ESL-TR-93/10-04

LoanSTAR Monitoring and Analysis Program

Potential Operation and Maintenance (O&M) Savings in the Basic Science Building at UTMB

Submitted to the
State Energy Conservation Office of Texas
by the
Monitoring Analysis Group (Task E)

Dr. Mingsheng Liu
Aamer Athar
Dr. David E. Claridge, Principal Investigator
Dr. Agami Reddy
Dr. Jeff S. Haberl

October, 1993

DISCLAIMER

EXECUTIVE SUMMARY

This report presents the results of a study of the potential energy savings due to optimizing the Heating, Ventilation and Air Conditioning (HVAC) operation schedule in the Basic Science Building at University of Texas Medical Branch (UTMB), Galveston, Texas. An optimized HVAC operation schedule has been developed using a simplified HVAC systems model analysis along with the LoanSTAR measured hourly energy use data and EMCS measured operating parameters at UTMB. An annual savings of $156,000 can be realized by implementing this optimized schedule in the EMCS control program. The majority of the energy savings are due to the reduction in chilled water consumption and a substantial reduction of reheat. Our analysis indicates that the indoor comfort level will not be degraded by this measure. These measures can reduce the building's current annual energy costs by $156,000, or 23%.

Subsequently the suggested O&M measure, i.e. raising cold deck temperature from 59 ^{o}F to 54 ^{o}F was implemented on July 2, 1993. A simple regression analysis of energy use data has confirmed that, as of October 25, 1993, Basic Science Building has saved $58,000 by way of reduced chilled water and steam consumption.

Table of Contents

POTENTIAL OPERATION AND MAINTENANCE SAVINGS IN THE BASIC SCIENCE BUILDING AT UTMB

1. INTRODUCTION

The Basic Science Building is a 137,856 ft^2, free standing seven story building. The exterior surface is made of brick and is approximately 73,000 ft^2 in area. There are 30 small (6 ft^2) windows on the east and west sides of the building. The building consists primarily of offices, classrooms, labs and storage. This building was placed in service on January, 1971 and is expected to serve UTMB for many more years.

The building is provided 75% outside air drawn in by two 150 hp constant volume dual duct AHUs, each capable of handling 110,000 cfm. Variable frequency drives were installed on these AHUs in 1992 which reduced the percentage of outside air from 100% to 75%. Currently the fans are supplying air at a rate of 1.24 cfm/ft^2. Chilled water and steam is supplied by the main chiller plant. Steam is converted into hot water by a hot water converter (8,600 lb/hr). A variable frequency drive chilled water pump (75 hp, 4,572 gpm) supplies chilled water to the AHUs. The building HVAC system is operated 24 hours a day all year long. Lighting in the building is provided exclusively by fluorescent fixtures. Lighting intensity varies widely throughout the building.

The hourly building energy consumption data (electricity, chilled water & steam) are being measured by the LoanSTAR program [1] as well as by a Steffa Energy Management & Control System (EMCS) at this building (see Appendix B for detail). According to the LoanSTAR measured results, this building consumed 3.65 million kWh, 55,500 MMBtu chilled water[1] and 33,400 MMBtu steam from July 1992 to June 1993. The total cost of these utilities comes out to be \$670,864/yr or \$4.87/ft^2. The following unit price has been used to calculate the total utility cost: \$0.02659/kWh for

[1] Chilled water consumption is predicted using a simulation model. LoanSTAR chilled water consumption data is missing from July 1992 to November 1992 due to a hardware problem.

electricity, $7.30/MMBtu for chilled water and $5.055/MMBtu for steam. Figure 1 and Table 1 show the breakdown of energy consumption and cost.

