%
Inflation Rate	5.0%	5.0%	5.0%	5.0%
Life Cycle	20.0 yr	20.0 yr	20.0 yr	20.0 yr
Property Tax Rate	2.0%	0.0%	0.0%	2.0%
Fuel Escalation Rate	3.1%	3.1%	3.1%	3.1%

was done so that the economic analysis results would be less likely to indicate conclusions as to system viability based on less-than-representative electrical rates at one location. The four census regions are shown in Figure 6.1. Average fuel prices are summarized in Table 6.6.

A PNL-developed economic forecasting tool, The Economic Analysis Model (TEAM) was used for the analysis. TEAM generates projections of the cash flows associated with constructing and operating a thermal energy storage system and calculates a variety of economic figures-of-merit to help determine the overall feasibility of the system being analyzed (Brown et al. 1987). Because of its flexibility, TEAM provided an easy means of performing the preliminary evaluations described here for which only conceptual cost and performance data were available.

FIGURE 6.1. United States Census Regions (Lippiatt and Ruegg 1990)

TABLE 6.6. Commercial Average Electricity Prices for 1990
(Lippiatt and Ruegg 1990)

<u>Census Region</u>	<u>\$/MBtu</u>	<u>Cents/kWh</u>
1	24.85	8.5
2	24.31	8.3
3	23.59	8.0
4	18.87	6.4
U.S. Average	23.11	7.9

6.3 POTENTIAL HIDDEN COSTS

In addition to the costs identified in Table 6.4, three potential incremental costs were also identified that could not be easily nor reliably quantified. These are

- commissioning costs to 1) verify that the BSTS controls are functioning correctly, 2) train permanent staff on the BSTS system, and 3) balance the BSTS water piping loop - Programming the BSTS system controller could also be part of commissioning.
- construction costs that would increase if more massive or special concrete materials were used to enhance thermal storage capacity
- additional mechanical equipment room space that will be required for BSTS controls/piping/manifolds, at the expense of usable floor space.

The design and installation of a field prototype BSTS system is needed to quantify these costs. However, it is clear that they could be significant.

6.4 RESULTS

Results of the economic evaluation were extremely unfavorable to the BSTS. Even for El Paso, where predicted energy savings were greatest and incremental installed costs of the BSTS system the least, the discounted payback period exceeded the maximum number of years (31) in the TEAM model. Even the simple payback period was greater than 31 years.

This result is rather obvious when one observes that incremental first costs range from a low of approximately \$108,000 for the El Paso Model 6 case

to over \$320,000 for the optimized BSTS slab control (Model 4) in Los Angeles, yet annual energy savings range from only \$328 (Milwaukee Model 5) to \$7036 (El Paso Model 4). When these findings are coupled with maintenance costs and replacement costs for pumps/controls after 10 years, it is clear that, given the assumptions used for this analysis, the BSTS system has little chance of being economically competitive.

The question then arises, under what set of circumstances could a BSTS system be an economically attractive option? Clearly, higher fuel costs and additional energy savings would help lower the payback period. However, although simulations suggest that an adaptive/predictive controller might help the building realize much higher annual HVAC energy savings, the premium such a capability adds to the building's construction costs more than offsets the savings for energy.

A hypothetical building case was developed to assess what set of circumstances would provide reasonable discounted payback periods for a BSTS system. A series of sensitivity calculations was performed. In one of these calculations, simple and discounted payback periods of 5 and 6 years, respectively, were obtained.^(a) The values of key parameters used for this case include

- incremental costs for the BSTS system of \$100,000 (the same as Model 6 in El Paso)
- electricity costs of \$0.15/kWh, which is in the upper range of commercial rates found in major cities
- HVAC annual energy savings obtained through use of the BSTS system amounting to 133,333 kWh (\$20,000 in year 1) - This would be the equivalent of a 22.4% reduction in annual HVAC energy usage. By comparison, simulated values of approximately 15 to 18% were obtained for the base case building in El Paso. This additional energy savings could potentially be realized through use of control optimization strategies that are learned over time by building staff (e.g., additional slab precooling in the morning without occupant complaints may be possible); however, this may be an unrealistic expectation.

(a) A rough estimate of the net present value of the project in this is approximately \$550,000 for an estimated system life of 20 years.

- 0% property tax (i.e., the building is assumed to be used as a state, local, or federal government facility)
- fuel escalation rate of 10% per year - This escalation rate is highly unlikely, for it is higher than national averages for electricity, which typically run in the fractions of one percent (Lippiatt and Ruegg 1990).

Analysis of the hypothetical building case does indicate the possibility of conditions under which the BSTS system is economically attractive. Currently, however, this is unlikely.

7.0 COMPARISON WITH OTHER HVAC TECHNOLOGIES

Because the building with the BSTS system was found to save energy when compared to a building with a conventional WSHP system, its performance when compared to other WSHP system technologies was addressed next. In Section 7.1, application of an air economizer system to the base building model is described. Section 7.2 presents a model of a conventional water thermal storage system when added to the base building. In Section 7.3, the WSHP system is replaced by a separate central water chilling and electric boiler heating systems that are connected to cooling and heating coils in zone air handling units. Finally, the economics of the three alternative technologies are discussed briefly in Section 7.4.

7.1 BASE BUILDING WITH AIR ECONOMIZER

The BSTS system takes advantage of the cooling tower as an indirect evaporative cooling device during the cooling-load-dominated season. This raised the question of how its performance would compare to the energy savings that can be obtained by instead adding a standard water or air economizer cycle in the base building.

In a water economizer system, water leaving the cooling tower is circulated through cooling coils located in the ventilation airstream upstream of the primary cooling source (in this case, the heat pumps) whenever water temperatures are lower than the supply air temperature. In an air economizer system, the amount of outside air used for mechanical ventilation of the building is increased when its energy content is lower than that of indoor air, up to a maximum of 100% of the total supply airflow.

Although both the air and water economizers should save energy if properly controlled, because the air-side economizer does not have the additional energy exchange inefficiencies that are inherent to the water economizer system (i.e., air-to-water in the cooling and water-to-air in the cooling coil), its energy-saving potential is generally the greater of the two. Thus, for the comparison presented below, the air economizer cycle was simulated.

In Section 7.1.1, the air economizer system developed for use with the base building TRNSYS model is described. Section 7.1.2 describes results of annual simulations made in each of the six climate regions and compares the economizer performance to the BSTS system model.

7.1.1 Air Economizer System Description

To help ensure that a baseline performance comparison with the BSTS system model was done impartially, the air economizer system was developed starting with the same base building model used as the starting point when the BSTS system was developed. In an actual physical system, adding an air economizer system to a WSHP system could entail significant ventilation air ducting modifications. For example, in the test building, ventilation air is supplied by a single fan system that introduces air into the plenum. For the economizer cycle, control of the outside air fraction to individual zones is required; this would entail installing dedicated air ductwork to heat pumps with control-actuated dampers to modulate the amount of outside air (up to 100% of total zonal airflow) based on zone temperatures.

Control of outside air to each zone heat pump was achieved through the use of a new TRNSYS module TYPE#70, developed by PNL researchers (see Appendix C). For this module, an enthalpy-based control scheme was employed. In enthalpy control during occupied cooling periods, the fraction of the total supply air represented by outside air is increased whenever the enthalpy of the outside air is lower than the return air enthalpy. This outside air fraction is modulated to maintain mixed air temperatures as close to the minimum design supply air setpoint of 57°F as possible, without falling below this temperature.

Because the primary energy-saving benefit of a WSHP system is its inherent heat reclamation capability, the application of an air economizer to WSHP systems is normally limited to the cooling-dominated summer months. Otherwise, using additional outside air to cool the core in winter and intermediate seasons, while reducing core heat pump operation, also reduces the amount of heat rejected to the water loop that can be used by perimeter heat pumps to offset heating loads, resulting in boiler operation.

To maximize the air economizer benefits during hot summer months when outside air temperatures are often above the room temperature during occupied hours, a night purge cycle from 4:00 a.m. to 7:00 a.m. to precool zones was also implemented. The economizer system also included an outside air ventilation fan of increased size to account for energy penalties associated with increased airflow rates.

7.1.2 Results

Annual HVAC energy consumption for each of the six climate regions is summarized in Table 7.1. Appendix J includes monthly energy consumption comparisons for the base building model, the base building with economizer, and the building model with the BSTS system and nonoptimized slab temperature control (Model 3). The air economizer was found to reduce HVAC energy consumption when compared to the base building in most months in all climates, with the majority of energy savings occurring in the cooling-load-dominated milder months.

Annual energy savings using the air economizer in the base building were greatest in Los Angeles and Seattle (17.1% and 11.7%, respectively), cities whose climates include a large number of milder, cooling-dominated days. The economizer system in both of these cities also performed noticeably better than the BSTS system with slab temperature control (Model 3), as it did in

TABLE 7.1. Comparison of Total Annual HVAC Energy Consumption for the Base Building (Model 1), BSTS Building with Slab Temperature Control (Model 3), and Base Building with Air Economizer

<u>Climate Zone</u>	<u>Building Total Annual HVAC Energy Consumption, kWh</u>		
	<u>Model 1</u>	<u>Model 3</u>	<u>Economizer</u>
El Paso	594,327	501,208	565,883
Houston	744,874	710,217	710,327
Los Angeles	463,337	411,265	384,174
Milwaukee	1,018,760	994,797	966,213
Seattle	612,459	556,584	540,527
Washington, D.C.	756,298	715,847	752,167

Milwaukee. This result is not surprising, considering that the BSTS system's primary benefit comes through use of the cooling tower, free-cooling cycle, which operates essentially as a water-side economizer system. Thus, a BSTS system suffers when compared to an air-side economizer for some of the same reasons that a normal water-side economizer does.

In contrast, the El Paso, Washington, D.C., and Houston climates appear to be better served by the BSTS system. This can be attributed to a combination of factors:

- These cities are quite hot during the day for much of the summer, precluding use of both the air-side economizer and the BSTS system's free-cooling cycle.
- The night purge of the air-side economizer is not as effective as the BSTS free-cooling cycle at precooling the building's mass during those nighttime periods when outdoor air dry-bulb/wet-bulb temperatures are low enough that these energy saving cycles can be employed.

In conclusion, the use of an all-air economizer appears to be an attractive alternative to the BSTS system in the Los Angeles and Seattle climates on the basis of energy performance.

7.2 BASE BUILDING WITH CONVENTIONAL THERMAL STORAGE

Water thermal storage in water source heat pump systems is typically considered when energy is available over a diurnal cycle, but not when it is needed most. This is true when heat is rejected from heat pumps operating in the cooling mode during the occupied daytime hours and required during nighttime hours for heating. For the test building, reduction in required boiler operation through increased storage might also help reduce electrical demand charges, an additional cost-saving benefit.

This section describes development and simulation results of a model of the base building that includes a simple water thermal storage tank with no additional controls. The model, described in Section 7.2.1, was developed to see how well the BSTS system performs compared to a WSHP system with a more conventional thermal storage system. Section 7.2.2 includes a discussion of results of annual simulations made in each of the six climate regions.

7.2.1 Thermal Storage Water Tank Model Description

Implementation of the conventional thermal storage system was straightforward and entailed modifying the base building model. Using the base building model helped to ensure an impartial baseline for later performance comparisons with the BSTS system. The primary modification to the model consisted of inserting the TRNSYS module TYPE#4, "Stratified Fluid Storage Tank," directly into the piping run leaving the cooling tower. This subroutine, included as part of the TRNSYS library, models a fluid-filled sensible energy storage tank. The tank is subject to thermal stratification based on a user-specified vertical tank segmentation (Solar Energy Laboratory 1988). Tank insulation is accounted for in this model by specifying a tank loss coefficient per unit exterior area as part of the input.

The storage tank for each of the six cities was sized as follows. The average total daily building loads in winter and summer were estimated from the heat pump energy consumption during those days in the base building. The maximum of these two loads is then used to compute the water storage required for that entire load based on 2.4 gpm/ton (equivalent to 12 gal/1000 Btu). The heat pump seasonal COPs used to back out the building loads were assumed to be 2.2 for cooling and 2.5 for heating. The computed storage tank volumes are given in Table 7.2.

No water-side controls were added to the system, making the thermal storage tank strictly an additional thermal capacitance on the system.

7.2.2 Results

Six simulations (one for each climate/city) of the base building with the water thermal storage tank were performed. The predicted annual energy consumptions are summarized in Table 7.3 for each city. Appendix K includes monthly energy consumption for the base building, the base building with the water thermal storage tank, and the building with the BSTS system and nonoptimized slab temperature control (Model 3).

TABLE 7.2. Water Thermal Storage Tank Volumes

<u>Climate Zone</u>	<u>Volume, ft³</u>
El Paso	26,490
Houston	30,700
Los Angeles	20,790
Milwaukee	20,020
Seattle	12,650
Washington, D.C.	25,020

TABLE 7.3. Comparison of Total Annual HVAC Energy Consumption (kWh) for Base Building (Model 1), BSTS Building with Slab Temperature Control (Model 3), and Base Building with Water Thermal Storage

<u>Climate Zone</u>	<u>Building Total Annual HVAC Energy Consumption, kWh</u>		
	<u>Model 1</u>	<u>Model 3</u>	<u>Water Storage</u>
El Paso	594,327	501,208	576,303
Houston	744,874	710,217	741,611
Los Angeles	463,337	411,265	463,468
Milwaukee	1,018,760	994,797	973,359
Seattle	612,459	556,584	540,763
Washington, D.C.	756,298	715,847	687,961

An examination of the energy consumption of the individual HVAC components showed that adding the water thermal storage tank had the desired impact of reducing the amount of boiler operation required during the year, but had negligible effect on other components. The annual boiler energy consumption for each simulation is compared to the base building simulation annual energy consumption in Table 7.4. The predicted total annual boiler energy consumption in Seattle and Washington, D.C., was reduced by almost 50%, resulting in net annual HVAC energy consumption savings of 11.7% and 9%, respectively, relative to the base building. In El Paso and Houston, boiler consumption was reduced to zero. However, because of the much smaller annual heating loads at these locations, this reduced total HVAC annual energy consumption is correspondingly small (3% in El Paso, 0.5% in Houston). In contrast, Los Angeles

**TABLE 7.4. Total Annual Boiler Energy Consumption of Base Building (Model 1)
Compared to Base Building with Water Thermal Storage**

<u>Climate Zone</u>	<u>Building Total Annual Boiler Energy Consumption, kWh</u>	
	<u>1</u>	<u>Water Storage</u>
El Paso	17,787	0
Houston	5,739	0
Los Angeles	0	0
Milwaukee	359,510	310,196
Seattle	137,733	75,004
Washington, D.C.	135,887	70,538

saw no net improvement when the water thermal storage tank was simulated because its winter heating requirements are small enough that no boiler is required in the base building. Finally, Milwaukee boiler operation was reduced by 12%, which resulted in a 5% decrease in total HVAC energy consumption. Reduction in boiler operation was limited because Milwaukee's much colder climate results in long periods during which cooling loads are either nonexistent or insufficient to offset building heating loads.

Because the water thermal storage tank reduces boiler operation significantly, it was also found to bring down peak demand considerably during months in which some boiler operation was previously required. For example, as shown in Table 7.5, the peak demand during each of the months November through February is essentially halved in El Paso compared to the base case. Similar results were found in other simulations, indicating that significant cost savings could be obtained using this technology in areas where utility demand charges are high.

This reduction in boiler operation is a phenomenon that is not nearly as prevalent with the BSTS system. In the BSTS system, thermal storage of rejected heat in the floor slabs is of shorter duration than in the water tank, because the floor slabs are in constant thermal communication with the adjacent zones and because they have more limited thermal storage capacity than the water tank. In addition, the available source water temperatures that are a heat source to the heat pumps may still be below the zone

TABLE 7.5. Comparison of Monthly Peak Demand for Base Building (Model 1) and Base Building with Water Thermal Storage (WTS)

	Monthly Peak Demand, kW											
	El Paso		Houston		Los Angeles		Milwaukee		Seattle		Washington, D.C.	
	Base	WTS	Base	WTS	Base	WTS	Base	WTS	Base	WTS	Base	WTS
January	733	337	654	378	283	280	855	834	739	733	776	775
February	546	249	653	420	285	284	797	800	739	734	774	776
March	302	299	355	389	325	321	733	733	739	733	775	776
April	312	310	361	401	313	311	731	725	610	300	372	523
May	316	311	369	407	319	315	322	314	314	325	365	362
June	317	314	373	408	319	314	337	322	318	311	378	368
July	319	316	373	409	319	315	329	321	321	325	381	371
August	321	317	373	410	320	317	327	321	316	323	382	371
September	319	314	373	410	318	316	321	324	314	309	380	371
October	311	312	367	404	317	314	727	317	738	316	366	353
November	539	307	357	394	298	307	733	733	739	318	775	386
December	732	326	821	393	307	298	846	815	739	734	774	631

temperature heating setpoints. Consequently, water storage in the floor slabs is inappropriate, because it would impose an additional net heating load on the space.

In conclusion, the application of a conventional water thermal storage tank to the base building reduces boiler energy consumption and demand charges during days having fairly balanced heating and cooling loads that occur at different times. This is in contrast to the BSTS system, which provides its energy-saving benefit through the free-cooling cycle during cooling-load-dominated periods.

7.3 CENTRAL HEATING/COOLING SYSTEM

As a final comparison, the WSHP/BSTS systems were replaced by separate central water chilling and electric boiler heating systems connected to cooling and heating coils in air-handling units for each zone. Initially, the intent was to compare peak demand of the BSTS system to a central system with

chilled water storage. However, because the BSTS building model was found to offer no electrical demand-side advantages (see Section 6.0) over the base building and, in some cases, was detrimental, addition of a chilled water storage tank was omitted from the central system. Instead, the analysis focused on annual energy consumption comparisons between the two systems.

7.3.1 Model Description

Development of the central system model entailed major modifications to the existing base building model. Elements that were retained from the original building included all zone models, the building load and occupancy schedules, the floor slab models (without BSTS logic), and the outside air preheater, glycol heat recovery loop, and ventilation air supply/exhaust fans.

New subsystem components were added to model the three separate water loops (chilled water, hot water, and condenser water) of the central system, and a supply air distribution system. The chilled water loop includes the centrifugal liquid chiller (TYPE#53) that produces chilled water for distribution in parallel to five cooling coil (TYPE#52), one for each of the five building zones. A flow diverter (TYPE#64) is used as a water control valve for each cooling coil, routing chilled water through the coil whenever zone temperatures exceed the cooling setpoint. Return water is mixed in a water manifold (TYPE#72) before being returned as a single flow stream to the chiller. This water manifold component was developed by PNL specifically for this application. A listing of the FORTRAN subroutine for this component is included as part of Appendix C.

The condenser water loop connects the condenser of the TYPE#53 water chiller to the TRNSYS cooling tower (TYPE#42) component (previously developed for the WSHP base building model). As was the case with the WSHP building models, the water pumps were not explicitly modeled in the system. Instead, their energy consumption was added at the end of each TRNSYS time step whenever the appropriate water circuit was operating. The chilled water and condenser water loops are shown schematically in Figure 7.1.

The hot water loop is similar to the chilled water loop, but includes an auxiliary electric boiler component (TYPE#35) (previously developed for the

FIGURE 7.1. Central System Chilled Water and Condenser Water Loop

WSHP base building model) instead of the chiller, and uses multiple heat exchanger components (TYPE#63) instead of cooling coils. The hot water loop is shown schematically in Figure 7.2.

Supply air is distributed to each zone using separate constant volume zone fans. A new PNL-developed component, TYPE#69, includes the fan and a mixing box function to mix a fixed fraction of ventilation air (drawn from the plenum zone) with return air from the zone. Mixed air then passes across the heating and cooling coils before being introduced to the zone. A typical air-handling unit configuration is shown in Figure 7.3. A listing of the FORTRAN subroutine for TYPE#69 is included in Appendix C.

7.3.2 Central System Component Sizing

Component sizing of the central system was based on the equipment sizes in the existing WSHP base building model. For example, supply air fans in air handling units were sized for the same air capacity as the zone heat pumps in the base building model. The water chiller and boiler were sized assuming (conservatively) no diversity in the load, and so their sizes were just set equal to the sum total of the nominal heat pump cooling and heating capacities for all five heat pumps, respectively. The chiller was assumed to use 0.7 kW/ton power at full load, with part-load performance based on typical operating characteristics for centrifugal chillers. The chilled water setpoint was 44°F, with a water flow rate based on a 12°F temperature rise across the chilled water coils at full load. The hot water setpoint was 120°F, with a water flow rate based on a 20°F temperature drop across the hot water coils. Water pumps were assumed to have a 50-ft head pressure drop and to be 70% efficient.