Table 1: Measured Energy Consumption and Cost
July 1992 - June 1993

	Electricity kWh	Chilled Water MMBtu	Steam MMBtu	Total
Consumption	3,647,482	55,489	33,394	
Costs ($)	$96,987	$405,069	$168,808	$670,864
% of Total Cost	14%	60%	26%	

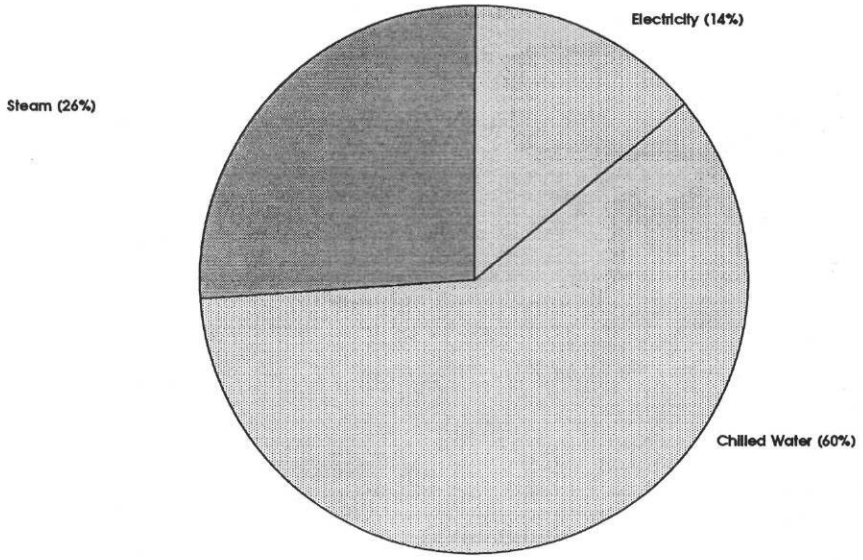

Figure 1: Energy Cost Distribution for the Basic Science Building. Total Annual Energy Cost is $670,864

Figure 2 shows measured average daily chilled water and steam energy consumption vs. ambient temperature. Substantial steam consumption exists during the hot summer days, and the consumption increases as the temperature decreases, indicating that substantial reheat is present and also reflecting a large amount of domestic hot water consumption.

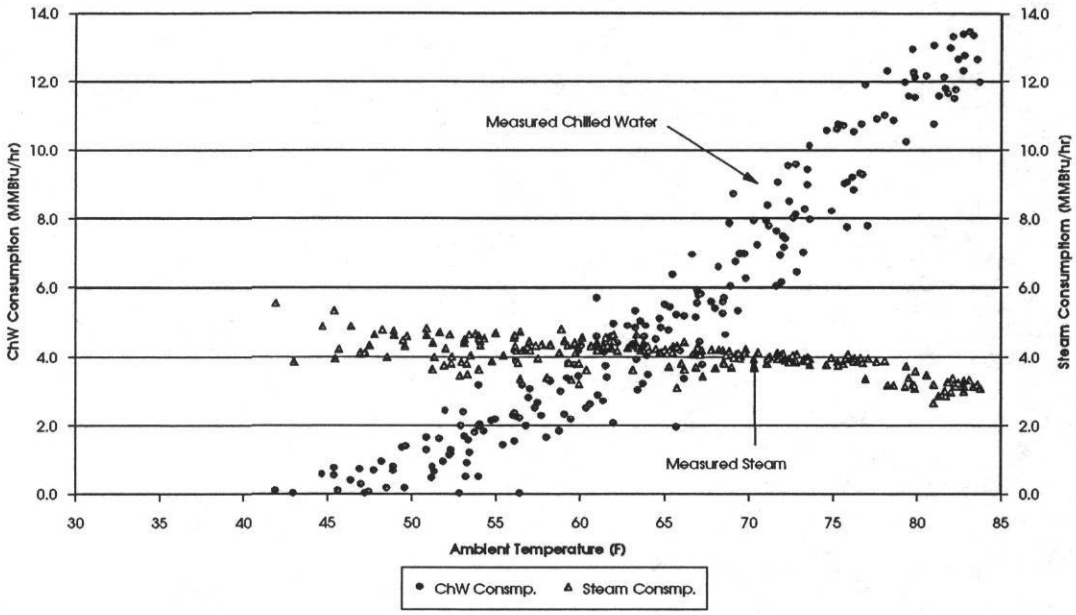

Figure 2: Measured Chilled Water and Steam Energy Consumption vs. Average Daily Ambient Temperature

Both the air handling units and their associated equipment are controlled by the newly installed Steffa Energy Management Control System. It can continuously monitor and control different parameters like cold deck temperature, hot deck temperature & speed of the fans based on space and ambient temperatures.

This report briefly describes the methodology used to identify different O&M measures. It presents the simplified HVAC system model used for the present O&M analysis and HVAC operation optimization. The selected O&M measure is discussed in detail along with recommendations and conclusions.

2. METHODOLOGY

The methodology used to explore the O&M opportunities is outlined below:

1. **LoanSTAR information base browse:** The LoanSTAR information base includes:
   - (i) the LoanSTAR Data Base (LSDB), which contains continuously measured hourly energy use and weather data;
   - (ii) the Site Description Notebook, which contains updated information about the building's Heating, Ventilation and Air Conditioning (HVAC) system, lighting, building envelope, occupancy and other relevant information from the audit report;
   - (iii) weekly inspection plots (IPNs), which give an updated performance of the building every week;
   - (iv) the Monthly Energy Consumption Report (MECR), which presents an overview of monthly energy performance ;
   - (v) the Annual Energy Consumption Report (AECR), which summarizes yearly energy performance and the overall energy performance history of the building.

   Browsing this information base gives O&M staff a draft list of O&M candidates in the building.

2. **Site visit/system examination:** The purpose of the site visit includes:
   - (i) discussing potential O&M measures with UTMB physical plant personnel;
   - (ii) verifying information gathered from LoanSTAR database by a simple walk-through with the building operator;
   - (iii) examining the possibility/feasibility of potential O&M measures;
   - (iv) exploring new O&M measures; and
   - (v) collecting system information, such as cold deck and hot deck temperature schedules, air flow, and nighttime setback schedule as well as miscellaneous information from the EMCS such as EMCS measured energy performance.

3. **Data quality check:** Before using the LoanSTAR data to estimate O&M savings, they are compared with EMCS measured data. If the two sets of data are fairly consistent,

the LoanSTAR data is used in the analysis without correction. If the LoanSTAR measured data and EMCS measured data are unacceptably different, the LoanSTAR data is checked using other methods. This quality check provides reliable data for the savings analysis. The data quality check in this building indicates that the LoanSTAR measured data are reliable (see Appendix B).

4. System modeling and calibration: The HVAC systems and the building are modeled by a set of equations, which are programmed into a computer simulation code. The simplified computer model uses measured daily average ambient and dew point temperatures to predict daily average hourly chilled water and hot water energy consumption. Finally, the predicted energy consumption is compared with measured consumption. If the predicted consumption matches measured energy consumption, then the simplified computer model and its associated parameters, such as air flow, cold deck and hot deck settings, and internal gains are calibrated. Otherwise, calibration is required which involves adjusting parameter estimates such that better agreement with monitored data is achieved.

5. O&M simulation & savings calculations: The cold deck and hot deck schedules are optimized to consume minimum energy while the following conditions are satisfied:

i) room temperature should be unchanged;
ii) room relative humidity should be less than 60%;
iii) the air flow rate to each room should not change;
iv) the maximum CFM through the cold and hot decks and ducts should be less than their capacities or design values; and
v) there should be no extra implementation costs involved.

Energy savings are taken as the difference between the base model (calibrated model) predicted annual energy consumption and optimized model predicted annual energy consumption.

6. Feedback from UTMB physical plant personnel: UTMB personnel comments on the proposed optimized schedule, and necessary information to modify the proposed schedule, if needed, is provided. The simplified model simulation may indicate that some of the EMCS measured values are incorrect. These parameters are discussed during the feedback meeting and are jointly measured by both LoanSTAR and UTMB personnel.

7. **Refinement of simulation & savings calculations:** All the suggestions and findings are incorporated into the simplified model and potential savings are recalculated.

8. **Short-term test of optimized schedule and implementation.** The fixed temperature setting for the cold deck and hot deck are derived from the optimized schedule under certain ambient temperature conditions. UTMB staff disable the EMCS system temporarily and use the suggested settings instead for a few days. Although this test would not show the full potential of optimized schedule savings, it provides an opportunity to expose some hidden problems, if any. If there are no problems after this test, the optimized schedule is programmed into the EMCS system by the UTMB staff.