7.3.3 Results

Predicted annual HVAC energy consumption of the building with central system is given in Table 7.6 for each of the six climates. A month-by-month comparison of HVAC energy consumption of the chiller to the base building and the building model with the BSTS system with nonoptimized slab temperature control (Model 3) is given in Appendix L for each of the six cities.

FIGURE 7.2. Central System Hot Water Loop

FIGURE 7.3. Central System Air Handling Unit

TABLE 7.6. Comparison of Total Annual HVAC Energy Consumption for Base Building (Model 1), BSTS Building with Slab Temperature Control (Model 3), and Central System

Climate Zone	Building Total Annual HVAC Energy Consumption, kWh		
	Model 1	Model 3	Central System
El Paso	594,327	501,208	520,001
Houston	744,874	710,217	654,101
Los Angeles	463,337	411,265	422,214
Milwaukee	1,018,760	994,797	1,089,135
Seattle	612,459	556,584	550,555
Washington, D.C.	756,298	715,847	774,333

Energy consumption is predicted to increase over the base case WSHP system in Washington, D.C., and Milwaukee by 2.4% and 7.0%, respectively. These increases are directly related to the high winter heating loads in these climates. Recall that with the central system, rejected heat created in meeting the core cooling load is not available as a heat source for the perimeter, as is the case with a WSHP system. Thus, energy required to meet the heating load is extracted from the auxiliary boiler in the central system.

Simulations of the central system model in the other climates uniformly predict substantial energy savings over the base case WSHP system. These improvements can be attributed to the higher COP of the centrifugal chiller (approximately 5.0) when compared to the heat pumps (approximately 3.0) when operating in cooling mode. In Houston, the central system had significantly higher energy savings (12.1% versus 4.6%) than even the BSTS system, which is predicted to be relatively ineffective there because generally high ambient wet-bulb temperatures limit free cooling from the cooling tower. In contrast, the BSTS system in the drier El Paso and Los Angeles climates has higher energy savings than the central system.

In summary, for the building model studied here, use of the central system appears to offer energy benefits over the base building in warm and mild climates because of the chiller's high efficiency relative to WSHPs. Results in colder climates suggest that the WSHP system would be preferable because of its ability to reclaim heat from the building core. Finally, the limitations of this study should be recognized, i.e., that only a single central system model was used in a single building for this comparative analysis. Different building configurations, system controls, and components could bias these results significantly. However, based on energy performance, these findings do suggest that consideration of both a central and a WSHP floor-by-floor system is warranted in commercial office building applications similar to the one analyzed here.

7.4 ALTERNATIVE HVAC TECHNOLOGY ECONOMICS

A simple life-cycle cost analysis was performed to assess the economic viability of the alternative HVAC technologies described in Sections 7.1 through 7.3. Because 1) the components and configuration for these HVAC systems are well understood, and 2) the procedures used for the analysis are nearly identical to those used in Section 6.0 for the BSTS system, the objective of the following subsections is primarily to summarize results.

7.4.1 Costs

In the case of the air-side economizer (Section 7.1) and thermal storage (Section 7.2) HVAC technologies, the analyses performed were based on incremental costs over the base building WSHP system. Thus, the estimating procedures closely resemble those used in Section 6.2, with one exception. Installed costs for these cases came from Means Mechanical Cost Data (1988) rather than Konkell (1987), because the Means guidebook included needed equipment pricing not available in Konkell.

For the central system (Section 7.3), a first-order approximation of the incremental cost difference between it and the base building's WSHP system was derived from cost estimates included in a study on WSHPs (Pietsch 1988) that was performed for the Electric Power Research Institute.

A summary of the incremental installed costs of each of the alternative HVAC technologies is shown in Table 7.7. Note that an additional, significant future replacement cost is not reflected in the table that was nonetheless used in the economic analysis of the central system. The cost relates to the typical lifetimes of the different HVAC systems. Documented information on the expected average life of WSHPs is not available. However, because they use many of the same components also used in unitary air conditioners, it was assumed for this analysis that they would have a comparable life (12 years). A centrifugal water chiller, on the other hand, can be expected to have a useful life of 20 years.

TABLE 7.7. Incremental Installed Costs of Alternative HVAC Technologies

<u>Climate Zone</u>	<u>Building Model</u>		
	<u>Air-Economizer</u>	<u>Thermal Storage</u>	<u>Central System</u>
El Paso	\$163,000	\$143,000	\$79,000
Houston	\$179,000	\$164,000	\$86,000
Los Angeles	\$227,000	\$172,000	\$109,000
Milwaukee	\$193,000	\$155,000	\$93,000
Seattle	\$203,000	\$149,000	\$98,000
Washington, D.C.	\$185,000	\$160,000	\$89,000

7.4.2 Results

Due primarily to its lower incremental cost and longer useful life, the results of the economic evaluation indicated that the central system had the shortest payback period (approximately 12 years) of the three alternative technologies in all locations. This payback period is still high compared to those on which building energy conservation strategy decisions are normally made (2 to 6 years). However, it could be that the longer expected useful life of the centrifugal chiller relative to WSHPs also means it will have a higher reliability during its life. If so, this could easily explain why many building owners prefer central chilled water systems to WSHP systems.

The economic evaluation of the air-economizer and water thermal storage alternative technologies indicated a very high simple payback (>31 years) for all locations. As was the case with the BSIS systems, the net income gained due to the energy savings provided by these alternative systems was just too small to justify their high incremental costs. Thus, given the economic assumptions, neither of these options is competitive, at least for the test building.

As illustrated in Table 7.5, winter electric demand peaks are predicted to be reduced in several of the geographic locations when water thermal storage is used in the WSHP system. The economics when separate demand and use electrical charges are used was therefore studied as a separate sensitivity case to see if water thermal storage would then become more attractive as an option.

The sensitivity case used electrical rates for El Paso, which had by far the highest demand charges. The monthly general service rate structure for El Paso includes three additive components: a base monthly rate, a demand charge, and an energy usage charge. Costs for these components are as follows:^(a)

Monthly Rate - \$13.00

Demand Charge - \$14.50 per kilowatt for all kilowatts of demand

Energy Charge - \$0.01464 per kilowatt-hour for all kilowatt-hours.

Results of this sensitivity analysis indicated that a simple payback period of approximately 6 years was possible. This is low enough to confirm that, in some locations, water thermal storage is probably a viable economic option.

(a) Obtained by a telephone conversation with Janet Castle of El Paso Electric Company on December 12, 1989.

8.0 CONCLUSIONS AND RECOMMENDATIONS

The BSTS system was shown to reduce mechanical cooling loads for the building modeled in this study. When properly applied, it is a significant energy-saving technology in comparison to identical buildings without the BSTS system, with annual energy savings as high as 18%. However, in contrast to conventional thermal storage technologies, the BSTS system has a negligible impact on peak loads and thus does not reduce building peak electrical demand.

The HVAC annual energy savings predicted by simulation of the building with the BSTS system were found to be highly sensitive to climate. The BSTS system performed best in the hotter, drier climates of Los Angeles and El Paso, where summer cooling at lower ambient wet-bulb temperatures dominates building energy use. Under these climatic conditions, the cooling tower may be operated frequently to provide free cooling to the slabs. The milder Seattle climate also was predicted to have substantial energy savings. More moderate benefits were achieved in Houston and Washington, D.C. Little improvement in annual performance was predicted in Milwaukee. Thus, the benefits in cold climate regions and hot humid regions appear to be limited.

Performance of the BSTS system was found to be enhanced through optimization of controls. Overcooling of floor slabs was found to be a problem when using a simple BSTS control that responds to changing zone temperatures, because of the thermal lag caused by the slab's mass. The best-performing BSTS system appears to be one that controls the use of the BSTS system for a specific building zone slab based on the slab temperature, where the allowed slab temperature setpoint is based on the predicted load for the next day.

Other parameters were also found to have an impact on the BSTS system performance. Configuration of the hydronic piping circuit so that water leaving core heat pumps was used for perimeter BSTS slabs and vice-versa offered second-order improvements in performance. Running heat pumps and the piping through slabs in the same zone in series also had little impact on BSTS system effectiveness, but drove up pumping costs considerably. Surprisingly, modification of the slab's thermal storage characteristics had a smaller than expected impact on simulation results. This may have been because the impacts

that occur over a daily cycle are both positive and negative and may be offsetting. Slab surface conductance was found to strongly impact summer BSTS system performance. Delaying heat transfer between the zone and the slab by increasing this conductance should be pursued in an actual design.

Additional simulation studies of the existing building would be required to determine how cold the slab can be cooled during cooling-dominated periods without adversely impacting occupant comfort. The MRT model developed for this work could be used to monitor operative temperatures under the more extreme slab conditions that would result. The colder slab would further reduce zone cooling loads and potentially allow zone thermostat cooling set-points to be raised.

The initial analyses performed to assess the economic viability of the BSTS system were negative. Simply put, the cost of installing the technology appears to be far too high compared with its energy-saving potential over its life. One sensitivity case in which a very favorable set of economic conditions was used did suggest that the technology may be economically practical in a very limited number of applications. Also, the BSTS system has been modeled in only a single building configuration with a single type of HVAC system, and might have improved cost and energy effectiveness in other building designs. However, conventional energy-saving technologies for WSHP applications performed nearly as well or better than the BSTS system in simulations and though they appear to show little or no improvement over the BSTS system's economics, at least for the building studied would probably be considered first.

Finally, a central system was simulated and found to offer energy savings comparable to the BSTS system in the cooling-load-dominated climates. The cost premium of the central system was found to be lower than the BSTS system. And, though paybacks for the test building were still high relative to the base case WSHP system, it is likely that central systems are competitive in many instances, particularly when system life and reliability are important.

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APPENDIX A

METERED DATA ANALYSIS

APPENDIX A

METERED DATA ANALYSIS

The metered data for the actual base building consist of electricity consumption in amperes and kilowatt-hours for the boilers, cooling tower, heat pumps, supply and exhaust fans, lights, outlets, hot water, and elevators. Although the building's 97 unitary heat pumps operate independently, only the total heat pump electricity consumption measurement was available. In addition, no water temperatures or control signals were monitored. This data aggregation, plus the lack of any building temperature measurements, limited the analysis possibilities.

The 1985 data were analyzed because the 1983 and 1984 data sets are incomplete. In the 1985 data, there are occasional missing hours and a period of approximately 3 weeks of missing data in late November and early December. A total of 507 hours (6%) are missing. Most of the analysis was performed by directly summing various quantities such as the monthly or yearly electricity consumed by a specific piece of equipment or the number of hours of operation.

The primary end-use loads of the actual base building are heating and cooling equipment (47.9% of the total energy consumption), lights (34.4%) and miscellaneous (17.7%). The building consumed the equivalent of 73,460 Btu/yr-ft² in 1985, or a total of 1,413,900 kWh with an average monthly demand of 410 kW (Table A.1).

TABLE A.1. Seattle City Light Metered Data Analysis

SCL Building Specifications

Gross Floor Area	89,550 ft ²
Year Built	1976
Shell Materials	Concrete
Principal Use	Office

Billed Consumption - 1985

Kilowatt-hours (kWh)	1,413,900
Kilowatt-hours per square foot (kWh/ft ²)	15.8

<u>Electric End-Uses^(a)</u>	<u>kWh</u>	<u>kWh/ft²</u>	<u>Percent</u>
HVAC	649,116	7.2	47.9
Lights	466,032	5.2	34.4
Misc Equipment	<u>239,148</u>	<u>2.7</u>	<u>17.7</u>
Total	1,354,296	15.1	100.0

(a) Within the precision range of the monitoring equipment.

APPENDIX B

TRANSIENT SYSTEM SIMULATION PROGRAM (TRNSYS)

APPENDIX B

TRANSIENT SYSTEM SIMULATION PROGRAM (TRNSYS)

The Transient System Simulation (TRNSYS) Program was selected as the analysis tool for this study because it can perform a dynamic whole-building system analysis and it can be readily modified and new building subsystem component models incorporated (Solar Energy Laboratory 1988). TRNSYS was designed with a modular structure. A wide variety of component models is available in the standard TRNSYS library. For the system modeled in this project, assembling the component modules according to the TRNSYS format was a relatively straightforward procedure.

The modular nature of TRNSYS gave the program flexibility and facilitated the addition of user-formulated mathematical models (see Section 3.2 of the main report). Unlike other programs without a modular structure (e.g., DOE-2), the main program of TRNSYS itself does not have to be modified as new models are added.

Available in TRNSYS is a building load model that uses temperature level control, a dynamic modeling approach, as opposed to energy rate control used in programs such as DOE-2. Depending on the objective of the study, each approach offers different advantages.

In energy rate control, the model calculates the thermal loads based only upon the net thermal gains or losses from the space. The loads are considered to be independent of the HVAC equipment operation. The program determines the heat addition or extraction to maintain the space at specified heating and cooling temperatures (the thermal loads). An HVAC equipment model subsequently calculates the energy required to satisfy these loads. The advantage of the energy rate control is that once the building loads are determined, the researcher may reuse them for various equipment simulations without having to repeat the load calculations. However, in this type of simulation, no energy balance is ensured and much of the detailed dynamics of the equipment, the control, and the building itself are lost.

On the other hand, for temperature level control, the room condition is determined by not only the ambient conditions, but also the heating and cooling equipment inputs. The interactive feedback of the building subsystems is captured. This is a more realistic reflection of the actual thermal responses of the building. The temperature level control requires more complicated modeling schemes, including a controller to command the operation of the building system. In addition, as the building load is no longer assumed to be independent of the instantaneous inputs of its subsystems, it must be calculated for each simulation.

The system under study is complex; its performance is affected by the interactive feedback from the subsystems involved. For example, the energy performance of a water-source heat pump is strongly dependent on the room temperature and humidity, as well as on the water temperature and flow rate. This room condition, in turn, depends on the sensible and latent capacities of the heat pump and the ambient conditions, along with the thermal integrity of the building. The water loop temperature is a function of boiler and cooling tower operation and the coincident operation of other heat pumps that also take heat from or reject heat to the loop. The operation of the duct heater used to temper the ventilation air also affects the operation of the space conditioning equipment. Finally, the use of the building structure for thermal storage and the use of the water loop as its heat source and sink further complicates these interactions by affecting the building load and the water loop temperature. Because the objective of this project was to study the dynamic interaction of the building subsystems, the temperature rate control modeling approach was required. TRNSYS offered such a modeling scheme.

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APPENDIX C

TRNSYS COMPONENT MODELS

APPENDIX C

TRNSYS COMPONENT MODELS

This appendix contains code listings of those subsystem component modules developed by Pacific Northwest Laboratory researchers specifically for this study: the boiler, the cooling tower, the heat pump, the water loop thermostat (used to control the cooling tower), the floor slab, economizer controller, water piping manifold, and fan/mixing box.

Each module is written as a FORTRAN subroutine. These subroutines were linked after compilation with 1) the main TRNSYS source and 2) those additional existing TRNSYS component modules also required for the building model. The executable files TNVS created were used for the simulations described in the main report.


```

C
C *****
C *                THERESIS                *
C *  THERMAL ENERGY STORAGE IN SLAB  *
C *****
C
C      BY AMANDA CHIU
C
C *** THE MODEL ASSUMES ALL FLUID AT THE AVERAGE INLET AND OUTLET
C      TEMPERATURE. AVERAGE TEMPERATURE SOLVED BY ITERATION.
C *** MAXIMUM NUMBER OF NODES = 25 X 25
C
C *****
C
C PARAMETERS: 26
C * NOTE * PAR(4), PAR(22) & PAR(24) MUST BE CONSISTENT, AS FOLLOWS:
C      PAR(4)*PAR(24) = TRNSYS TIME STEP (IN SECONDS)
C      PAR(22) & PAR(24) MUST SATISFY STABILITY REQUIREMENT
C
C 1  IU = THERESIS UNIT INDEX (1-10)  CHANGED FROM 4 TO 10 ON 8-7-89 JSS
C
C PIPE CIRCUITING:
C 2  EQTPL = PIPE LENGTH FOR EACH PARALLEL CIRCUIT (FT)
C 3  NPARLL = NUMBER OF CIRCUITS IN PARALLEL
C
C 4  NTERAT = NUMBER OF NUMERICAL TIME STEPS WITHIN EACH TRNSYS TIME
C      STEP ([DELTS] * [NTERAT] = [TRNSYS TIME STEP])
C
C FLUID TEMP CONVERGENCE:
C 5  MAXIT = MAXIMUM NUMBER OF ITERATION (FOR EACH CIRCUIT IN EACH
C      TRNSYS RUN TIME STEP) FOR CONVERGENCE OF AVG FLUID TEMP
C      BEFORE TERMINATION OF RUN
C 6  ERROR = TOLERANCE OF AVERAGED WATER TEMPERATURE IN EACH CIRCUIT
C      FOR INTERATION CONVEGENCE (F)
C
C 7  TINIT = INITIAL SLAB TEMPERATURE (F)
C
C SLAB:
C 8  COND = CONDUCTIVITY OF SLAB (BTU/HR-FT-F)
C 9  RHO = DENSITY OF SLAB (LB/CU FT)
C 10 CP = SPECIFIC HEAT OF SLAB (BTU/LB-F)
C 11 THICK = TOTAL THICKNESS OF SLAB (INCH)
C
C SURFACE INSULATION:
C 12 CONDT = THERMAL CONDUCTANCE OF INSULATION (CARPETING, ETC.)
C      ABOVE THE SLAB (BTU/HR-SQ FT-F)
C 13 CONDB = THERMAL CONDUCTANCE OF INSULATION UNDER THE SLAB
C      (BTU/HR-SQ FT-F)
C      * NOTE * CONDT & CONDB ARE FOR A GIVEN THICKNESS OF
C      INSULATION (LESS THAN OR EQUAL TO 0.00 IF NONE)
C
C PIPES:
C 14 CONDP = THERMAL CONDUCTITIVITY OF PIPE (BTU/HR-FT-F)
C 15 RHOP = DENSITY OF PIPE (LB/CU FT)
C 16 CPP = SPECIFIC HEAT OF PIPE (BTU/LB-F)
C 17 R = OUTER RADIUS OF PIPE (INCH)
C 18 RIP = INNER RADIUS OF PIPE (INCH)
C 19 S = PIPE SPACING, CENTER-TO-CENTER (INCH)
C
C FLUID FLOW HEAT TRANSFER:
C 20 CONV1 = CONVECTION HEAT TRANSFER COEFFICIENT OF WATER IN PIPE
C      -- W/FLOW (BTU/HR-SQ FT-F)
C 21 CONV2 = CONVECTION HEAT TRANSFER COEFFICIENT OF WATER IN PIPE
C      -- NO FLOW (BTU/HR-SQ FT-F)
C
C NUMERICAL SOLN SPEC.:

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C 22 DELX = LENGTH (WIDTH) OF EACH NODE (INCH)
C      * NOTE * DELX SHOULD BE LESS THAN R.
C              PARTIAL NODE IS NOT ALLOWED => [THICK/DELX]
C              SHOULD BE AN INTEGER, OTHERWISE THE TOTAL THICKNESS
C              OF THE NODES WILL BE SOMEWHAT GREATER THAN [THICK].
C              THE SAME APPLIES TO [S/2/DELX].
C 23 NCEN = NODE NUMBER FOR THE PIPE-FIELD PLANE. THE PIPES SHOULD ALW
CAYS
C              BE EMBEDDED IN THE SLAB. ENTER 0 IF CENTERED, BUT IF THE
C              NUMBER OF NODES ALONG THE THICKNESS OF SLAB IS EVEN, THEY
C              WILL BE SLIGHTLY OFF-CENTERED.
C              (NODE NUMBER BEGINS WITH 1 AT THE BOTTOM OF SLAB)
C
C 24 DELTS = NUMERICAL ITERATION TIME STEP (SEC)
C              MUST BE LESS THAN TRNSYS TIME STEP AND
C              [DELTS] * [NTERAT] = [TRNSYS TIME STEP]
C      * NOTE * DELX & DELTS MUST SATISFY STABILITY REQUIREMENT !
C
C 25 FRAC = AVERAGE CROSS-SECTIONAL AREA OF NODES AROUND PIPE WALL
C              (FRACTION OF AREA OF WHOLE INTERIOR NODES)
C 26 RADD = ADDITIONAL THERMAL RESISTANCE FOR NODES AROUND PIPE WALL
C              (HR-FT-F/BTU)
C
C INPUTS: 5
C
C 1  TTOP = TOP SURFACE TEMPERATURE (F)
C 2  TBOM = BOTTOM SURFACE TEMPERATURE (F)
C 3  TFINEQ = ENTERING FLUID TEMPERAUTRE (F)
C 4  FLOW = MASS FLOW RATE OF WATER (LB/HR)
C              (IF FLOW=0 => BYPASS)
C      * AS THE TEMP GRADIENT ALONG THE DIRECTION OF FLOW IS NOT DIRECTLY
C      SIMULATED, IF TFINEQ IS VERY LONG &/OR FLOW IS
C      VERY SMALL, THE FLUID TEMPERATURE WILL NOT CONVERGE *
C 5  ICONV = INDEX TO DETERMINE IF CONVECTION TO PIPE
C              IS TO BE INCLUDED
C
C OUTPUTS: 7
C
C      * ALL HEAT TRANSFER RATES ARE FOR TOTAL NUMBER OF CIRCUITS
C      AND TOTAL PIPE LENGTH *
C 1  QTZ = AVERAGED HEAT TRANSFER RATE FROM FLUID TO SLAB (BTUH)
C              (+ = HEAT GAIN TO WATER, - = HEAT LOSS FROM WATER)
C 2  QTOPTZ = AVERAGED HEAT TRANSFER RATE FROM TOP SURFACE TO
C              ROOM (BTUH)
C              (+ = HEAT GAIN TO ROOM, - = HEAT LOSS FROM ROOM)
C 3  QBOMTZ = AVERAGED HEAT TRANSFER RATE FROM BOTTOM SURFACE
C              TO ROOM (BTUH)
C              (+ = HEAT GAIN TO ROOM, - = HEAT LOSS FROM ROOM)
C 4  TFOUT = TEMPERATURE OF FLUID LEAVING SERIES-CIRCUITS AT THE END O
CF
C              TRNSYS TIME STEP (F)
C 5  TBULK = BULK/AVERAGED TEMPERATURE OF SLAB (F)
C              (IT IS ONLY AN APPROXIMATION ... ALL NODES ARE WEIGHTED
C              EQUALLY)
C 6  IRUN = NUMBER OF ITERATION BEFORE THE AVG FLUID TEMP CONVERGED
C 7  TFLUID = AVG FLUID TEMP USED FOR FINAL NUMERICAL SOLN (F)
C
C *****
C
C      subroutine type47(time, xin, out, t, dt, par, info)
C      implicit real*8(a-h,o-z)
C      dimension par(26), xin(5), out(7), info(9), tdump(10, 25, 25)
C      dimension ipipe(10, 25, 25), tt(25, 25), ttnew(10, 25, 25), const(
&10, 26)
C      common /flxdum/ jtdum, ctdum, jbdum, cbdum
C      common /constn/ aa, bb, cctop, ddtop, ccbom, ddbom, ctf, ct3, ctt

```

```

&, cwater
common /nodesp/ ipipe, m, n, msp, nbp, ntp, ncen, nipipe, icount
common /sim/ timeo, tfinal, delt
common /lunits/ lur, luw, iform
C
C *** FIRST CALL OF SIMULATION
data pi / 3.1416 /
# 135 "type47_89.for"
if (info(7) .eq. (-1)) then
temph = par(20)
ni = info(3)
np = info(4)
info(6) = 7
call typeck(1, info, ni, np, 0)
C
C *** INITIALIZATION
# 141 "type47_89.for"
end if
# 144 "type47_89.for"
iu = idint(par(1))
eqtpl = par(2)
nparll = idint(par(3))
nterat = idint(par(4))
maxit = idint(par(5))
error = par(6)
ttop = xin(1)
tbom = xin(2)
tfineq = xin(3)
tfin = tfineq
C
C IDEBUG WAS UTILIZED IN EARLY SIMULATIONS AS A DEBUGGING TOOL
C TRNSYS SIZE CONSTRAINTS DURING INITIAL DEVELOPMENT OF THE BSTS
C SYSTEM MODEL (9/88) REQUIRED THIS INPUT BE REMOVED. RATHER THAN
C REMOVE IT FROM CODE, IT HAS BEEN SET HERE = 0, OR off!
C T. J. MARSEILLE 10-6-88
C
# 154 "type47_89.for"
flow = xin(4)
C
C IF "ICONV" IS INPUT AS 1, THE CONVECTION COEFFICIENT BETWEEN
C THE PIPE AND SLAB IS SET TO (ESSENTIALLY) ZERO. THIS FEATURE
C WAS ADDED DURING THE EARLY BSTS MODEL DEVELOPMENT (9/88) AS A WAY
C TO TURN on AND off
C WATER FLOW THROUGH THE SLAB. T.J. MARSEILLE 10-6-88.
C
# 162 "type47_89.for"
idebug = 0
# 170 "type47_89.for"
iconv = idint(xin(5))
if (iconv .ne. 1) then
par(20) = .000001
else
par(20) = temph
C
C *** CALCULATE & STORE CONSTANTS
# 175 "type47_89.for"
end if
C
C *** RETRIEVE STORED CONSTANTS
# 178 "type47_89.for"
call inital(info, par, const, iu, idebug)
# 181 "type47_89.for"
aa = const(iu,1)
bb = const(iu,2)
cctop = const(iu,3)
ddtop = const(iu,4)

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```

      ccbom = const(iu,5)
      ddbom = const(iu,6)
      if (flow .gt. 0.) then
        resisp = const(iu,7)
        ctf = const(iu,9)
        ctt = const(iu,12)
      else
        resisp = const(iu,8)
        ctf = const(iu,10)
        ctt = const(iu,13)
      end if
      ct3 = const(iu,11)
      jtdum = idint(const(iu,14))
      ctdum = const(iu,15)
      jbdum = idint(const(iu,16))
      cbdum = const(iu,17)
      m = idint(const(iu,18))
      n = idint(const(iu,19))
      msp = idint(const(iu,20))
      ntp = idint(const(iu,21))
      nbp = idint(const(iu,22))
      ncen = idint(const(iu,23))
      icount = idint(const(iu,24))
c
c *** RESET NODAL TEMPERATURE AT FIRST CALL OF THE TIME STEP
# 208 "type47_89.for"
      cwater = const(iu,25)
# 211 "type47_89.for"
      if ((time .eq. 0.) .or. (info(7) .eq. (-1))) then
        tinit = par(7)
        do 5 i = 1, m
          do 5 j = 1, n
            if (ipipe(iu,i,j) .eq. 0) then
              tdump(iu,i,j) = tfin
              tt(i,j) = tfin
              ttnew(iu,i,j) = tfin
            else
              tdump(iu,i,j) = tinit
              tt(i,j) = tinit
              ttnew(iu,i,j) = tinit
            end if
          5 continue
        qt = 0.
        qtopt = 0.
        qbomt = 0.
        tfout = tfin
        tbulk = tinit
        irun = 0
        tfluid = tfin
        goto 999
      else
        if (info(7) .eq. 0) then
          do 10 i = 1, m
            do 10 j = 1, n
              10 tdump(iu,i,j) = ttnew(iu,i,j)
              const(iu,26) = out(7)
            end if
          end if
c
# 240 "type47_89.for"
      end if
# 242 "type47_89.for"
      20 irun = 0
c
c *** START ITERATION FOR EACH CIRCUIT
# 243 "type47_89.for"
      tfluid = tfin

```

```

# 246 "type47_89.for"
25 continue
c
c *** INITIALIZE TEMPERATURE
# 247 "type47_89.for"
irun = irun + 1
# 250 "type47_89.for"
if (flow .le. 0.) tfluid = const(iu,26)
do 50 i = 1, m
do 50 j = 1, n
if (ipipe(iu,i,j) .eq. 0) then
tt(i,j) = tfluid
else
tt(i,j) = tdump(iu,i,j)
end if
c * AT TOP & BOTTOM SURFACE
# 258 "type47_89.for"
50 continue
# 260 "type47_89.for"
do 100 i = 1, m
if (ccbom .eq. 0.) tt(i,1) = tbom
if (cctop .eq. 0.) tt(i,n) = ttop
c
# 263 "type47_89.for"
100 continue
# 265 "type47_89.for"
if (idebug .eq. 1) then
write(unit=luw, fmt=*)
&'*****'
# 267 "type47_89.for"
write(unit=luw, fmt=1000) time, info(7), iu
write(unit=luw, fmt=*) ' INITIAL CONDITIONS'
write(unit=luw, fmt=2490) ttop
do 182 j = n, 1, -1
write(unit=luw, fmt=2500) (tt(i,j), i = 1, m)
182 continue
write(unit=luw, fmt=2490) tbom
c
# 274 "type47_89.for"
end if
c
c *** CALCULATE EXIT FLUID TEMPERATURE
# 276 "type47_89.for"
call numerc(tt, ttnew, ttop, tbom, flow, tfnew, nterat, tfluid,
&tbulk, qt, qtopt, qbomt, idebug, iprtb, iprte, resisp, iu)
# 281 "type47_89.for"
if (flow .le. 0.) goto 887
tfout = (((2. * npar11) * qt) * eqtpl) / (flow * 1.) + tfin
tfavg = 0.5 * (tfin + tfout)
terror = dabs(tfavg - tfluid)
if (dabs(terror) .gt. error) then
if (irun .lt. maxit) then
tfnew = tfavg
if (idebug .eq. 1) write(unit=luw, fmt=2000) qt, qtopt, qbomt,
&tfluid, tfout, tfavg, tfin, tbulk, irun, terror
tfluid = tfnew
goto 25
else
write(unit=*, fmt=*)
&' EXECUTION TERMINATED-WATER TEMP DID NOT CONVERGE '
# 294 "type47_89.for"
write(unit=luw, fmt=*)
&' EXECUTION STOP-WATER TEMP DID NOT CONVERGE'
# 295 "type47_89.for"
stop
end if

```

```

        end if
C
# 298 "type47_89.for"
      goto 888
# 300 "type47_89.for"
      887 tfout = tfnew
      tfluid = tfnew
C
C *** CONVERT HEAT TRANSFER RATES TO PER TOTAL PIPE LENGTH & SPACING
# 302 "type47_89.for"
      888 continue
# 305 "type47_89.for"
      qt = ((qt * eqtpl) * nparll) * 2.
      qtopt = ((qtopt * eqtpl) * nparll) * 2.
C
C *** OUTPUTS
# 307 "type47_89.for"
      qbomt = ((qbomt * eqtpl) * nparll) * 2.
# 310 "type47_89.for"
      999 out(1) = qt
      out(2) = qtopt
      out(3) = qbomt
      out(4) = tfout
      out(5) = tbulk
      out(6) = dfloat(irun)
C
# 316 "type47_89.for"
      out(7) = tfluid
# 318 "type47_89.for"
      1000 format(21h TIME, INFO(7), IU ,f10.4,2i5)
      2000 format(7h *MAIN*/,
&48h      QT      QTOPT      QBOMT      TFLUID      TFOUT      TFAVG,
&28h      TFIN      TBULK RUN      TERROR/,8f8.2,i4,f8.4/)
      2010 format(i3)
      2490 format(f6.2)
      2500 format(13f6.2)
      return
C
C *****
C
# 326 "type47_89.for"
      end
C
C *** CALCULATES & STORES PARAMETERS
C
      subroutine inital(info, par, const, iu, idebug)
      implicit real*8(a-h,o-z)
      real*8 mcpn, mcpp
      dimension ipipe(10, 25, 25), info(9), par(26), const(10, 26)
      common /flxdum/ jtdum, ctdum, jbdum, cbdum
      common /constn/ aa, bb, cctop, ddtop, ccbom, ddbom, ctf, ct3, ctt
&, cwater
      common /nodesp/ ipipe, m, n, msp, nbp, ntp, ncen, nipipe, icount
      common /lunits/ lur, luw, iform
C
      data pi / 3.1416 /
      data rhow / 62.4 /
      data cpw / 1. /
# 342 "type47_89.for"
      cond = par(8)
      rho = par(9)
      cp = par(10)
      thick = par(11)
      condh = par(12)
      condb = par(13)
      condp = par(14)

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```

      rhop = par(15)
      cpp = par(16)
      r = par(17)
      rip = par(18)
      s = par(19)
      conv1 = par(20)
      conv2 = par(21)
      delx = par(22)
      ncen = idint(par(23))
      delts = par(24)
      frac = par(25)
C
C *** CALCULATE NODAL PARAMETERS
# 360 "type47_89.for"
      radd = par(26)
# 363 "type47_89.for"
      dely = delx
      m = idnint(((s / 2) / delx) + 0.4999) + 1
      n = idnint((thick / dely) + 0.4999) + 1
      if ((m .gt. 25) .or. (n .gt. 25)) then
        write(unit=*, fmt=*) ' EXECUTION TERMINATED - TOO MANY NODES!', m
        &, m
# 368 "type47_89.for"
        write(unit=luw, fmt=*) ' EXECUTION TERMINATED - TOO MANY NODES!',
        &m, n
# 369 "type47_89.for"
        stop
      end if
      if (ncen .le. 0) ncen = nint((n + 1) + 0.4999) / 2
      nbp = ncen - idnint((rip / dely) + 0.4999)
      ntp = ncen + idnint((rip / dely) + 0.4999)
      msp = idnint((rip / delx) + 0.4999) + 1
      if ((ntp .ge. n) .or. (nbp .le. 1)) .or. (msp .ge. m) then
        write(unit=*, fmt=*)
        &'EXECUTION TERMINATED - INVALID NODE ARRANGEMENT!'
# 377 "type47_89.for"
        write(unit=luw, fmt=*)
        &'EXECUTION TERMINATED - INVALID NODE ARRANGEMENT!'
# 378 "type47_89.for"
        stop
      end if
      thick = thick / 12.
      r = r / 12.
      rip = rip / 12.
      s = s / 12.
      delx = delx / 12.
      dely = delx
      delt = delts / 3600.
      alpha = cond / (rho * cp)
      aa = (alpha * delt) / (delx * delx)
C
C *** SET UP DUMMY PARAMETERS FOR CALCULATING HEAT FLUX
# 389 "type47_89.for"
      bb = 1. - (4. * aa)
# 392 "type47_89.for"
      if (condt .le. 0.) then
        jtdum = n - 1
        ctdum = cond
        cctop = 0.
        ddtop = 0.
      else
        jtdum = n
        ctdum = condt * delx
        cctop = ((condt * 2) * delt) / ((rho * cp) * delx)
        ddtop = (1 - (4. * aa)) - cctop
      end if

```

```

        if (condb .le. 0.) then
            jbdum = 2
            cbdum = cond
            ccbom = 0.
            ddbom = 0.
        else
            jbdum = 1
            cbdum = cond * delx
            ccbom = ((condb * 2) * delt) / ((rho * cp) * delx)
            ddbom = (1 - (4. * aa)) - ccbom
C
C *** PIPE WALL
# 413 "type47_89.for"
        end if
C
C *** THERMAL RESISTANCE OF PIPE PER ADJACENT NODE
# 416 "type47_89.for"
        call pipe(rip, delx, dely, idebug, iu)
# 419 "type47_89.for"
        rdum = ((2 * nipipe) * (dlog(r / rip) / ((2. * pi) * condp))) +
&radd
# 420 "type47_89.for"
        resisl = rdum + ((2 * nipipe) / (((2. * pi) * rip) * conv1))
C
C *** AVERAGED AREA OF NODES ADJACENT PIPE
# 421 "type47_89.for"
        resisl = rdum + ((2 * nipipe) / (((2. * pi) * rip) * conv2))
# 424 "type47_89.for"
        mcpn = (rho * ((frac * delx) * delx)) * cp
        mcpp = ((rho * (pi * ((r ** 2) - (rip ** 2)))) * cpp) / (2 *
&nipipe)
# 426 "type47_89.for"
        ctf1 = delt / ((mcpn + mcpp) * resisl)
        ctf2 = delt / ((mcpn + mcpp) * resisl)
        ct3 = (cond * delt) / (mcpn + mcpp)
        ctt1 = (1. - ctf1) - (3. * ct3)
C
C *** WATER
# 430 "type47_89.for"
        ctt2 = (1. - ctf2) - (3. * ct3)
C
        cwater = (2 * delt) / ((rho * pi) * (rip ** 2))
C
C *** CHECK FOR SYSTEM INSTABILITY
# 435 "type47_89.for"
        if (idebug .eq. 1) write(unit=luw, fmt=3000) aa, bb, cctop, ddtop
&, ccbom, ddbom, ctf1, ct3, ctt1, resisl, ctf2, ctt2, resisl, jtdum
&, ctsum, jbdum, cbdum, cwater
# 441 "type47_89.for"
        if (((bb .lt. 0.) .or. (ctt1 .lt. 0.)) .or. (ctt2 .lt. 0.))
& .or. (ddtop .lt. 0.)) .or. (ddbom .lt. 0.)) then
            write(unit=*, fmt=*) ' EXECUTION TERMINATED - SYSTEM UNSTABLE!'
            write(unit=luw, fmt=*) ' EXECUTION TERMINATED - SYSTEM UNSTABLE!'
            stop
C
C *** STORE CONSTANTS
# 446 "type47_89.for"
        end if
# 449 "type47_89.for"
        const(iu,1) = aa
        const(iu,2) = bb
        const(iu,3) = cctop
        const(iu,4) = ddtop
        const(iu,5) = ccbom
        const(iu,6) = ddbom
        const(iu,7) = resisl

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      const(iu,8) = resis2
      const(iu,9) = ctfl
      const(iu,10) = ctf2
      const(iu,11) = ct3
      const(iu,12) = ctt1
      const(iu,13) = ctt2
      const(iu,14) = dfloat(jtdum)
      const(iu,15) = ctdum
      const(iu,16) = dfloat(jbdum)
      const(iu,17) = cbdum
      const(iu,18) = dfloat(m)
      const(iu,19) = dfloat(n)
      const(iu,20) = dfloat(msp)
      const(iu,21) = dfloat(ntp)
      const(iu,22) = dfloat(nbp)
      const(iu,23) = dfloat(ncen)
      const(iu,24) = dfloat(icount)
C
# 473 "type47_89.for"
      const(iu,25) = cwater
# 475 "type47_89.for"
3000 format(9h *INITAL*/,42h      AA      BB CCTOP DDTOP CCBOM DDBOM,
&28h      CTF1      CT3      CTT1 RESIS1/,10f7.4/,
&49h      CTF2      CTT2 RESIS2 JTDUM CTDUM JBDUM CBDUM,8h CWATER
&/,3f7.4,2(i7,f7.4),f7.4)
# 480 "type47_89.for"
      return
C
C *****
C
# 481 "type47_89.for"
      end
C
C *** CALCULATES PIPE NODE CONFIGURATIONS
C
      subroutine pipe(r, delx, dely, idebug, iu)
      implicit real*8(a-h,o-z)
      dimension ipipe(10, 25, 25), ipipez(25, 25)
      common /nodesp/ ipipe, m, n, msp, nbp, ntp, ncen, nipipe, icount
C
      common /lunits/ lur, luw, iform
# 493 "type47_89.for"
      if (idebug .eq. 1) write(unit=luw, fmt=1200) m, n, msp, nbp, ntp,
&ncen, iu
# 494 "type47_89.for"
      do 150 i = 1, m
      do 150 j = 1, n
      ipipez(i,j) = 2
C
# 497 "type47_89.for"
      150 ipipe(iu,i,j) = 2
# 499 "type47_89.for"
      do 250 i = 1, msp
      ydum = (r * r) - ((i - 1) * delx) ** 2)
      if (ydum .le. 0.) then
      yr = 0.
      else
      yr = dsqrt(ydum)
      end if
      do 250 j = 1, msp
C
C * IPIPE=0=FLUID,=1=PIPE WALL,2=SLAB
# 507 "type47_89.for"
      250 if (yr .gt. ((j - 1) * dely)) ipipez(i,j) = 0
# 510 "type47_89.for"
      do 300 i = 1, msp

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```

        idum = 0
        do 300 j = 1, msp
            if ((ipipez(i,j) .eq. 2) .or. (ipipez(i,j) .eq. 1)) then
                idum = idum + 1
                if (idum .eq. 1) ipipez(i,j) = 1
            end if
300    continue
        do 350 j = 1, msp
            idum = 0
            do 350 i = 1, msp
                if ((ipipez(i,j) .eq. 2) .or. (ipipez(i,j) .eq. 1)) then
                    idum = idum + 1
                    if (idum .eq. 1) ipipez(i,j) = 1
                end if
            end do
        end do

C
C *** TRANSFER CENTER OF ORIGIN BACK TO LOWER LEFT CORNER
# 525 "type47_89.for"
350 continue
# 528 "type47_89.for"
        do 800 i = 1, msp
            do 800 j = 1, msp
                ipipe(iu,i,(j + ncen) - 1) = ipipez(i,j)
            end do
        end do

C
C *** SUM PIPE NODES
# 531 "type47_89.for"
        do 800 i = 1, msp
            do 800 j = 1, msp
                ipipe(iu,i,((-j) + ncen) + 1) = ipipez(i,j)
            end do
        end do
# 534 "type47_89.for"
        nipipe = -1
        do 810 i = 1, msp
            do 810 j = nbp, ntp
                nipipe = nipipe + 1
            end do
        end do

C
C *** SUM ALL NODES
# 537 "type47_89.for"
        do 810 i = 1, msp
            do 810 j = nbp, ntp
                nipipe = nipipe + 1
            end do
        end do
# 540 "type47_89.for"
        icount = 0
        do 814 i = 1, m
            do 814 j = 1, n
                if (ipipe(iu,i,j) .ne. 0) icount = icount + 1
            end do
        end do
# 543 "type47_89.for"
        do 904 j = n, 1, -1
            if (idebug .eq. 1) then
                write(unit=luw, fmt=1400) nipipe, icount
            end if
            do 904 i = 1, m
                write(unit=luw, fmt=3000) j, (ipipe(iu,i,j), i = 1, m)
            end do
        end do
# 549 "type47_89.for"
        end if
# 551 "type47_89.for"
        1200 format(32h *PIPE*      M      N MSP NBP NTP NCEN/,7x,6i4,8h IIIUUU ,i4)
        1400 format(23h *PIPE*  NIPIPE  ICOUNT/,7x,2i8)
        3000 format(1x,3hIP ,22i3)
        return

C
C *****
C
# 555 "type47_89.for"
        end

C
C *** NUMERICAL SOLUTION
C
        subroutine numerc(tt, ttnew, ttop, tbom, flow, tfnew, nterat,
            stfluid, tbulk, qt, qtopt, qbomt, idebug, iprtb, iprte, resisp, iu)
            implicit real*8(a-h,o-z)
            dimension ipipe(10, 25, 25), ttnew(10, 25, 25), tt(25, 25)