3. SIMPLIFIED MODEL & CALIBRATION

3.1. Simplified Model and Input Data

The schematic of the air handling unit is shown in Figure 3. The two air handling units have a total air supply capacity of 210,000 cfm, with a total outdoor air intake of 175,000 CFM. According to the EMCS data, the following parameters were recorded on July 15, 1993 at 7:54 pm:

Cold deck temperature:	54 ^{o}F	Hot deck temperature:	84 ^{o}F
Return air temperature:	75 ^{o}F	Avg. space temperature:	72 ^{o}F

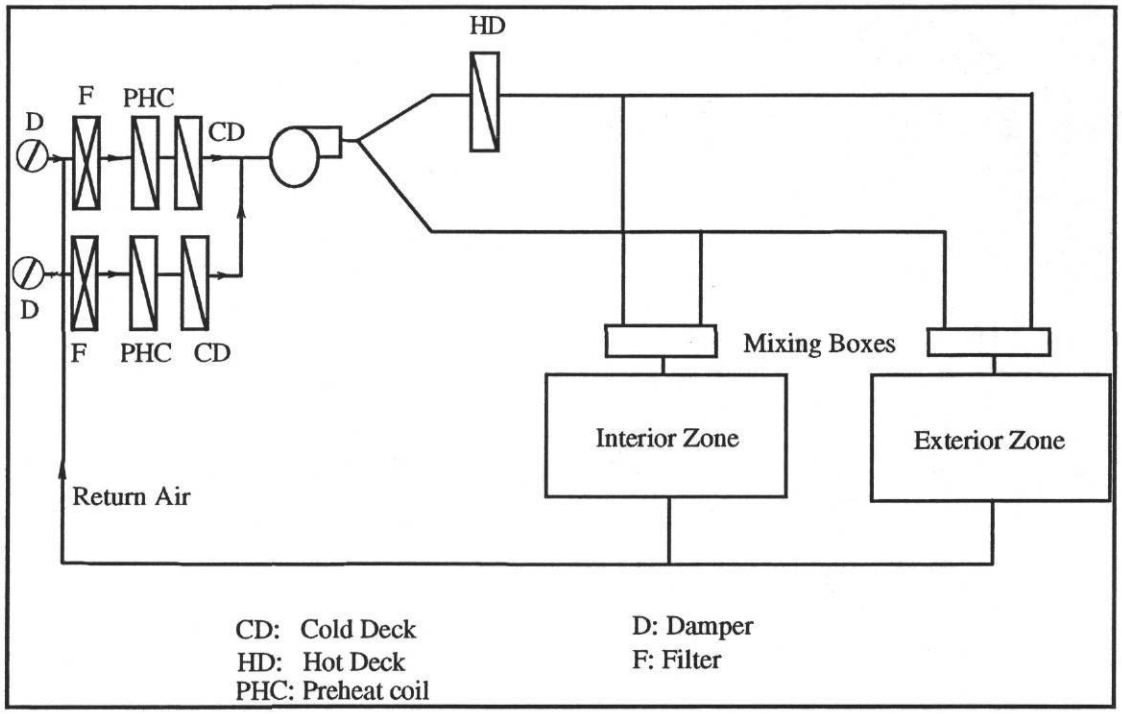

Figure 3: Schematic of Air Handling Unit

The main equations in the simplified model are presented in Appendix A. The basic parameters used in the model are discussed below.

According to the information supplied by the UTMB staff, the conditioned area is approximately 105,350 ft^2. The building has been divided into two zones: an interior zone and an exterior zone. The exterior zone is taken as the sum of areas which are directly connected with the exterior envelope. According to the building floor plans, both zones are approximately of equal size and each covers 52,675 ft^2. The internal heat gain is assumed to be 2.47 W/ft^2 while a factor of 0.8 is used to account for the heat gain reduction at night. The domestic hot water consumption is estimated to be 1.2 MMBtu/hr. Figure 4 shows a typical floor layout of Basic Science Building.