```

```

        common /flxdum/ jtdum, ctdum, jbdum, cbdum
        common /constn/ aa, bb, cctop, ddtop, ccbom, ddbom, ctf, ct3, ctt
        &, cwater
        common /nodesp/ ipipe, m, n, msp, nbp, ntp, ncen, nipipe, icount
C
C *** ITERATION
common /lunits/ lur, luw, iform
# 572 "type47_89.for"
        iterat = 0
        qt = 0.
        qtopt = 0.
        qbomt = 0.
        do 180 i = 1, m
        do 180 j = 1, n
C
# 578 "type47_89.for"
        180 ttnew(iu,i,j) = tt(i,j)
C
C * ALONG PIPE WALL
# 580 "type47_89.for"
        50 iterat = iterat + 1
# 583 "type47_89.for"
        do 200 i = 2, msp
        do 200 j = nbp, ntp
        if (ipipe(iu,i,j) .eq. 1) then
            dum = 0.
            do 190 idum = i - 1, i + 1, 2
            190 if (ipipe(iu,idum,j) .ne. 0) dum = dum + tt(idum,j)
            do 192 jdum = j - 1, j + 1, 2
            192 if (ipipe(iu,i,jdum) .ne. 0) dum = dum + tt(i,jdum)
            ttnew(iu,i,j) = ((ctf * tfluid) + (ct3 * dum)) + (ctt * tt(i,j))
            end if
        200 continue
        do 240 j = nbp, ntp
        if (ipipe(iu,1,j) .eq. 1) then
            dum = 0.
            do 230 jdum = j - 1, j + 1, 2
            230 if (ipipe(iu,1,jdum) .ne. 0) dum = dum + tt(1,jdum)
            ttnew(iu,1,j) = ((ctf * tfluid) + (ct3 * ((2 * tt(2,j)) + dum)))
            & + (ctt * tt(1,j))
# 600 "type47_89.for"
            end if
C
C * LEFT & RIGHT BOUNDRY
# 601 "type47_89.for"
        240 continue
# 604 "type47_89.for"
        do 310 j = 2, nbp - 1
            ttnew(iu,1,j) = (aa * (((2 * tt(2,j)) + tt(1,j + 1)) + tt(1,j - 1)
            &)) + (bb * tt(1,j))
        310 continue
        do 315 j = ntp + 1, n - 1
            ttnew(iu,1,j) = (aa * (((2 * tt(2,j)) + tt(1,j + 1)) + tt(1,j - 1)
            &)) + (bb * tt(1,j))
        315 continue
        do 320 j = 2, n - 1
            ttnew(iu,m,j) = (aa * (((2 * tt(m - 1,j)) + tt(m,j + 1)) + tt(m,j
            & - 1))) + (bb * tt(m,j))
C
C *** INTERIOR NODES
# 615 "type47_89.for"
        320 continue
# 618 "type47_89.for"
        do 400 i = 2, msp
        do 400 j = 2, n - 1
        if (ipipe(iu,i,j) .ne. 2) goto 400

```

```

      ttnew(iu,i,j) = (aa * (((tt(i - 1,j) + tt(i + 1,j)) + tt(i,j - 1))
& + tt(i,j + 1))) + (bb * tt(i,j))
400 continue
      do 500 i = msp + 1, m - 1
      do 500 j = 2, n - 1
      ttnew(iu,i,j) = (aa * (((tt(i - 1,j) + tt(i + 1,j)) + tt(i,j - 1))
& + tt(i,j + 1))) + (bb * tt(i,j))
C
C *** TOP & BOTTOM OF SLAB
# 628 "type47_89.for"
500 continue
# 631 "type47_89.for"
      if (cctop .eq. 0.) then
      do 510 i = 1, m
510 ttnew(iu,i,n) = ttop
      else
      do 515 i = 2, m - 1
515 ttnew(iu,i,n) = (((2 * aa) * tt(i,n - 1)) + (aa * (tt(i - 1,n) +
& tt(i + 1,n)))) + (cctop * ttop) + (ddtop * tt(i,n))
      ttnew(iu,1,n) = (((2 * aa) * tt(1,n - 1)) + ((2 * aa) * tt(2,n)))
& + (cctop * ttop) + (ddtop * tt(1,n))
      ttnew(iu,m,n) = (((2 * aa) * tt(m,n - 1)) + ((2 * aa) * tt(m - 1,
& n))) + (cctop * ttop) + (ddtop * tt(m,n))
C
# 642 "type47_89.for"
      end if
# 644 "type47_89.for"
      if (ccbom .eq. 0.) then
      do 520 i = 1, m
520 ttnew(iu,i,1) = tbom
      else
      do 525 i = 2, m - 1
525 ttnew(iu,i,1) = (((2 * aa) * tt(i,2)) + (aa * (tt(i - 1,1) + tt(i
& + 1,1)))) + (ccbom * tbom) + (ddbom * tt(i,1))
      ttnew(iu,1,1) = (((2 * aa) * tt(1,2)) + ((2 * aa) * tt(2,1))) + (
& ccbom * tbom) + (ddbom * tt(1,1))
      ttnew(iu,m,1) = (((2 * aa) * tt(m,2)) + ((2 * aa) * tt(m - 1,1)))
& + (ccbom * tbom) + (ddbom * tt(m,1))
C
C *** TOTAL HEAT TRANSFER FROM FLUID
# 655 "type47_89.for"
      end if
# 658 "type47_89.for"
      qit = 0.
      do 540 i = 2, msp
      do 540 j = nbp, ntp
540 if (ipipe(iu,i,j) .eq. 1) qit = qit + ((tt(i,j) - tfluid) / resisp
&)
# 662 "type47_89.for"
      do 560 j = nbp, ntp
560 if (ipipe(iu,1,j) .eq. 1) qit = qit + (((tt(1,j) - tfluid) /
& resisp) / 2.)
C
C *** NO FLOW CONDITIONS
# 664 "type47_89.for"
      qt = qt + qit
C
C *** CALCULATE HEAT TRANSFER RATE AT TOP AND BOTTOM SURFACE
# 667 "type47_89.for"
      if (flow .le. 0.) tfnew = tfluid + (qit * cwater)
# 670 "type47_89.for"
      bflux = 0.
      do 850 i = 2, m - 1
850 bflux = bflux + (tt(i,jbdum) - tbom)
      bflux = bflux + (((tt(1,jbdum) + tt(m,jbdum)) - (2 * tbom)) / 2.)
      qbom = bflux * cbdum

```

```

C
# 675 "type47_89.for"
qbomt = qbomt + qbom
# 677 "type47_89.for"
tflux = 0.
do 900 i = 2, m - 1
900 tflux = tflux + (tt(i,jtdum) - ttop)
tflux = tflux + (((tt(1,jtdum) + tt(m,jtdum)) - (2 * ttop)) / 2.)
qtop = tflux * ct dum
C
C *** UPDATE TEMPERATURE
# 682 "type47_89.for"
qtopt = qtopt + qtop
# 685 "type47_89.for"
do 600 i = 1, m
do 600 j = 1, n
if ((flow .le. 0.) .and. (ipipe(iu,i,j) .eq. 0)) then
tt(i,j) = tfnew
else
tt(i,j) = ttnew(iu,i,j)
end if
600 continue
C
C *** CALCULATE BULK SLAB TEMPERATURE
# 693 "type47_89.for"
if (flow .le. 0.) tfluid = tfnew
# 696 "type47_89.for"
tbulk = 0.
do 940 i = 1, m
do 940 j = 1, n
940 if (ipipe(iu,i,j) .ne. 0) tbulk = tbulk + tt(i,j)
C
# 700 "type47_89.for"
tbulk = tbulk / icount
# 702 "type47_89.for"
if (idebug .eq. 1) then
write(unit=luw, fmt=2000) iterat, qit, qtop, qbom, qt, qtopt,
&qbomt, tbulk, tfluid
# 704 "type47_89.for"
write(unit=luw, fmt=2490) ttop
do 602 j = n, 1, -1
602 write(unit=luw, fmt=2500) (tt(i,j), i = 1, m)
write(unit=luw, fmt=2490) tbom
C
C *** END OF ITERATION
# 708 "type47_89.for"
end if
# 711 "type47_89.for"
if (iterat .lt. nterat) goto 50
qt = qt / nterat
qtopt = qtopt / nterat
C
# 714 "type47_89.for"
qbomt = qbomt / nterat
# 716 "type47_89.for"
800 continue
2000 format(/,32h ITERAT QIT QTOP QBOM,
&40h QT QTOPT QBOMT TBULK TFLUID/,i8,8f8.2)
2010 format(i3)
2490 format(f6.2)
2500 format(13f6.2)
return
end

```

```

C
C *** BOILER MODEL MODIFIED FOR PROPORTIONAL CONTROL (11/23/87)
C   BY AMANDA CHIU
C
C THIS SUBROUTINE MODLES AN AUXILIARY HEATER.  WHENEVER THE INPUT
C CONTROL FUNCTION IGAM NOT = 0.0, HEAT IS ADDED AT A RATE BETWEEN
C 0.0 TO QMAX DETERMINED BY IGAM
C IN ORDER TO BRING THE FLOWSTREAM TEMPERATURE UP TO TSET.
C
      subroutine type40(time, xin, out, t, dt, par, info)
      implicit real*8(a-h,o-z)
      dimension xin(10), out(20), par(10), info(10)
C
      real*8 igam
C FIRST CALL OF SIMULATION
# 14 "type40_pnl.for"
      if (info(7) .ge. 0) goto 100
C   INFO(9)=0
# 16 "type40_pnl.for"
      info(6) = 3
C SET PARAMETER AND INPUT VARIABLES
# 18 "type40_pnl.for"
      call typeck(1, info, 3, 3, 0)
# 20 "type40_pnl.for"
      100 qmax = par(1)
          tset = par(2)
          cp = par(3)
          tin = xin(1)
C *** IGAM = 0 TO 1
# 24 "type40_pnl.for"
          flow = xin(2)
# 26 "type40_pnl.for"
          igam = xin(3)
C NO FLOW
# 27 "type40_pnl.for"
          if (flow .gt. 0.) goto 10
# 29 "type40_pnl.for"
          out(2) = 0.
          out(3) = 0.
C CHECK INLET TEMPERATURE AND CONTROL FUNCTION
# 31 "type40_pnl.for"
          return
C HEATER OFF
# 33 "type40_pnl.for"
          10 if ((tin .lt. tset) .and. (igam .gt. 0)) goto 20
# 35 "type40_pnl.for"
          tout = tin
          qaux = 0.
C HEATER ON
C *** QMAX PROPORTIONAL - 0 TO QMAX
# 37 "type40_pnl.for"
          goto 50
# 40 "type40_pnl.for"
          20 tout = tin + ((qmax * igam) / (flow * cp))
          tout = min(tset,tout)
C SET OUTPUTS
# 42 "type40_pnl.for"
          qaux = (flow * cp) * (tout - tin)
# 44 "type40_pnl.for"
          50 out(1) = tout
          out(2) = flow
          out(3) = qaux
          return
      end

```

```

C
C *****
C * COOLING TOWER *
C *****
C
C MODEL MODIFIED FOR PROPORTIONAL CONTROL (11/23/87)
C BY AMANDA CHIU
C
C *** WHILLIER MODEL TO CALCULATE PERFORMANCE OF COOLING TOWER
C 4 CELLS MAXIMUM
C
C *****
C *** ONE COOLING TOWER PER SIMULATION ***
C
C PARAMETERS: 6
C
C 1 FANMAX = MAXIMUM FAN CAPACITY, FOR ONE CELL (CFM)
C 2 POWMAX = MAXIMUM FAN POWER REQUIREMENT, FOR ONE CELL (KW)
C 3 NCELL = TOTAL NUMBER OF CELLS (DIMENSIONLESS)
C (MIN=1, MAX=4)
C 4 A = COEFFICIENT 'A' (DIMENSIONLESS)
C 5 B = COEFFICIENT 'B' (DIMENSIONLESS)
C FOR EQUATION  $E = A + B * R$ 
C WHERE E = COOLING TOWER EFFECTIVENESS (DIMENSIONLESS)
C R = COOLING TOWER CAPACITY FACTOR (DIMENSIONLESS)
C 6 TAU = COOLING TOWER CAPCITY TIME CONSTANT (MIN)
C (SET TO 0 FOR NEGLIGIBLE TRANSIENT DELAY)
C
C INPUTS: 5+NCELL
C
C *** FOR PROPORTIONAL CONTROL, INPUT(1) & INPUT(6) SHOULD BE
C CONNECTED TO THE SAME SAME PROPORTIONAL SIGNAL ***
C 1 ITOWER = COOLING TOWER ON-OFF INDEX (PROPORTIONAL - 0 TO 1)
C 2 TDB = AMBIENT DRY-BULB TEMPERATURE (F)
C 3 TWB = AMBIENT WET-BULB TEMPERATURE (F)
C 4 TCWR = INLET WATER TEMPERATURE (F)
C 5 FLCW = INLET WATER MASS FLOWRATE (LB/HR)
C 6 to 5+NCELL
C FRAC = FRACTIONAL FAN OPERATION FOR EACH CELL (DIMENSIONLESS)
C (FANS ARE ASSUMED TO OBEY THE FAN LAW)
C * IF ITOWER, FLCW, OR ALL FRAC = 0, TOWER IS OFF *
C
C OUTPUTS: 5
C
C 1 TCWS = OUTLET WATER TEMPERATURE (F)
C 2 DT = RANGE (F)
C 3 POWER = TOTAL FAN POWER (KW)
C 4 QAVG = TOWER CAPACITY AVERAGED OVER TIME STEP (KBTUH)
C 5 QNEW = TOWER CAPACITY AT THE END OF TIME STEP (KBTUH)
C
C *****
C
C subroutine type42(time, xin, out, t, dtdt, par, info)
C implicit real*8(a-h,o-z)
C dimension xin(9), out(5), par(6), info(9), frac(4)
C common /sim/ timeo, tfinal, delt
C
C *** FIRST CALL OF SIMULATION
C data cpw / 1. /
C # 60 "type42_pnl.for"
C if (info(7) .ne. (-1)) goto 100
C ni = info(3)
C np = info(4)
C info(6) = 5
C call typeck(1, info, ni, np, 0)

```

```

C
C *** SET PARAMETERS & INPUTS -- FANMAX IN CFM, FLCW IN GPM
# 65 "type42_pnl.for"
100 continue
# 68 "type42_pnl.for"
fanmax = par(1) * 60.
powmax = par(2)
ncell = par(3)
a = par(4)
b = par(5)

C
C * FOR PROPORTIONAL CONTROL
# 73 "type42_pnl.for"
tau = par(6) / 60.
# 76 "type42_pnl.for"
itower = 0
if (xin(1) .gt. 0) itower = 1
tdb = xin(2)
twb = xin(3)
tcwr = xin(4)
flcw = xin(5)
do 2 i = 1, ncell
2 frac(i) = xin(5 + i)

C
# 84 "type42_pnl.for"
if (info(7) .le. 0) qold = out(5)
# 86 "type42_pnl.for"
fdum = 0.
power = 0.
do 5 i = 1, ncell
5 fdum = fdum + frac(i)

C
# 90 "type42_pnl.for"
5 power = power + (powmax * (frac(i) ** 3))
# 92 "type42_pnl.for"
dt = 0.
if (((itower .eq. 0) .or. (fdum .le. 0)) .or. (flcw .le. 0))
&goto 800
# 94 "type42_pnl.for"
call psych1(tcwr, tcwr, h, scwr, vdum, wdum)
call psych1(tdb, twb, h, sa, va, wa)
do 20 i = 1, ncell
if (frac(i) .gt. 0.) then
fla = (fanmax * frac(i)) / va
qamax = max(fla * (scwr - sa), 1.d-5)
qwmax = max(((flcw * cpw) * (tcwr - twb)) / ncell, 1.d-5)
r = min(qamax, qwmax) / max(qamax, qwmax)
r = min(1.d0, max(0.d0, r))
eps = a + (b * r)
if (eps .gt. 1.) eps = 1.
dte = (((eps * ncell) * min(qamax, qwmax)) / flcw) / cpw
dt = dt + dte
end if
20 continue
dt = dt / ncell
qss = flcw * dt
if (tau .ne. 0.) then
qnew = qss - ((qss - qold) * dexp(- (delt / tau)))
qavg = qss - (((qss - qold) * (tau / delt)) * (1. - dexp(- (delt /
&tau))))
# 114 "type42_pnl.for"
dt = qavg / flcw
else
qnew = qss
qavg = qss
end if

```



```

        tcws = tcwr - dt
C
# 120 "type42_pnl.for"
      goto 900
# 122 "type42_pnl.for"
      800 tcws = tcwr
         power = 0.
         qavg = 0.
         qnew = 0.
C
C      WRITE(6,1100) TCWS,DT,POWER,QAVG,QNEW,QSS
c1100 FORMAT(2F6.2,4E8.2/)
C      WRITE(6,1200)
c1200 FORMAT('*****'/)
C
C *** SET OUTPUTS
# 126 "type42_pnl.for"
      900 continue
# 134 "type42_pnl.for"
      out(1) = tcws
      out(2) = dt
      out(3) = power
      out(4) = qavg
C
# 138 "type42_pnl.for"
      out(5) = qnew
# 140 "type42_pnl.for"
      return
C
C *****
C *   PSYCHROMETRIC FUNCTION   *
C *****
C
# 141 "type42_pnl.for"
      end
      subroutine psych1(tdb, twb, h, s, v, w)
      implicit real*8(a-h,o-z)
C
      data p / 14.7 /
# 150 "type42_pnl.for"
      t = ((twb - 32.) / 1.8) + 273.15
      psat = dexp(((((-5800.2206/ t)) + 1.3914993) - (0.04860239 * t)
&) + (0.41764768e-4 * (t ** 2)) - (0.14452093e-7 * (t ** 3))) + (
&6.545967*dlog(t)))
# 153 "type42_pnl.for"
      psat = psat * 0.000145
      wsat = (0.62198 * psat) / (p - psat)
      if (dabs(tdb - twb) .gt. .01) then
      w = (((1093. - (0.556 * twb)) * wsat) - (0.24 * (tdb - twb))) / ((
&1093. + (.444 * tdb)) - twb)
# 157 "type42_pnl.for"
      else
      w = wsat
      end if
      v = ((.3705 * (tdb + 460.)) * (1. + (1.6078 * w))) / p
      h = (.24 * tdb) + (w * (1061. + (.444 * tdb)))
C
# 162 "type42_pnl.for"
      s = h - ((w * 1.003) * twb)
# 164 "type42_pnl.for"
      return
      end

```

```

C
C *****
C * PROPOTIONAL CONTROL THERMOSTAT *
C *****
C
C *** MAXIMUM 5 THERMOSTATS
C
C
C MODIFIED 10/12/89 BY T.J. MARSEILLE
C CONTROL NO LONGER USES NIGHT PURGE TO
C TURN ON COOLING TOWER. LOGIC TURNS ON COOLING
C TOWER DURING SUMMER MONTHS WHENEVER THE OUTSIDE
C WETBULB DROPS BELOW "WET" F, ASSUMING WATER TEMPERATURES
C DON'T DROP BELOW "SETLOW" F (I.E., TO ENSURE BOILER IS NOT
C NEEDED). IF TEMPERATURES RISE ABOVE "WET" F WETBULB OUTSIDE
C THE COOLING TOWER WORKS TO MAINTAIN THE SETPOINT TCW.
C
C MODIFIED 8/16/88 BY T. J. MARSEILLE
C COOL ON/OFF INDEX HARDWIRED TO BE 1.0
C DURING 4-6 A.M. PERIOD OF THE SUMMER, UNLESS
C THE LOOP WATER TEMPERATURE DROPS BELOW 65 F,
C OR ON SATURDAY OR SUNDAY.
C THIS MODULE SHOULD ONLY BE USED WITH THESE MODIFICATIONS
C WHEN THE HVAC/MASS INTERACTION MODEL IS BEING USED TO
C PULLDOWN THE BUILDING IN THE MORNING.
C
C MODIFIED 7/5/89 BY T.J. MARSEILLE
C READS MONTH OF YEAR FROM WEATHER CARD (INPUT 4),
C AND DECIDES WHETHER TO USE 4-6 A.M. PURGE BASED
C ON THIS RATHER THAN HOUR OF YEAR. ALSO, REQUIRES
C PARAMETER INPUT SAYING WHICH MONTHS WILL USE PURGE
C (PARAMETERS 7 & 8)
C
C *****
C
C PARAMETERS: 9
C
C 1 TH = HEATING SETPOINT (F) - TEMP WHEN HEATING FIRST COMES ON
C 2 TC = COOLING SETPOINT (F) - TEMP WHEN COOLING FIRST COMES ON
C 3 TDELH = HEATING THROTTLING RANGE (F)
C      *** HEATING=MAX @ TEMP=TH-TDELH ***
C 4 TDELC = COOLING THROTTLING RANGE (F)
C      *** COOLING=MAX @ TEMP=TC+TDELC ***
C 5 TSETBK = HEATING SETBACK TEMPERATURE DIFFERENCE (F)
C      (* ENTER 0 FOR NO SETBACK*)
C 6 SETLOW = BSTS COOLING TOWER SETPOINT (F)
C 7 IMTH1 = MONTH BEFORE WHICH NO SUMMER PURGE IS USED
C 8 IMTH2 = MONTH AFTER WHICH NO SUMMER PURGE IS USED
C 9 WET = SETPOINT WET BULB TEMP. FOR FREE COOLING
C
C INPUTS: 6
C 1 TEMP = TEMPERATURE WHICH COMMANDS THE T' STAT (F)
C 2 ISETBK = HEATING SETBACK SCHEDULE (0=NO SETBACK, 1=SETBACK)
C 3 IWEEK = WEEKEND OVERRIDE OF HVAC /MASS C OOLDOWN LOGIC
C 4 IMONTH = MONTH OF YEAR
C 5 WAMB = OUTSIDE AIR AMBIENT HUMIDITY RATIO
C 6 TDB = OUTSIDE AIR TEMPERATURE
C
C OUTPUTS: 3
C 1 HEAT = HEATING ON/OFF (0.0-1.0)
C 2 COOL = COOLING ON/OFF (0.0-1.0)
C 3 SLAB = RESET SLAB COOLING CONTROL (0 OR 1)
C
C *****
C
C subroutine type43(time, xin, out, t, dt, par, info)

```

```

        implicit real*8(a-h,o-z)
        dimension par(15), xin(10), out(20), info(10)
        common /sim/ time0, timef, delt
C
C *** FIRST CALL OF SIMULATION
        common /lunits/ lur, luw, iform
# 72 "type43_cool.for"
        if (info(7) .ge. 0) goto 100
        ni = info(3)
        np = info(4)
        info(6) = 2
        call typeck(1, info, ni, np, 0)
C
C *** SET PARAMETERS
# 77 "type43_cool.for"
        100 continue
# 80 "type43_cool.for"
        slab = 0.
        tseth = par(1)
        tsetc = par(2)
        tdelh = par(3)
        tdelc = par(4)
        tback = par(5)
        setlow = par(6)
        imth1 = par(7)
        imth2 = par(8)
C
C *** SET INPUTS
# 89 "type43_cool.for"
        wet = par(9)
# 92 "type43_cool.for"
        temp = xin(1)
        iback = idint(xin(2))
        iweek = idint(xin(3))
        imonth = idint(xin(4))
        wamb = xin(5)
C
C
C *** ADJUST HEATING SETBACK TO ACCOUNT FOR SETBACK, IF ANY
# 97 "type43_cool.for"
        tdb = xin(6)
C
C *** TEMP FOR MAX(1) AND MIN(0) SIGNALS
        tseth = tseth - (tback * iback)
# 104 "type43_cool.for"
        tqmaxh = tseth - tdelh
        tqmaxc = tsetc + tdelc
        tqminh = tseth
C
C *** ERRORS
# 107 "type43_cool.for"
        tqminc = tsetc
# 110 "type43_cool.for"
        if (((tqmaxh .gt. tqminh) .or. (tqminh .gt. tqminc)) .or. (tqminc
& .gt. tqmaxc)) then
            write(unit=*, fmt=1000)
            write(unit=luw, fmt=1000)
1000 format(50h** TYPE43 ERROR -- INCORRECT THERMOSTAT SETTINGS **)
            call typeck(4, info, 0, 0, 0)
C
C *** ON/OFF CONTROL SIGNAL
# 116 "type43_cool.for"
        end if
# 119 "type43_cool.for"
        iheat = 0
        icool = 0

```

```

        if (temp .lt. tqminh) then
            iheat = 1
        else
            if (temp .gt. tqminc) icool = 1
C
C *** PROPORTIONAL CONTROL SIGNAL
# 125 "type43_cool.for"
        end if
# 128 "type43_cool.for"
        heat = 0.0
        cool = 0.0
        if (iheat .ne. 0) then
            if (tqmaxh .eq. tqminh) then
                heat = 1
            else
                heat = (temp - tqminh) / (tqmaxh - tqminh)
                if (heat .gt. 1.) heat = 1.
            end if
        else
            if (icool .ne. 0) then
                if (tqmaxc .eq. tqminc) then
                    cool = 1
                else
                    cool = (temp - tqminc) / (tqmaxc - tqminc)
                    if (cool .gt. 1.) cool = 1.
                end if
            end if
C
C HVAC/MASS LOGIC TO USE COOLING TOWER TO COOL LOOP WATER
C IN SUMMER WHEN WETBULB TEMPERATURES ARE LOW ENOUGH
C
# 146 "type43_cool.for"
        end if
# 151 "type43_cool.for"
        if (iweek .eq. 0) goto 2000
C
C WET BULB TEMPERATURE CALCULATION
C
C *** CALCULATE WET-BULB TEMP
# 152 "type43_cool.for"
        if ((imonth - imth1) .lt. 0.) .or. ((imonth - imth2) .gt. 0.))
            &goto 2000
# 157 "type43_cool.for"
        h = ((0.5583 * tdb) + (wamb * ((1.0328 * tdb) + 2467.9511))) -
            &17.8667
# 158 "type43_cool.for"
        if ((h + 17.68) .le. 0.) then
            write(unit=*, fmt=*)
            &' PSYCHROMETRIC PROBLEM IN TYPE 43, CHECK TEMP & HUMIDITY'
            write(unit=luw, fmt=*)
            &' PSYCHROMETRIC PROBLEM IN TYPE 43, CHECK TEMP & HUMIDITY'
        end if
        y = dlog(h + 17.68)
        twb = 80.1415 + (y * ((-78.192) + (y * (25.0362 - (y * 1.7586)))))
        if (twb .gt. tdb) twb = tdb
        if ((twb .le. wet) .and. (temp .ge. setlow)) then
            cool = 1.0
            slab = 1.
        end if
C
C *** OUTPUTS
# 171 "type43_cool.for"
        2000 continue
# 174 "type43_cool.for"
        out(1) = heat
        out(2) = cool

```

```
C
# 176 "type43_cool.for"
      out(3) = slab
# 178 "type43_cool.for"
      return
    end
```

```

C
C      *****
C      *      WATER SOURCE HEAT PUMP      *
C      *****
C
C      BY AMANDA CHIU
C      *****
C
C      *** MAXIMUM 10 HEAT PUMPS PER SIMULATION ***
C
C      PARAMETERS:  9
C
C      1  CAPTN(1) = NOMINAL CAPACITY - HEATING (KBTUH)
C          (*** POSITIVE ***)
C      2  POWN(1) = NOMINAL POWER - HEATING (KW)
C      3  CAPTN(2) = NOMINAL TOTAL CAPACITY - COOLING (KBTUH)
C          (*** NEGATIVE ***)
C      4  CAPSN(2) = NOMINAL SENSIBLE CAPACITY - COOLING (KBTUH)
C          (*** NEGATIVE ***)
C      5  POWN(2) = NOMINAL POWER - COOLING (KW)
C      6  TAU = CAPACITY TIME CONSTANT (MIN)
C          SET TO 0 FOR NEGLIGIBLE TRANSIENT DELAY
C      7  POWFN = NOMINAL FAN POWER (KW)
C      8  CFMN = NOMINAL AIR FLOWRATE (CFM)
C      9  MGPHN = NOMINAL WATER FLOWRATE (LB/HR)
C
C      INPUTS: 11
C
C      1  IH = HEATING ON-OFF CONTROL INDEX
C      2  IC = COOLING ON-OFF CONTROL INDEX
C      3  IFAN = FAN ON-OFF CONTROL INDEX
C      4  MGPH = WATER MASS FLOWRATE (LB/HR)
C      5  TWI = ENTERING WATER TEMPERATURE (F)
C      6  TRDB = INDOOR/ROOM DRY-BULB TEMPERATURE (F)
C      7  WR = INDOOR/ROOM HUMIDITY RATIO (LB/LB)
C      8  TADB = OUTDOOR/AMBIENT DRY-BULB TEMPERATURE (F)
C      9  WA = OUTDOOR/AMBIENT HUMIDITY RATIO (LB/LB)
C     10  CFM = INDOOR FAN FLOWRATE (CFM)
C     11  FRACA = VOLUMETRIC FRACTIONAL OUTDOOR AIR ENTERING INDOOR COIL
C              (DIMENSIOLESS)
C
C      OUTPUTS: 12
C
C      1  CAPTF = HEATING/COOLING TOTAL CAPACITY AT THE END OF THE
C              TIME STEP (KBTUH)
C              (HEATING=POSITIVE; COOLING=NEGATIVE)
C      2  CAPTAV = HEATING/COOLING TOTAL CAPACITY, AVERAGED OVER THE
C              TIME STEP (KBTUH)
C              (HEATING=POSITIVE; COOLING=NEGATIVE)
C      3  CAPSF = HEATING/COOLING SENSIBLE CAPACITY AT THE END OF THE
C              TIME STEP (KBTUH)
C              (HEATING=POSITIVE; COOLING=NEGATIVE)
C      4  CAPSAV = HEATING/COOLING SENSIBLE CAPACITY, AVERAGED OVER THE
C              TIME STEP (KBTUH)
C              (HEATING=POSITIVE; COOLING=NEGATIVE)
C      5  TODB = AIR TEMPERATURE LEAVING INDOOR COIL (F)
C      6  WO = AIR HUMIDITY RATIO LEAVING INDOOR COIL (LB/LB)
C      7  MAIR = SUPPLY AIR MASS FLOW RATE (LB/HR)
C      8  POW = COMPRESSOR POWER (KW)
C      9  HRAV = HEAT TRANSFER TO THE WATER LOOP (KBTUH)
C              (HEAT REJECTION=POSITIVE; HEAT ABSORPTION=NEGATIVE)
C     10  TWO = LEAVING WATER TEMPERATURE (F)
C     11  POWF = FAN POWER (KW)
C     12  IHP = HEAT PUMP INDEX
C          (1 TO 10, MAXIMUM)
C

```

```

C  NOMENCLATURES
C
C  ??S=STEADY-STATE, UNADJUSTED VALUES
C  ??J=STEADY-STATE, ADJUSTED VALUES
C  ??N=NOMINAL VALUES @ ARI POINT
C  ??I=INITIAL VALUES
C  ??F=FINAL VALUES
C  ??AV=AVERAGED OVER TIME STEP
C  X(I,J,K)  I:1=M,2=A1,3=A2, FOR Y=M+A1*X1+A2*X2;
C             J:1=UNADJUSTED, 2=ADJ. FOR INDOOR TEMP, 3=ADJ. FOR CFM
C             K:IHC=0=NO HEATING/COOLING, 1=HEATING, 2=COOLING
C             X:CT=TOTAL CAP, CS=SENSIBLE CAP, PW=POWER
C
C  *****
C
C      subroutine type50(time, xin, out, t, dtdt, par, info)
C      implicit real*8(a-h,o-z)
C      real*8 mw, mair, mgph, mgphn, ma, mr
C      dimension xin(11), out(12), par(9), info(9), ct(3, 3, 2), cs(3, 3
C      &, 2), pw(3, 3, 2), captn(2), capsn(2), pown(2), capti(10), capsi(
C      &10)
C      common /sim/ timeo, tfinal, delt
C      common /lunits/ lur, luw, iform
C
C  *** FIRST CALL OF SIMULATION
C      data capti / 10*0. /
C      data capsi / 10*0. /
C      data ihp / 0 /
C      data nhpu / 0 /
C      data iwarn / 0 /
C      data ct / 0.4101, 0.0763, 0.0073, 1.3366, -0.0048, 0., 0.8545,
C      &0.1455, 0., 1.1274, 0.0355, -0.0019, -0.1294, 0., 0.0169, 0.8605,
C      &0.1395, 0. /
C      data cs / 9*0., 0.9659, -0.0003, 0.0004, 0.8834, 0.0429, -0.0495,
C      &0.6898, 0.3102, 0. /
C      data pw / 0.6885, 0.0410, 0.0039, 0.6438, 0.0051, 0., 1.14, -0.14
C      &, 0., 0.5904, -0.0960, 0.0059, 0.6446, 0., 0.0053, 0.8401, 0.1599
C      &, 0. /
C  # 103 "type50_pnl.for"
C      idebug = 0
C      if (info(7) .ne. (-1)) goto 100
C      ni = info(3)
C      np = info(4)
C      info(6) = 12
C      call typeck(1, info, ni, np, 0)
C      nhpu = nhpu + 1
C      out(12) = nhpu
C      if (nhpu .gt. 10) call typeck(6, info, 0, 0, 0)
C
C  *** SET PARAMETERS AND INPUTS
C  # 112 "type50_pnl.for"
C      100 continue
C  # 115 "type50_pnl.for"
C      ihp = idint(out(12) + 0.1)
C      captn(1) = par(1)
C      capsn(1) = captn(1)
C      pown(1) = par(2)
C      captn(2) = par(3)
C      capsn(2) = par(4)
C      pown(2) = par(5)
C      tau = par(6) / 60.
C      powfn = par(7)
C      cfmn = par(8)
C      mgphn = par(9)
C
C  # 126 "type50_pnl.for"

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      gpmn = mgphn / (60. * 8.333)
# 128 "type50_pnl.for"
      ih = idint(xin(1))
      ic = idint(xin(2))
      if ((ih .eq. 1) .and. (ic .eq. 0)) ihc = 1
      if ((ih .eq. 0) .and. (ic .eq. 1)) ihc = 2
      if ((ih .eq. 0) .and. (ic .eq. 0)) ihc = 0
      if ((ih .eq. 1) .and. (ic .eq. 1)) call typeck(3, info, 0, 0, 0)
      ifan = idint(xin(3))
      mgph = xin(4)
      gpm = (mgph / 8.333) / 60.
      twi = xin(5)
      trdb = xin(6)
      wr = xin(7)
      tadb = xin(8)
      wa = xin(9)
      cfm = xin(10)
      frac = xin(11)

C
C   REVISION 6/88 BY T. J. MARSEILLE
C
C   SET IFAN = 1 IF IHC = 1 (SO FAN ALWAYS OPERATES WHEN HP NEEDED)
C
# 144 "type50_pnl.for"
      powf = 0.
# 150 "type50_pnl.for"
      if (ihc .ne. 0) ifan = 1
C
# 151 "type50_pnl.for"
      if (ifan .ne. 0) powf = powfn * ((cfm / cfmn) ** 3)
C
C *** CALCULATE MIXED-AIR CONDITIONS ENTERING INDOOR COIL
# 153 "type50_pnl.for"
      if ((ihc + ifan) .eq. 0) goto 800
# 156 "type50_pnl.for"
      vr = (0.0252 * (trdb + 460)) * (1. + (1.6078 * wr))
      mr = ((cfm * (1. - frac)) / vr) * 60.
      va = (0.0252 * (tadb + 460)) * (1. + (1.6078 * wa))
      ma = ((cfm * frac) / va) * 60.
      mair = mr + ma
      tidb = ((ma * tadb) + (mr * trdb)) / mair
C
# 162 "type50_pnl.for"
      wi = ((ma * wa) + (mr * wr)) / mair
C
C *** SET INITIAL VALUES
# 164 "type50_pnl.for"
      if (ihc .eq. 0) goto 700
# 167 "type50_pnl.for"
      if (info(7) .le. 0) then
        capti(ihp) = out(1)
        capsi(ihp) = out(3)
C
C *** CALCULATE WET-BULB TEMP
C * (NOTE: EQUATIONS ASSUME AVG NORMAL ROOM CONDITION)
# 170 "type50_pnl.for"
      end if
# 174 "type50_pnl.for"
      h = ((0.5583 * tidb) + (wi * ((1.0328 * tidb) + 2467.9511))) -
        &17.8667
# 175 "type50_pnl.for"
      if ((h + 17.68) .le. 0.) then
        write(unit=*, fmt=*)
        &' PSYCHROMETRIC PROBLEM IN TYPE 50, CHECK TEMP & HUMIDITY'
        write(unit=*, fmt=1060) time, info(1), tadb, wa, trdb, wr, tidb,
        &wi, h

```



```

# 179 "type50_pnl.for"
      write(unit=luw, fmt=*)
      &' PSYCHROMETRIC PROBLEM IN TYPE 50, CHECK TEMP & HUMIDITY'
      write(unit=luw, fmt=1060) time, info(1), tadb, wa, trdb, wr, tidb
      &, wi, h
# 182 "type50_pnl.for"
      end if
      y = dlog(h + 17.68)
      tiwb = 80.1415 + (y * ((-78.192) + (y * (25.0362 - (y * 1.7586))))
      &)
C
C *** CALCULATE ADJUSTED, STEADY-STATE PERFORMANCE
C
# 185 "type50_pnl.for"
      if (tiwb .gt. tidb) tiwb = tidb
# 189 "type50_pnl.for"
      capt = (ct(1,1,ihc) + ((ct(2,1,ihc) * gpm) / gpmn)) + (ct(3,1,ihc)
      &) * twi)
C
# 190 "type50_pnl.for"
      captj = (capt * ((ct(1,2,ihc) + (ct(2,2,ihc) * tidb)) + (ct(3,2,
      &ihc) * tiwb))) * (ct(1,3,ihc) + ((ct(2,3,ihc) * cfm) / cfmn))
# 194 "type50_pnl.for"
      if (ihc .eq. 1) then
        capss = capt
        capsj = captj
      else
        capss = (cs(1,1,2) + ((cs(2,1,2) * gpm) / gpmn)) + (cs(3,1,2) *
        &twi)
# 199 "type50_pnl.for"
        capsj = (capss * ((cs(1,2,2) + (cs(2,2,2) * tidb)) + (cs(3,2,2) *
        &tiwb))) * (cs(1,3,2) + ((cs(2,3,2) * cfm) / cfmn))
C
# 202 "type50_pnl.for"
      end if
# 204 "type50_pnl.for"
      pows = (pw(1,1,ihc) + ((pw(2,1,ihc) * gpm) / gpmn)) + (pw(3,1,ihc)
      & * twi)
C
# 205 "type50_pnl.for"
      powj = (pows * ((pw(1,2,ihc) + (pw(2,2,ihc) * tidb)) + (pw(3,2,ihc)
      &) * tiwb))) * (pw(1,3,ihc) + ((pw(2,3,ihc) * cfm) / cfmn))
# 209 "type50_pnl.for"
      if (((powj .le. 0.) .or. (powj .ge. 3.0)) .or. ((captj .le. 0.)
      &.or. (captj .ge. 3.0))) .or. ((capsj .le. 0.) .or. (capsj .ge.
      &3.0))) then
        iwarn = iwarn + 1
        write(unit=*, fmt=*) ' ERROR IN HEAT PUMP COEFFICIENT IN TYPE 50'
        write(unit=*, fmt=1070) time, info(1), tidb, tiwb, twi, mgph, cfm
        &, powj, captj, capsj
        write(unit=luw, fmt=*)
        &' ERROR IN HEAT PUMP COEFFICIENT IN TYPE 50'
# 217 "type50_pnl.for"
        write(unit=luw, fmt=1070) time, info(1), tidb, tiwb, twi, mgph,
        &cfm, powj, captj, capsj
        if (iwarn .ge. 10) then
          write(unit=*, fmt=*) ' EXECUTION TERMINATED - TYPE 50 ERROR'
          write(unit=luw, fmt=*) ' EXECUTION TERMINATED - TYPE 50 ERROR'
          end if
C
C *** CALCULATE TRANSIENT & AVERAGE PERFORMANCE
# 223 "type50_pnl.for"
      end if
# 226 "type50_pnl.for"
      pow = pown(ihc) * powj
      capt = captn(ihc) * captj

```

```

        caps = cap3n(ihc) * capsj
c
c *** VERY SHORT TIME CONSTANT, FINAL & AVG = STEADY STATE VALUES
# 229 "type50_pnl.for"
      if (dabs(caps) .gt. dabs(capt)) caps = capt
# 232 "type50_pnl.for"
      if (tau .gt. 0.0001) then
        aa = deltt / tau
        bb = exp(- aa)
        cc = (1. / aa) * (1. - bb)
      else
        bb = 0.
        cc = 0.
      end if
      captf = capt - ((capt - capti(ihp)) * bb)
      captav = capt - ((capt - capti(ihp)) * cc)
      if (ihc .eq. 1) then
        capsf = captf
        capsav = captav
      else
        capsf = caps - ((caps - capsi(ihp)) * bb)
        if (dabs(capsf) .ge. dabs(captf)) capsf = captf
        capsav = caps - ((caps - capsi(ihp)) * cc)
        if (dabs(capsav) .ge. dabs(captav)) capsav = captav
      end if
c
c *** ONLY FAN OPERATING
# 251 "type50_pnl.for"
      goto 750
# 254 "type50_pnl.for"
      700 captf = 0.
          captav = 0.
          capsf = 0.
          capsav = 0.
c
c *** CALCULATE HEAT ABSORBED/REJECTED & OUTLET TEMP OF WATER AND AIR
# 258 "type50_pnl.for"
      pow = 0.
# 261 "type50_pnl.for"
      750 hrav = (pow * 3.4121) - captav
c * INCLUDE FAN HEAT
# 262 "type50_pnl.for"
      two = twi + ((hrav * 1000.) / mgph)
# 264 "type50_pnl.for"
      if (ihc .eq. 1) then
        cap = captav + (powf * 3.4121)
        mw = 0.
      else
        if (ihc .eq. 2) then
          cap = capsav + (powf * 3.4121)
          mw = ((captav - capsav) * 1000.) / 1061.
        else
          cap = powf * 3.4121
          mw = 0.
        end if
      end if
      wo = wi + (mw / mair)
      hi = (0.24 + (0.444 * wi)) * tidb
      ho = hi + ((cap * 1000.) / mair)
      todb = ho / (0.24 + (0.444 * wo))
c
c *** NO HEATING OR COOLING
c
# 280 "type50_pnl.for"
      goto 810
# 284 "type50_pnl.for"

```

```

800 captf = 0.
    mair = 0.
    captav = 0.
    capsf = 0.
    capsav = 0.
    pow = 0.
    powf = 0.
    hrav = 0.
    two = twi
    wo = wi

C
C *** OUTPUT
# 294 "type50_pnl.for"
    todb = tidb
# 297 "type50_pnl.for"
810 out(1) = captf
    out(2) = captav
    out(3) = capsf
    out(4) = capsav
    out(5) = todb
    out(6) = wo
    out(7) = mair
    out(8) = pow
    out(9) = hrav
    out(10) = two
    out(11) = powf

C
# 308 "type50_pnl.for"
    out(12) = ihp
# 310 "type50_pnl.for"
    if (idebug .eq. 1) then
        write(unit=luw, fmt=1040)
1040 format(54h*-----*)
        write(unit=luw, fmt=1030) ihp, time, info(7), info(8), (xin(i), i
            & = 1, 11)
# 314 "type50_pnl.for"
1030 format(74h IHP
    &, i4/, f6.4, 2i3, 3f10.2, f7.0, 2f6.2, f6.4, f6.2, f6.4, f6.0, f4.2/)
    write(unit=luw, fmt=1000) capti(ihp), capts, captj, capt, captf,
    &captav, capsi(ihp), capss, capsj, caps, capsf, capsav, pows, powj
    &, pow
1000 format(f10.2, 2f10.4, 3f10.2/, f10.2, 2f10.4, 3f10.2/, 2f10.4, f10.2/)
    write(unit=luw, fmt=1010) (out(i), i = 1, 12)
1010 format(3hOUT, 5f10.2/, 3x, f10.6, 6f10.2/)
    write(unit=luw, fmt=1020) aa, bb, cc, vr, mr, va, ma, tidb, wi,
    &tiwb
# 323 "type50_pnl.for"
1020 format(5hA,B,C, 4f7.3, f7.0, f7.3, f7.0, f7.3, f6.4, f6.2/)
    write(unit=luw, fmt=1025) capti(ihp), capsi(ihp)
1025 format(5h CAPI, 2f10.4/)
    end if
1060 format(45h      TIME UNIT      TADB      WA      TRDB      WR,
    &26h      TIDB      WI      H/, f8.2, i5, 3(f8.0, f8.4), f10.2/)

C
# 330 "type50_pnl.for"
1070 format(47h      TIME UNIT      TIDB      TIWB      TWI      GPM      CFM,
    &24h      POWJ      CAPTJ      CAPSJ/, f8.2, i5, 3f6.0, 2f8.0, 3f8.2/)
# 334 "type50_pnl.for"
    return
end

```

```

subroutine type69(time, xin, out, t, dt, par, info)
implicit real*8(a-h,o-z)
dimension xin(7), par(15), out(20), info(10)
common /sim/ timeo, tfinal, delt
c      *****
c      * ZONE FAN MODEL *
c      *****
c
c      MODIFIED FAN FOR FAN-COIL UNIT. CALCULATES MIXED AIR CONDITIONS,
c      AND TURNS ON FAN IF EITHER CNTRL1 (HEATING) OR CNTRL2 (COOLING)
c      IS GREATER THAN ZERO.
c
c      TJM 9/8/89
c
c      Modified 5/26/90 by tj marseille so ventilation fan energy
c      is output even when heating or cooling are not required,
c      assuming it is occupied hours
c
c      common /lunits/ lur, luw, iform
# 16 "type69_89.for"
      if (info(7) .ge. 0) goto 100
      ni = info(3)
      np = info(4)
      info(6) = 5
      info(9) = 0
      call typeck(1, info, ni, np, 0)
100 continue
      cfm = par(1)
      pwr = par(2)
      frac = par(3)
      trdb = xin(1)
      wr = xin(2)
      tadb = xin(3)
c *** CALCULATE MIXED-AIR CONDITIONS ENTERING INDOOR COIL
# 29 "type69_89.for"
      wa = xin(4)
# 31 "type69_89.for"
      vr = (0.0252 * (trdb + 460)) * (1. + (1.6078 * wr))
      mr = ((cfm * (1. - frac)) / vr) * 60.
      va = (0.0252 * (tadb + 460)) * (1. + (1.6078 * wa))
      ma = ((cfm * frac) / va) * 60.
      mair = mr + ma
      tidb = ((ma * tadb) + (mr * trdb)) / mair
c *** CALCULATE WET-BULB TEMP
# 37 "type69_89.for"
      wi = ((ma * wa) + (mr * wr)) / mair
# 39 "type69_89.for"
      h = ((0.5583 * tidb) + (wi * ((1.0328 * tidb) + 2467.9511))) -
&17.8667
# 40 "type69_89.for"
      if ((h + 17.68) .le. 0.) then
        write(unit=*, fmt=*)
&' PSYCHROMETRIC PROBLEM IN TYPE 50, CHECK TEMP & HUMIDITY'
        write(unit=*, fmt=1060) time, info(1), tadb, wa, trdb, wr, tidb,
&wi, h
# 44 "type69_89.for"
        write(unit=luw, fmt=*)
&' PSYCHROMETRIC PROBLEM IN TYPE 50, CHECK TEMP & HUMIDITY'
        write(unit=luw, fmt=1060) time, info(1), tadb, wa, trdb, wr, tidb
&, wi, h
# 47 "type69_89.for"
1060 format(45h      TIME UNIT      TADB      WA      TRDB      WR,
&26h      TIDB      WI      H/,f8.2,i5,3(f8.0,f8.4),f10.2/)
# 50 "type69_89.for"
      end if

```

```

        y = dlog(h + 17.68)
        tiwb = 80.1415 + (y * ((-78.192) + (y * (25.0362 - (y * 1.7586))))
        &)
# 53 "type69_89.for"
        if (tiwb .gt. tidb) tiwb = tidb
        out(1) = tidb
        out(2) = tiwb
c
# 56 "type69_89.for"
        out(3) = wi
# 58 "type69_89.for"
        if ((xin(5) .le. 0.0) .and. (xin(6) .le. 0.0)
& .and. (xin(7) .le. 0.0)) then
            out(4) = 0.0
            out(5) = 0.0
        else
            out(4) = mair
            out(5) = pwr
        endif
        return
    end

```

```

C
C *****
C * ECONOMIZER CONTROLLER *
C *****
C
C *****
C
C PARAMETERS: 3
C
C 1 TZS = SETPOINT (F) - ZONE TEMP ABOVE WHICH ECONOMIZER IS USED
C 7 IMTH1 = MONTH BEFORE WHICH NO SUMMER ECONOMIZER PURGE IS USED
C 8 IMTH2 = MONTH AFTER WHICH NO SUMMER ECONOMIZER PURGE IS USED
C
C INPUTS: 14
C 1 IMONTH = MONTH OF YEAR
C 2 TAMB = AMBIENT OUTSIDE AIR TEMPERATURE
C 3 WAMB = AMBIENT OUTSIDE AIR HUMIDITY RATIO
C 4 TDB(1) = NORTH ZONE DRY BULB TEMPERATURE
C 5 W(1) = NORTH ZONE HUMIDITY RATIO
C 6 TDB(2) = SOUTH ZONE DRY BULB TEMPERATURE
C 7 W(2) = SOUTH ZONE HUMIDITY RATIO
C 8 TDB(3) = EAST ZONE DRY BULB TEMPERATURE
C 9 W(3) = EAST ZONE HUMIDITY RATIO
C 10 TDB(4) = WEST ZONE DRY BULB TEMPERATURE
C 11 W(4) = WEST ZONE HUMIDITY RATIO
C 12 TDB(5) = CORE ZONE TEMPERATURE
C 13 W(5) = CORE ZONE HUMIDITY RATIO
C 14 I WEEK = WEEKEND OVERRIDE
C
C OUTPUTS: 5
C 1 FRAC(1) = OUTSIDE AIR FRACTION FOR NORTH ZONE
C 2 FRAC(1) = OUTSIDE AIR FRACTION FOR SOUTH ZONE
C 3 FRAC(3) = OUTSIDE AIR FRACTION FOR EAST ZONE
C 4 FRAC(4) = OUTSIDE AIR FRACTION FOR WEST ZONE
C 5 FRAC(5) = OUTSIDE AIR FRACTION FOR CORE ZONE
C *****
C
C      subroutine type70(time, xin, out, t, dt, par, info)
C      implicit real*8(a-h,o-z)
C      dimension par(15), xin(14), out(20), info(12)
C      dimension frac(5), tdb(5), w(5), h(5), night(5)
C      common /sim/ time0, timef, delt
C
C *** FIRST CALL OF SIMULATION
C      common /lunits/ lur, luw, iform
C # 47 "type70_89a.for"
C      if (info(7) .ge. 0) goto 100
C      ni = info(3)
C      np = info(4)
C      info(6) = 2
C      call typeck(1, info, ni, np, 0)
C
C *** SET PARAMETERS
C # 52 "type70_89a.for"
C 100 continue
C # 55 "type70_89a.for"
C      tzs = par(1)
C      imth1 = par(2)
C
C *** SET INPUTS
C # 57 "type70_89a.for"
C      imth2 = par(3)
C # 60 "type70_89a.for"
C      imonth = idint(xin(1))

```

```

        tamb = xin(2)
        wamb = xin(3)
        j = 3
        do 1 i = 1, 5
            tdb(i) = xin(i + j)
            w(i) = xin((i + j) + 1)
            j = j + 1
1        continue
        iweek = idint(xin(14))
C
C *** INITIALIZE VENTILATION FRACTIONS
C
# 70 "type70_89a.for"
    hour = time - (dint(time / 24.) * 24.)
# 74 "type70_89a.for"
    do 6 i = 1, 5
        if ((hour .gt. 18.0) .or. (hour .lt. 7.0)) then
            frac(i) = 0.0
            night(i) = -1
        else
            frac(i) = 0.1178
            night(i) = 1
        end if
C
C     CHECK TO SEE IF MONTH FALLS IN RANGE SPECIFIED FOR ECONOMIZER
C
# 82 "type70_89a.for"
    6 continue
C
C
C     ENTHALPY OF OUTSIDE AIR
C
# 86 "type70_89a.for"
    if ((imonth - imth1) .lt. 0.) .or. ((imonth - imth2) .gt. 0.))
        &goto 2000
# 91 "type70_89a.for"
    hamb = (0.24 * tamb) + (wamb * (1061 + (0.444 * tamb)))
C
C     ENTHALPY OF ZONE
C
# 92 "type70_89a.for"
    do 2 i = 1, 5
# 96 "type70_89a.for"
        h(i) = (0.24 * tdb(i)) + (w(i) * (1061 + (0.444 * tdb(i))))
        if ((hamb .ge. h(i)) .or. (tdb(i) .le. 72.)) then
            goto 10
C
C
# 99 "type70_89a.for"
    end if
# 102 "type70_89a.for"
    if (((hour .ge. 4.0) .and. (hour .lt. 7.0)) .and. (tdb(i) .ge. tzs
&)) then
# 103 "type70_89a.for"
        frac(i) = -1.0
        goto 10
    end if
C
C     LIMIT FRAC SO AIR ENTERING HEAT PUMP IS 57 F
C
# 106 "type70_89a.for"
    if (night(i)) 10, 7, 7
# 110 "type70_89a.for"
    7 if (tamb .lt. 57.0) then
        frac(i) = (tdb(i) - 57.0) / (tdb(i) - tamb)
    else

```

```

        frac(i) = 1.0
      end if
    10 continue
    2 continue
C
C *** OUTPUTS
# 117 "type70_89a.for"
2000 continue
# 120 "type70_89a.for"
      do 5 i = 1, 5
        out(i) = frac(i)
C
C   RESET VALUES IF WEEKEND (NO VENTILATION)
# 122 "type70_89a.for"
      5 continue
# 125 "type70_89a.for"
      if (iweek .eq. 0) then
        out(1) = 0.0
        out(2) = 0.0
        out(3) = 0.0
        out(4) = 0.0
        out(5) = 0.0
      end if
      return
      end

```



```

C
C      *****
C      * FLOW MIXING MODEL *
C      *****
C
C      WATER MIXING TYPE, NEEDED TO REDUCE NUMBER OF UNITS IN FIVE
C      ZONE CHILLED WATER/HOT WATER CENTRAL SYSTEM BUILDING MODEL
C
C      TJM 9/10/89
C
C      subroutine type72(time, xin, out, t, dt, par, info)
C      implicit real*8(a-h,o-z)
C      dimension xin(20), par(15), out(20), info(10)
C      dimension xmdot(10), temp(10)
C      common /sim/ timeo, tfinal, delt
C      common /lunits/ lur, luw, iform
C
C      # 17 "type72_89.for"
C      if (info(7) .ge. 0) goto 100
C      np = info(4)
C      ni = info(3)
C      info(6) = 2
C      info(9) = 0
C      call typeck(1, info, ni, np, 0)
C
C      100 continue
C      totflo = par(1)
C      inp = par(2)
C      j = 0
C      xmtot = 0.0
C      do 1 i = 1, inp
C      temp(i) = xin(i + j)
C      xmdot(i) = xin((i + 1) + j)
C      xmtot = xmtot + xmdot(i)
C      j = j + 1
C
C
C      # 33 "type72_89.for"
C      1 continue
C
C      # 35 "type72_89.for"
C      3 out(1) = xmtot
C      product = 0.0
C      do 4 i = 1, inp
C      product = product + (xmdot(i) * temp(i))
C
C      4 continue
C      if (xmtot .ge. 0.0) then
C      out(2) = product / xmtot
C      else
C      out(2) = product / totflo
C      end if
C      end

```

APPENDIX D

THE MEAN RADIANT TEMPERATURE MODULE

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THE MEAN RADIANT TEMPERATURE MODULE

The BSTS system is capable of causing the floors of occupied zones in the BSTS building to be at significantly different temperatures than the ambient air temperatures in the zones. Because of these temperature differences, the mean radiant temperature (MRT) experienced by building occupants in one of the building zones can be different from the ambient air temperature in the zone. To understand the engineering significance of this aspect of the BSTS, it is important to understand the relationship between mean radiant temperature and human comfort. In the following paragraphs, this relationship is discussed. Also, a TRNSYS module developed to calculate the time-varying mean radiant temperatures in zones in a simulation building is presented. Finally, the TRNSYS MRT module is used in simulations of the BSTS building in six climates to provide some information on the range of MRTs and operative temperatures (discussed later) that will be seen in the BSTS buildings in these climates.

Factors that influence human thermal comfort or satisfaction with the thermal environment, include ambient air temperature, mean radiant temperature, air velocity and humidity, as well as personal clothing and activity. The mean radiant temperature of a subject is a measure of the radiant thermal exchange between that subject and the surroundings. For normal activities and relatively low air velocity and humidity conditions, the mean radiant temperature can have as much impact on thermal comfort as ambient air temperature does.

ASHRAE Standard 55-1981 (ASHRAE 1981) establishes winter and summer comfort zones or ranges of operative temperatures and humidity conditions for which comfort is acceptable for certain clothing and activity levels. The operative temperature used in this determination of comfort is a weighted average of ambient and mean radiant temperatures. For air speeds less than 80 ft/min and MRTs less than 120°F, the operative temperature can be estimated

with a simple average of the ambient and mean radiant temperatures. Thus, knowledge of the ambient and mean radiant temperatures in a zone can give some indication of the thermal comfort level of individuals in the zone.

To monitor mean radiant temperatures in BSTS building zones, an MRT calculation module was developed. The module incorporates calculation methods presented by Olesen et al. (1989). Table D.1 provides a listing of the source code developed for the MRT module. The MRT module calculates the mean radiant temperature of a seated subject in a user-defined rectangular room within the zone of interest. The room is allowed one window. Temperatures of the six surfaces in the room are required as input values for the module. Also, the window temperature must be given if the zone has a window. The module also requires specific dimensional parameters to describe the room.

The mean radiant temperature is calculated from the plane radiant temperatures in six directions from the point where the subject is placed in the room using Equation (D.1). The plane radiant temperature in each direction is functionally related to the temperatures of the surfaces it faces multiplied by the view factors between the plane and the respective surfaces. Olesen et al. (1989) provide more details on this. Plane radiant temperatures are calculated for small planes located at the center of the seated subject and facing each of the six surfaces in the room. Equation (D.1) weighs the plane radiant temperatures in each of the six directions according to respective area factors for a seated person. Some flexibility is allowed in the subject's location in the room. The module does not assume a specific orientation for the seated subject, so the weighing coefficient used for the side plane radiant temperatures is a mean value for front, side, and back exposures of a seated individual.

$$T_{mrt} = 0.127*(T_{pr}[i[] + T_{pr}[down]) + 0.186*(T_{pr}[right] + T_{pr}[left] + T_{pr}[up] + T_{pr}[down]) \quad (D.1)$$

where T_{mrt} = mean radiant temperature

T_{pr} = plane radiant temperature in direction indicated in brackets.

TABLE D.1. TRNSYS Mean Radiant Temperature Module

SUBROUTINE TYPE60 (TIME,XIN,OUT,T,DTDT,PAR,INFO)

```

C      *****
C      *  ROOM MEAN RADIANT TEMPERATURE MODULE  *
C      *****
C
C      JS SCHLIESING
C
C      *****
C
C      PARAMETERS:  12
C
C          1, IU    - MRT MODULE (UNIT) IDENTIFICATION NUMBER (1-10)
C          2, L     - ROOM LENGTH (FT) (WALLS 2 & 4)
C          3, W     - ROOM WIDTH (FT) (WALLS 1 & 3)
C          4, H     - ROOM HEIGHT (FT)
C          5, FR    - LOCATION FRACTION FOR SEATED PERSON IN 1-3 DIRECTION
C          6, WW    - WINDOW WIDTH (FT) (WINDOW ON SURFACE 1)
C          7, WH    - WINDOW HEIGHT ABOVE HC (FT) (BOTTOM LEVEL WITH HC)
C          8, HC    - HEIGHT FROM FLOOR TO CENTER OF PERSON (FT)
C          9, TS    - HOUR TO START PRINTING
C         10, TE    - HOUR TO STOP PRINTING
C         11, IOUT  - FORTRAN OUTPUT UNIT NUMBER
C         12, IAVG - 0:  PRINT INST TEMP, 1:  PRINT AVG HOURLY TEMP.
C
C      INPUTS:  11
C
C          1-6, ..... - TEMPERATURES OF SURFACES 1-6 (F), (FRONT, RT SIDE,
C                        BACK, LT SIDE, FLOOR, CEILING)
C          7, ..... - WINDOW TEMPERATURE (F)
C          8, TAMEX  - OUTDOOR AMBIENT TEMPERATURE (F)
C          9, TAMRM  - ROOM AMBIENT TEMPERATURE (F)
C         10, HUMRM  - ROOM HUMIDITY RATIO
C         11, TBULK  - SLAB BULK TEMPERATURE (F)
C
C      OUTPUTS:  0
C
C      *****
C
C      DIMENSION XIN(11),PAR(12),INFO(9),F(10,6,7),DAT(10,7),TPR(6)
C      DIMENSION SDAT(10,7)
C      COMMON /SIM/TIME0,TIMEF,DELT
C      REAL L,W,H,FR,L1,L2,H1,H2
C      F DIMENSIONS:  (IU) X (# OF DIRECTIONS) X (# OF SURFACES)
C      IU      = PAR(1)
C      L       = PAR(2)
C      W       = PAR(3)
C      H       = PAR(4)
C      FR      = PAR(5)

```

TABLE D.1. (contd)

```

WW      = PAR(6)
WH      = PAR(7)
HC      = PAR(8)
TS      = PAR(9)
TE      = PAR(10)
IOUT    = PAR(11)
IAVG    = PAR(12)
TAMEX   = XIN(8)
TAMRM   = XIN(9)
HUMRM   = XIN(10)
TBULK   = XIN(11)
L1      = FR*L
L2      = L=L1
H1      = HC
H2      = H-H1

```

C
C
C

*** FIRST CALL OF SIMULATION

```

IF ( INFO(7) .EQ. -1 ) THEN
  NI = 11
  NP = 12
  ND = 0
  INFO(6) = 0
  CALL TYPECK(1,INFO,NI,NP,ND)

```

C
C
C

AREA FACTORS LOOKING FROM SMALL PLANE IN DIRECTION 1

```

IF ((WH .EQ. 0.0) .AND. (22 .EQ. 0.0)) THEN
  F(IU,1,7) = 0.0

```

ELSE

```

  F(IU,1,7) = 2.*AF(1,2H,22/2.,L1)

```

ENDIF

```

F(IU,1,1) = 2.*AF(1,H1,2/2.,L1) + 2.*AF(1,H2,2/2.,L1) -

```

&

```

  F(IU,1,7)

```

```

F(IU,1,2) = AF(2,L1,H1,W/2.) + AF(2,L1,H2,W/2.)

```

```

F(IU,1,3) = 0.

```

```

F(IU,1,4) = F(IU,1,2)

```

```

F(IU,1,5) = 2.*AF(2,L1,W/2.,H1)

```

```

F(IU,1,6) = 2.*AF(2,L1,W/2.,H2)

```

C
C

AREA FACTORS LOOKING FROM SMALL PLANE IN DIRECTION 2

```

IF ((WH .EQ. 0.0) .AND. (WW .EQ. 0.0)) THEN

```

```

  F(IU,2,7) = 0.0

```

ELSE

```

  F(IU,2,7) = AF(2,WW/2.,WH,L1)

```

ENDIF

```

F(IU,2,1) = AF(2,W/1.,H1,L1) + AF(2,W/2.,H2,L1) -

```

&

```

  F(IU,2,7)

```

TABLE D.1. (contd)

```

&      F(IU,2,2) = AF(1,H1,L1,W/2.) + AF(1,H1,L2,W/2.) +
      AF(1,H2,L1,W/2.) + AF(1,H2,L2,W/2.)
      F(IU,2,3) = AF(2,W/2.,H1,L2) + AF(2,W/1.,H2,L2)
      F(IU,2,4) = 0.
      F(IU,2,5) = AF(2,W/2.,L1,H1) + AF(2,W/2.,L2,H1)
      F(IU,2,6) = AF(2,W/2.,L1,H2) + AF(2,W/1.,L2,H2)

C
C      AREA FACTORS LOOKING FROM SMALL PLANE IN DIRECTION 3
      F(IU,3,7) = 0.
      F(IU,3,1) = 0.
      F(IU,3,2) = AF(2,L2,H1,W/2.) + AF(2,L2,H2,W/2.)
      F(IU,3,3) = 2.*AF(1,H1,W/2.,L2) + 2.*AF(1,H2,W/2.,L2)
      F(IU,3,4) = F(IU,3,2)
      F(IU,3,5) = 2.*AF(2,L2,W/2.,H1)
      F(IU,3,6) = 2.*AF(2,L2,W/2.,H2)

C
C      AREA FACTORS LOOKING FROM SMALL PLANE IN DIRECTION 4
      IF ((WH .EQ. 0.0) .AND. (WW .EQ. 0.0)) THEN
        F(IU,4,7) = 0.0
      ELSE
        F(IU,4,7) = F(IU,2,7)
      ENDIF
      F(IU,4,1) = F(IU,2,1)
      F(IU,4,2) = 0.
      F(IU,4,3) = F(IU,2,3)
      F(IU,4,4) = F(IU,2,2)
      F(IU,4,5) = F(IU,2,5)
      F(IU,4,6) = F(IU,2,6)

C
C      AREA FACTORS LOOKING FROM SMALL PLANE IN DIRECTION 5
      F(IU,5,7) = 0.
      F(IU,5,1) = 2.*AF(2,H1,W/2.,L1)
      F(IU,5,2) = AF(2,H1,L1,W/2.) + AF(2,H1,L2,W/2.)
      F(IU,5,3) = 2.*AF(2,H1,W/2.,L2)
      F(IU,5,4) = F(IU,5,2)
      F(IU,5,5) = 2.*AF(1,W/2.,L1,H1) + 2.*AF(1,W/2.,L2,H1)
      F(IU,5,6) = 0.

C
C      AREA FACTORS LOOKING FROM SMALL PLANE IN DIRECTION 6
      IF ((WH .EQ. 0.0) .AND. (WW .EQ. 0.0)) THEN
        F(IU,6,7) = 0.0
      ELSE
        F(IU,6,7) = 2.*AF(2,WH,WW/2.,L1)
      ENDIF
      F(IU,6,1) = 2.*AF(2,H2,W/2.,L1) - F(IU,6,7)
      F(IU,6,2) = AF(2,H2,L1,W/2.) + AF(2,H2,L2,W/2.)
      F(IU,6,3) = 2.*AF(2,H2,W/2.,L2)

```

TABLE D.1. (contd)

```

C      F(IU,6,4) = F(IU,6,2)
C      F(IU,6,5) = 0.
C      F(IU,6,6) = 2.*AF(1,W/2.,L1,H2) + 2.*AF(1,W/2.,L2,H2)
C
C      CHECK CALCULATION OF AREA FACTORS
C      DO I=1,6
C          SUM = 0.0
C          DO J=1,7
C              SUM = SUM + F(IU,I,J)
C          END DO
C          WRITE(*,2) IU,SUM,I
C          FORMAT(1X,'MRT UNIT: ',I2,' AREA FACTOR SUM; ',F6,3,
C              &      'DIRECTION: ',I2)
C      END DO
C
C      DAT(IU,1) = TIME
C      DAT(IU,2) = 0.0
C      DAT(IU,3) = 0.0
C      DAT(IU,4) = 0.0
C      DAT(IU,5) = 0.0
C      DAT(IU,6) = 0.0
C      DAT(IU,7) = 0.0
C      SDAT(IU,1) = TIME
C      SDAT(IU,2) = 0.0
C      SDAT(IU,3) = 0.0
C      SDAT(IU,4) = 0.0
C      SDAT(IU,5) = 0.0
C      SDAT(IU,6) = 0.0
C      SDAT(IU,7) = 0.0
C      RETURN
C      ENDIF
C
C      *** PRINT OUT MEAN RADIANT TEMPERATURE SAVED FROM LAST TIME STEP
C      IF ( INFO(7) .EQ. 0 ) THEN
C          IF (IAVG .EQ.1) THEN
C              GET AVERAGE HOURLY VALUES AND PRINT
C              SDAT(IU,1) = DAT(IU,1)
C              SDAT(IU,2) = SDAT(IU,2) + DAT(IU,2)
C              SDAT(IU,3) = SDAT(IU,3) + DAT(IU,3)
C              SDAT(IU,4) = SDAT(IU,4) + DAT(IU,4)
C              SDAT(IU,5) = SDAT(IU,5) + DAT(IU,5)
C              SDAT(IU,6) = SDAT(IU,6) + DAT(IU,6)
C              SDAT(IU,7) = SDAT(IU,7) + DAT(IU,7)
C              TDAT = SDAT(IU,1)
C

```


TABLE D.1. (contd)

```

C      ROUND TIME TO THIRD DECIMAL
      TDAT = FLOAT(NINT(TDAT*1000.))/1000.
C
C      PRINT AVERAGES ON THE HOUR EVERY HOUR
      IF ( AMOD(TDAT,1.) .EQ. 0.0) THEN
        IF ( SDAT(IU,1) .NE. TIME0) THEN
          WRITE (IOUT,10) SDAT(IU,1), (SDAT(IU,1)*DELT,I=2,7)
        ELSE
          WRITE (IOUT,10) SDAT(IU,1), (SDAT(IU,I),I=2,7)
        ENDIF
        DO 5 I=2,7
          SDAT(IU,I) = 0.0
5      CONTINUE
      ENDIF
    ELSE
C
C      NO NEED TO AVERAGE TEMPERATURES
      IF ( (DAT(IU,1) .GE. TS) .AND. (DAT(IU,1) .LE. TE) ) THEN
        WRITE (IOUT,10) (DAT(IU,I),I=1,7)
10      FORMAT(1X,5(F8.3,3X),F8,6,3X,F8.3)
      ENDIF
    ENDIF
  ENDIF
ENDIF

C
C      CALCULATE PLANE RADIANT TEMPERATURES AND MEAN RADIANT TEMPERATURE
C      USING VIEW FACTORS CALCULATED IN FIRST CALL TO UNIT
      DO I = 1,6
        TPR(I) = 0.0
        DO J = 1,7
          TPR(8) = TPR(I) + (XIN(J)+459.67)**4*F(IU,I,J)
        END DO
        TPR(I) = TPR(I)**0.25 - 459.67
      END DO

C
C      MEAN RADIANT TEMPERATURE CALCULATED FOR SEATED PERSON WITH UNKNOWN
C      ORIENTATION (OLESEN, 1989).
C
      TMRT = 0.127*( TPR(5) + TPR(6) ) + 0.186 * ( TPR(1) +
&      TPR(2) + TPR(3) +TPR(4) )

C
C      STORE CALCULATION RESULTS FOR THIS UNIT TIMESTEP AND ITERATIVE CALL.
C      FINAL RESULTS FOR A GIVEN TIMESTEP ARE PRINTED AT THE BEGINNING
C      OF THE NEXT TIMESTEP.
C

```

TABLE D.1. (contd)

```

      DAT(IU,1) = TIME
      DAT(IU,2) = TMRT
      DAT(IU,3) = TAMEX
      DAT(IU,4) = TAMRM
      DAT(IU,5) = XIN(5)
      DAT(IU,6) = HUMRM
      DAT(IU,7) = TBULK
      RETURN
      END

C
      FUNCTION AF(IOR,AA,BB,CC)
      PI = 3.1415927

C
C      FUNCTION TO DETERMINE THE AREA FACTOR (AF) BETWEEN A SMALL AREA AND
C      A LARGER AREA PARALLEL (IOR=1) OR PERPENDICULAR (IOR=2) TO IT.
C
      IF ( IOR .EQ. 1 ) THEN
C      PARALLEL SURFACES
        X = AA/CC
        Y = BB/CC
        AF = 1./2./PI * ( X/SQRT(1.+X**2)*ATAN(Y/SQRT(1.+X**2)) +
&      Y/SQRT(1.+Y**2)*ATAN(X/SQRT(1.+Y**2)) )
      ELSEIF ( IOR .EQ. 2 ) THEN
C      PERPENDICULAR SURFACES
        X = AA/BB
        Y = CC/BB
        AF = 1./2./PI * ( ATAN(1./Y) - Y/SQRT(X**2+Y**2)*
&      ATAN(1./SQRT(X**2+Y**2)) )
      ELSE
C      ERROR IN FUNCTION USAGE
        WRITE (*,*) 'ERROR IN USAGE OF FUNCTION AF'
        STOP
      ENDIF
      RETURN
      END

```

T_{mrt} = Mean Radiant Temperature

T_{pr} = Plane Radiant Temperature in direction indicated in brackets

In a given TRNSYS simulation, the user is allowed to define as many as 10 separate MRT units. Each unit will write MRT calculation results and other relevant zone temperature information to a user-selected logical unit. The user can choose between output at every TRNSYS time step and output of hourly average values after every simulation hour.

To examine the MRTs associated with the BSTS system operation in each of the six study locations, the BSTS input decks were modified with the addition of three MRT units each. MRT units were defined for the core, north, and south zones. The input instructions for the respective building zone Type 19 units were modified to provide output of the surface temperatures for input to the MRT units.

Simulations of the BSTS buildings were completed using average-day weather data for each month of the year. Figures D.1 through D.6 show the daily variation of core-zone MRT (T_{mrt}), operative temperature (T_{opr}), floor temperature (T_{flr}), and ambient air temperature (T_{amb}) for the months of January, April, and July in each study location. In all cases, the operative temperature is within 2°F of the zone ambient temperature, suggesting little need for concern about adverse comfort effects from mean radiant temperatures for the current BSTS control strategies.

In the current research effort, the MRT was used solely as a diagnostic tool. Future efforts will be able to use the MRT module as part of the design process. For example, energy savings can often be achieved by setting zone ambient temperature setpoints above or below normal levels. However, this can cause ambient temperatures to fall outside the comfort zone. If surface temperatures in the room can be adjusted so that the MRT causes the operative temperature to fall within the comfort zone, the change in setpoints could be acceptable. The MRT module will allow a thorough investigation of such strategies.

FIGURE D.1. Seattle Mean Radiant Temperatures

FIGURE D.2. El Paso Mean Radiant Temperatures

FIGURE D.3. Los Angeles Mean Radiant Temperatures

FIGURE D.4. Houston Mean Radiant Temperatures

FIGURE D.5. Washington, D.C. Mean Radiant Temperatures

FIGURE D.6. Milwaukee Mean Radiant Temperatures

REFERENCES

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Olesen, B. W., J. Rosendahl, L. N. Kalisperis, L. H. Summers, and M. Steinman. 1989. "Methods for Measuring and Evaluating the Thermal Radiation in a Room." *ASHRAE Transactions* 95(1).

APPENDIX E

MONTHLY ENERGY CONSUMPTION FOR HVAC COMPONENTS:
BASE CASE COMPARED TO BSTS SYSTEM WITH SIMPLE CONTROL

APPENDIX E

MONTHLY ENERGY CONSUMPTION FOR HVAC COMPONENTS: BASE CASE COMPARED TO BSTS SYSTEM WITH SIMPLE CONTROL

The following bar graphs portray simulated monthly energy consumption results of HVAC system components for five U.S. cities in both the base case building model and the BSTS building model with simple control. This information is supplementary to Section 4.0 of the main report, in which total annual building energy consumption results are discussed. The five cities are Los Angeles, El Paso, Houston, Washington, D.C., and Milwaukee.

In Figures E.1 through E.5, the total heat pump monthly energy consumption for each city is presented. Following these, Figures E.6 through E.10 are for the boiler, and Figure E.11 through E.15 are for the cooling tower.

FIGURE E.1. Heat Pump Monthly Energy Consumption for Base Building Compared to BSTS Building with Simple Control, El Paso

FIGURE E.2. Heat Pump Monthly Energy Consumption for Base Building Compared to BSTS Building with Simple Control, Houston

FIGURE E.3. Heat Pump Monthly Energy Consumption for Base Building Compared to BSTS Building with Simple Control, Los Angeles

FIGURE E.4. Heat Pump Monthly Energy Consumption for Base Building Compared to BSTS Building with Simple Control, Milwaukee

FIGURE E.5. Heat Pump Monthly Energy Consumption for Base Building Compared to BSTS Building with Simple Control, Washington, D.C.

FIGURE E.6. Boiler Monthly Energy Consumption for Base Building Compared to BSTS Building with Simple Control, El Paso

FIGURE E.7. Boiler Monthly Energy Consumption for Base Building Compared to BSTS Building with Simple Control, Houston

FIGURE E.8. Boiler Monthly Energy Consumption for Base Building Compared to BSTS Building with Simple Control, Los Angeles

FIGURE E.9. Boiler Monthly Energy Consumption for Base Building Compared to BSTS Building with Simple Control, Milwaukee

FIGURE E.10. Boiler Monthly Energy Consumption for Base Building Compared to BSTS Building with Simple Control, Washington, D.C.

FIGURE E.11. Cooling Tower Monthly Energy Consumption for Base Building Compared to BSTS Building with Simple Control, El Paso

FIGURE E.12. Cooling Tower Monthly Energy Consumption for Base Building Compared to BSTS Building with Simple Control, Houston

FIGURE E.15. Cooling Tower Monthly Energy Consumption for Base Building Compared to BSTS Building with Simple Control, Washington, D.C.

APPENDIX F

ANNUAL ENERGY CONSUMPTION FOR HVAC COMPONENTS:
BASE CASE COMPARED TO BSTS WITH SIMPLE CONTROL

APPENDIX F

ANNUAL ENERGY CONSUMPTION FOR HVAC COMPONENTS: BASE CASE COMPARED TO BSTS WITH SIMPLE CONTROL

The following bar graphs portray simulated annual energy consumption results of HVAC system components for six U.S. cities in both the base case building model and the BSTS building model with simple control. This information is supplementary to Section 4.0 of the main report, in which total building energy consumption results are discussed. The six cities are Seattle, Los Angeles, El Paso, Houston, Washington, D.C., and Milwaukee.

In Figure F.1, the total heat pump annual energy consumption for each city is presented. Figure F.2 is for the boiler, and Figure F.3 is for the cooling tower.

FIGURE F.1. Heat Pump Annual Energy Consumption for Base Building Compared to BSTS Building with Simple Control

FIGURE F.2. Boiler Annual Energy Consumption for Base Building Compared to BSTS Building with Simple Control

FIGURE F.3. Heat Pump Annual Energy Consumption for Base Building Compared to BSTS Building with Simple Control

APPENDIX G

GENERATION OF AVERAGE-DAY WEATHER DATA

APPENDIX G

GENERATION OF AVERAGE-DAY WEATHER DATA

TRNSYS simulations to evaluate the BSTS concept required hourly data for weather characteristics such as local dew point and dry-bulb temperatures, wind speed, and solar gain. Typical Meteorological Year (TMY) data sets were selected to provide the needed weather data because these sets were available for numerous stations in the United States, and because the data sets provided a full year (8760 hours) of typical hourly values for each of the weather characteristics.

Initial simulations of the base case and BSTS buildings were run for 1-year periods using complete sets of TMY weather characteristics. Thus, hour-by-hour variations in ambient conditions for the specific study locations were represented for an entire year. However, these 1-year simulations were extremely CPU-intensive because of the complexity of the building models required for the TRNSYS dynamic simulations. Consequently, an alternative average-day approach was developed to avoid the necessity of 1-year simulations.

The average-day approach consisted of subdividing an annual simulation run into 12 separate runs, one for each month of the year, and using an appropriate average-day weather data set for each of the 12 runs. The relative difference in energy performance among various building simulation cases could then be determined by examining the monthly average-day simulation results for each case.

Average-day weather data sets for each month were developed by averaging across days of the month the hourly values of each weather characteristic. The result was a set consisting of 24 hours of mean values for each characteristic for the given month and location. The study locations included Seattle, Los Angeles, Houston, Milwaukee, Washington, D.C., and El Paso. Figures G.1 through G.24 show hour-by-hour boxplot distributions of four weather characteristics in each of the six locations for January and July.

Each figure represents a separate characteristic and location, and boxplots at each hour in the figure represent the monthly spread in the values of the specific characteristic for that hour. The solid line drawn across the boxes represents the hour-by-hour variation in the monthly mean value of the characteristic, i.e., the value used in the average-day weather data set.

In simulations that used the average-day weather data, the simulation period for each month was set to 4 days to allow simulation startup transients to stabilize. Each of the four simulation days for a given month used the same average-day weather data. Figure G.25 shows south-zone temperature transients during the January 4-day run period for the BSTS building model simulation in Seattle. The slab bulk temperature for the zone has a value of 71°F at the beginning of the simulation. This temperature is the value provided in the simulation deck to satisfy TRNSYS initialization requirements. During the first three simulation days (hours 0 to 72), the mean bulk temperature of the slab drops as the cyclic steady-state temperature is approached, showing that the initial value of 71°F was too high. However, selection of appropriate initial values to use for all of the simulation variables would have been extremely time-consuming, so the resort was to use the repetitive TRNSYS runs to smooth out startup transients. TRNSYS output values used in the analyses were retained starting at the end of the third day in the run, at which time the effect of startup transients was assumed to have been minimized.

By using the average-day approach, only 4 days of simulations for each month were required to obtain comparative data. Thus, TRNSYS data for an annual average-day run could be developed with 48 simulated days (4 days times 12 months) while an annual run made using the complete 8760 hours of TMY data would require simulating 365 days.

The average-day weather simulations were subject to a number of limitations. Weekday/weekend distinctions were not made in the average-day runs. Thus, effects caused by weekend temperature setback and subsequent recovery could be examined only in complete annual runs. Furthermore, extreme weather conditions occurring during each month were dampened by using monthly averages. Also, day-to-day changes in weather patterns and building thermal

response to these changes could not be represented in the average day data. Nevertheless, the average-day simulations could provide a basis for monthly comparisons in case studies of the relative effects of various building configurations, HVAC system controls, and building thermal parameters. Therefore, average-day simulations were used as an intermediate study tool. Complete annual runs were used to finalize reported results and verify the significance of effects observed in the average-day runs.

a) January Weather Data

b) July Weather Data

FIGURE G.1. Hourly Distributions of Seattle Dry-Bulb Temperatures

a) January Weather Data

b) July Weather Data

FIGURE G.2. Hourly Distributions of Seattle Dew Point Temperatures

a) January Weather Data

b) July Weather Data

FIGURE G.3. Hourly Distribution of Seattle Solar Gains

a) January Weather Data

b) July Weather Data

FIGURE G.4. Hourly Distributions of Seattle Wind Speeds

a) January Weather Data

b) July Weather Data

FIGURE G.5. Hourly Distributions of Low Angeles Dry-Bulb Temperatures

a) January Weather Data

b) July Weather Data

FIGURE G.6. Hourly Distributions Los Angeles Dew Point Temperatures

a) January Weather Data

b) July Weather Data

FIGURE G.7. Hourly Distributions of Los Angeles Solar Gains

a) January Weather Data

b) July Weather Data

FIGURE G.8. Hourly Distributions of Los Angeles Wind Speeds

a) January Weather Data

b) July Weather Data

FIGURE G.9. Hourly Distributions of Houston Dry-Bulb Temperatures

a) January Weather Data

b) July Weather Data

FIGURE G.10. Hourly Distributions of Houston Dew Point Temperatures

a) January Weather Data

b) July Weather Data

FIGURE G.11. Hourly Distributions of Houston Solar Gains

a) January Weather Data

b) July Weather Data

FIGURE G.12. Hourly Distributions of Houston Wind Speeds

a) January Weather Data

b) July Weather Data

FIGURE G.13. Hourly Distributions of Milwaukee Dry-Bulb Temperatures

a) January Weather Data

b) July Weather Data

FIGURE G.14. Hourly Distributions of Milwaukee Dew Point Temperatures

a) January Weather Data

b) July Weather Data

FIGURE G.15. Hourly Distributions of Milwaukee Solar Gains

a) January Weather Data

b) July Weather Data

FIGURE G.16. Hourly Distributions of Milwaukee Wind Speeds

a) January Weather Data

b) July Weather Data

FIGURE G.17. Hourly Distributions of Washington, D.C., Dry-Bulb Temperatures

a) January Weather Data

b) July Weather Data

FIGURE G.18. Hourly Distributions of Washington, D.C., Dew Point Temperatures

a) January Weather Data

b) July Weather Data

FIGURE G.19. Hourly Distributions of Washington, D.C., Solar Gains

a) January Weather Data

b) July Weather Data

FIGURE G.20. Hourly Distributions of Washington, D.C., Wind Speeds

a) January Weather Data

b) July Weather Data

FIGURE G.21. Hourly Distributions of El Paso Dry-Bulb Temperatures

a) January Weather Data

b) July Weather Data

FIGURE G.22. Hourly Distributions of El Paso Dew Point Temperatures

a) January Weather Data

b) July Weather Data

FIGURE G.23. Hourly Distributions of El Paso Solar Gains

a) January Weather Data

b) July Weather Data

FIGURE G.24. Hourly Distributions of El Paso Wind Speeds

FIGURE G.25. South-Zone Temperature Transients During the January 4-Day Run Period for the BSTS Building Model Simulation in Seattle

APPENDIX H

SUMMARY OF MONTHLY DEMAND PEAKS

APPENDIX H

SUMMARY OF MONTHLY DEMAND PEAKS

The following tables provide monthly demand peaks in kilowatts for each of the BSTS simulation models. This information is supplementary to Section 6.0 of the main report, in which winter/summer peak demands for each city for the BSTS models are compared to the base case. The six cities are El Paso (Table H.1), Houston (Table H.2), Los Angeles (Table H.3), Milwaukee (Table H.4), Seattle (Table H.5), and Washington, D.C. (Table H.6).

TABLE H.1. Summary of Monthly Demand Peaks, El Paso

	<u>Model</u>			
	<u>1</u>	<u>2</u>	<u>3</u>	<u>5</u>
January	733	733	734	607
February	546	348	542	541
March	302	310	328	301
April	313	734	465	490
May	316	581	444	311
June	317	315	317	310
July	319	320	319	319
August	321	321	320	319
September	319	726	315	400
October	311	733	544	541
November	539	413	515	511
December	732	732	733	733

TABLE H.2. Summary of Monthly Demand Peaks, Houston

	Model			
	<u>1</u>	<u>2</u>	<u>3</u>	<u>5</u>
January	733	646	632	601
February	546	805	646	647
March	302	805	610	573
April	312	806	401	401
May	316	413	414	412
June	317	417	416	417
July	319	416	415	416
August	321	417	417	416
September	319	418	415	415
October	311	447	408	410
November	539	806	395	395
December	732	649	647	647

TABLE H.3. Summary of Monthly Demand Peaks, Los Angeles

	Model			
	<u>1</u>	<u>2</u>	<u>3</u>	<u>5</u>
January	733	623	493	493
February	546	611	355	355
March	302	702	502	502
April	312	653	291	291
May	316	474	316	316
June	317	316	318	318
July	319	318	318	318
August	321	321	320	320
September	319	315	318	318
October	311	711	335	335
November	539	609	287	287
December	732	709	530	530

TABLE H.4. Summary of Monthly Demand Peaks, Milwaukee

	<u>Model</u>			
	<u>1</u>	<u>2</u>	<u>3</u>	<u>5</u>
January	733	856	855	855
February	546	842	764	748
March	302	733	733	733
April	312	733	732	732
May	316	733	518	543
June	317	612	337	336
July	319	485	324	328
August	321	356	326	326
September	319	733	372	478
October	311	733	731	717
November	539	816	734	733
December	732	850	846	844

TABLE H.5. Summary of Monthly Demand Peaks, Seattle

	<u>Model</u>			
	<u>1</u>	<u>2</u>	<u>3</u>	<u>5</u>
January	733	733	733	733
February	546	734	734	734
March	302	733	733	734
April	312	733	607	607
May	316	734	532	486
June	317	632	310	291
July	319	611	315	315
August	321	517	294	307
September	319	725	385	366
October	311	733	732	728
November	539	733	733	733
December	732	816	734	734

TABLE H.6. Summary of Monthly Demand Peaks, Washington, D.C.

	Model			
	<u>1</u>	<u>2</u>	<u>3</u>	<u>5</u>
January	733	775	775	780
February	546	776	775	775
March	302	776	776	776
April	312	777	625	617
May	316	776	364	363
June	317	373	376	374
July	319	380	380	380
August	321	382	383	382
September	319	645	377	377
October	311	775	631	631
November	539	776	774	774
December	732	777	776	776

The building models for which monthly demand peaks are provided include

- base (non-BSTS) building - Model 1 (Section 3.4)
- BSTS system with simple control - Model 2 (Section 4.3)
- BSTS system with slab temperature control and seasonal (winter/summer) slab temperature control setpoints - Model 3 (Section 5.2.1)
- BSTS system with slab temperature control and seasonal control setpoints, but with "series" heat pump/slab water circuit arrangements for each zone - Model 5 (Section 5.2.3).

APPENDIX I

ECONOMIC ASSESSMENT COST ESTIMATE WORKSHEET

APPENDIX I

ECONOMIC ASSESSMENT COST ESTIMATE WORKSHEET

This appendix contains a listing of the computer worksheet file used in developing the incremental installed cost estimates for the BSTS system. These costs are summarized in Table 6.4 of the main report, wherein also are referenced the pricing sources and the estimating methodologies used in creating this worksheet.

TABLE I.1. Incremental Costs of BSTS Systems (Worksheet File)

	BSTS Model				
	2	3	4	5	6
Water Piping (per floor)					
Schedule 40 Piping	6537.00	5637.00	5637.00	469.00	469.00
Pipe Fittings	1963.00	1963.00	1963.00	1356.00	1356.00
Valves (inc. control)	5147.00	5147.00	5147.00	2694.00	2694.00
Pump	3840.00	3840.00	3840.00	0.00	0.00
Subtotal A (6 floors)	104922.00	104922.00	104922.00	36242.00	36242.00
Inflation adj. (to 1990)				(#5 and #6 adj. for 1st flr)	
1986					
1987					
1988					
1989					
1990					
Subtotal B	124724.33	124724.33	124724.33	43082.09	43082.09
Profit					
Markup					
Piping Total	145366.20	145366.20	145366.20	50212.18	50212.18
Controls (total building)					
Slab RTDs	0.00	66000.00	66000.00	66000.00	11000.00
Pump Controls	1000.00	1000.00	1000.00	1000.00	1000.00
Outdoor Air Sensors	0.00	0.00	1000.00	0.00	0.00
Wtr Loop Temp. Sensors	0.00	0.00	30000.00	0.00	0.00
Cooling Tower					
Sensors	2000.00	2000.00	2000.00	2000.00	2000.00
Controls	1000.00	1000.00	1000.00	1000.00	1000.00
PC & Software	10000.00	10000.00	10000.00	10000.00	10000.00
Controls Total	14000.00	80000.00	111000.00	80000.00	25000.00
BSTS Hydronic Tubing Costs					
BSTS Tubing	22500.00	22500.00	22500.00	22500.00	22500.00
BSTS Installation	13500.00	13500.00	13500.00	13500.00	13500.00
BSTS Manifold	8760.00	8760.00	8760.00	8760.00	8760.00
BSTS Tube Total	44760.00	44760.00	44760.00	44760.00	44760.00

TABLE I.2. Geography Indexes

El Paso Milwaukee	BSTS Model	0.76 0.99		Houston Seattle		0.87 1.04		Los Angeles Washington, D.C.		1.14 0.88	
		2		3		4		5		6	
El Paso	Piping	110478.31	110478.31	110478.31	110478.31	110478.31	110478.31	38161.25	38161.25	38161.25	38161.25
	Controls	14000.00	80000.00	80000.00	111000.00	111000.00	111000.00	80000.00	80000.00	25000.00	25000.00
	BSTS Tube	44760.00	44760.00	44760.00	44760.00	44760.00	44760.00	44760.00	44760.00	44760.00	44760.00
	Totals	169238.31	235238.31	235238.31	266238.31	266238.31	266238.31	162921.25	162921.25	107921.25	107921.25
	4% maint./yr	6769.53	9409.53	9409.53	10649.53	10649.53	10649.53	6516.85	6516.85	4316.85	4316.85
Houston	Piping	125014.93	125014.93	125014.93	125014.93	125014.93	125014.93	43182.47	43182.47	43182.47	43182.47
	Controls	14000.00	80000.00	80000.00	111000.00	111000.00	111000.00	80000.00	80000.00	25000.00	25000.00
	BSTS Tube	44760.00	44760.00	44760.00	44760.00	44760.00	44760.00	44760.00	44760.00	44760.00	44760.00
	Totals	183774.93	249774.93	249774.93	280774.93	280774.93	280774.93	167942.47	167942.47	112942.47	112942.47
	4% maint./yr	7351.00	9991.00	9991.00	11231.00	11231.00	11231.00	6717.70	6717.70	4517.70	4517.70
Los Angeles	Piping	165716.47	165717.47	165717.47	165717.47	165717.47	165717.47	57241.88	57241.88	57241.88	57241.88
	Controls	14000.00	80000.00	80000.00	111000.00	111000.00	111000.00	80000.00	80000.00	25000.00	25000.00
	BSTS Tube	44760.00	44760.00	44760.00	44760.00	44760.00	44760.00	44760.00	44760.00	44760.00	44760.00
	Totals	224477.47	290477.47	290477.47	321477.47	321477.47	321477.47	182001.88	182001.88	127001.88	127001.88
	4% maint./yr	8979.10	11619.10	11619.10	12859.10	12859.10	12859.10	7280.08	7280.08	5080.08	5080.08
Milwaukee	Piping	143912.54	143912.54	143912.54	143912.54	143912.54	143912.54	49710.05	49710.05	49710.05	49710.05
	Controls	14000.00	80000.00	80000.00	111000.00	111000.00	111000.00	80000.00	80000.00	25000.00	25000.00
	BSTS Tube	44760.00	44760.00	44760.00	44760.00	44760.00	44760.00	44760.00	44760.00	44760.00	44760.00
	Totals	202672.54	268672.54	268672.54	299672.54	299672.54	299672.54	174470.05	174470.05	119470.05	119470.05
	4% maint./yr	8106.90	10746.90	10746.90	11986.90	11986.90	11986.90	6978.80	6978.80	4778.80	4778.80
Seattle	Piping	151180.85	151180.85	151180.85	151180.85	151180.85	151180.85	52220.66	52220.66	52220.66	52220.66
	Controls	14000.00	80000.00	80000.00	111000.00	111000.00	111000.00	80000.00	80000.00	25000.00	25000.00
	BSTS Tube	44760.00	44760.00	44760.00	44760.00	44760.00	44760.00	44760.00	44760.00	44760.00	44760.00
	Totals	209940.85	275940.85	275940.85	306940.85	306940.85	306940.85	176980.66	176980.66	121980.66	121980.66
	4% maint./yr	8397.63	11037.63	11037.63	12277.63	12277.63	12277.63	7079.23	7079.23	4879.23	4879.23
Washington, D.C.	Piping	127922.26	127922.26	127922.26	127922.26	127922.26	127922.26	44186.71	44186.71	44186.71	44186.71
	Controls	14000.00	80000.00	80000.00	111000.00	111000.00	111000.00	80000.00	80000.00	25000.00	25000.00
	BSTS Tube	44760.00	44760.00	44760.00	44760.00	44760.00	44760.00	44760.00	44760.00	44760.00	44760.00
	Totals	186682.26	252682.26	252682.26	283682.26	283682.26	283682.26	168946.71	168946.71	113946.71	113946.71
	4% maint./yr	7467.29	10107.29	10107.29	11347.29	11347.29	11347.29	6757.87	6757.87	4557.87	4557.87

APPENDIX J

MONTHLY ENERGY CONSUMPTION FOR AIR ECONOMIZER SYSTEM

APPENDIX J

MONTHLY ENERGY CONSUMPTION FOR AIR ECONOMIZER SYSTEM

The following bar graphs portray simulated monthly total HVAC energy consumption results for six U.S. cities in the base case building model, the BSTS building model with slab temperature control and seasonal setpoints (Model 3), and the base building with an air economizer added. This information is supplementary to Section 7.0 of the main report, in which total annual building energy consumption results for the air economizer building model are discussed. The six cities are El Pas (Figure J.1), Houston (Figure J.2), Los Angeles (Figure J.3), Milwaukee (Figure J.4), Seattle (Figure J.5), and Washington, D.C. (Figure J.6).

FIGURE J.1. Total HVAC Monthly Energy Consumption for Base Building, BSTS Building with Slab Temperature Control, and Building with Air Economizer, El Paso

FIGURE J.2. Total HVAC Monthly Energy Consumption for Base Building, BSTS Building with Slab Temperature Control, and Building with Air Economizer, Houston

FIGURE J.3. Total HVAC Monthly Energy Consumption for Base Building, BSTS Building with Slab Temperature Control, and Building with Air Economizer, Los Angeles

FIGURE J.4. Total HVAC Monthly Energy Consumption for Base Building, BSTS Building with Slab Temperature Control, and Building with Air Economizer, Milwaukee

FIGURE I.5. Total HVAC Monthly Energy Consumption for Base Building, BSTS Building with Slab Temperature Control, and Building with Air Economizer, Seattle

FIGURE J.6. Total HVAC Monthly Energy Consumption for Base Building, BSTS Building with Slab Temperature Control, and Building with Air Economizer, Washington, D.C.

APPENDIX K

MONTHLY ENERGY CONSUMPTION FOR WATER THERMAL STORAGE SYSTEM

APPENDIX K

MONTHLY ENERGY CONSUMPTION FOR WATER THERMAL STORAGE SYSTEM

The following bar graphs portray simulated monthly total HVAC energy consumption results for six U.S. cities in the base case building model, the BSTS building model with slab temperature control and seasonal setpoints (Model 3), and the base building with an water thermal storage added. This information is supplementary to Section 7.0 of the main report, in which total annual building energy consumption results for the air economizer building model are discussed. The six cities are El Paso (Figure K.1), Houston (Figure K.2), Los Angeles (Figure K.3), Milwaukee (Figure K.4), Seattle (Figure K.5), and Washington, D.C. (Figure K.6).

FIGURE K.1. Total HVAC Monthly Energy Consumption for Base Building, BSTS Building with Slab Temperature Control, and Building with Water Thermal Storage, El Paso

FIGURE K.2. Total HVAC Monthly Energy Consumption for Base Building, BSTS Building with Slab Temperature Control, and Building with Water Thermal Storage, Houston

FIGURE K.3. Total HVAC Monthly Energy Consumption for Base Building, BSTS Building with Slab Temperature Control, and Building with Water Thermal Storage, Los Angeles

FIGURE K.4. Total HVAC Monthly Energy Consumption for Base Building, BSTS Building with Slab Temperature Control, and Building with Water Thermal Storage, Milwaukee

FIGURE K.5. Total HVAC Monthly Energy Consumption for Base Building, BSTS Building with Slab Temperature Control, and Building with Water Thermal Storage, Seattle

FIGURE K.6. Total HVAC Monthly Energy Consumption for Base Building, BSTS Building with Slab Temperature Control, and Building with Water Thermal Storage, Washington, D.C.

APPENDIX L

MONTHLY ENERGY CONSUMPTION FOR CENTRAL SYSTEM

APPENDIX L

MONTHLY ENERGY CONSUMPTION FOR CENTRAL SYSTEM

The following bar graphs portray simulated monthly total HVAC energy consumption results for six U.S. cities in the base case building model, the BSTS building model with slab temperature control and seasonal setpoints (Model 3), and the building model with a central chilled water/hot water system. This information is supplementary to Section 7.0 of the main report, in which total annual building energy consumption results for the air economizer building model are discussed. The six cities are El Paso (Figure L.1), Houston (Figure L.2), Los Angeles (Figure L.3), Milwaukee (Figure L.4), Seattle, (Figure L.5), and Washington, D.C. (Figure L.6).

FIGURE L.1. Total HVAC Monthly Energy Consumption for Base Building, BSTS Building with Slab Temperature Control, and Building with Central System, El Paso

FIGURE L.2. Total HVAC Monthly Energy Consumption for Base Building, BSTS Building with Slab Temperature Control, and Building with Central System, Houston

FIGURE L.3. Total HVAC Monthly Energy Consumption for Base Building, BSTS Building with Slab Temperature Control, and Building with Central System, Los Angeles

FIGURE L.4. Total HVAC Monthly Energy Consumption for Base Building, BSTS Building with Slab Temperature Control, and Building with Central System, Milwaukee

FIGURE L.5. Total HVAC Monthly Energy Consumption for Base Building, BSTS Building with Slab Temperature Control, and Building with Central System, Seattle

FIGURE L.6. Total HVAC Monthly Energy Consumption for Base Buildings, BSTS Building with Slab Temperature Control, and Building with Central System, Washington, D.C.

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