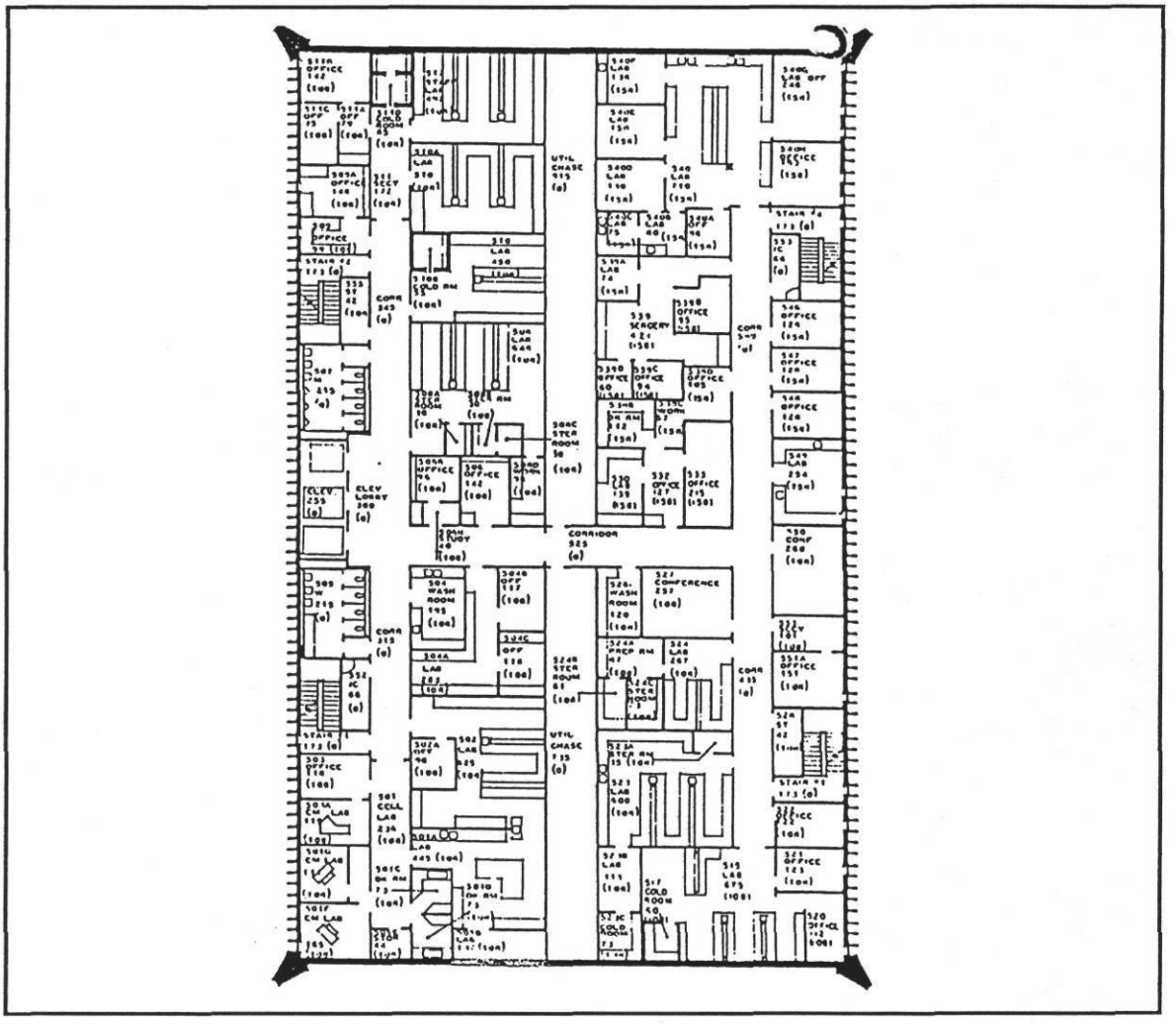

Figure 4: Typical Floor Layout of Basic Science Building

The building envelope area is calculated to be 73,000 ft^2, which includes 360 ft^2 of window area. Heat transfer coefficient value of 0.25 Btu/ft^2-^oF-hr was assumed for walls and 1 Btu/ft^2-^oF-hr for windows.

The air infiltration rate is taken as 0.4 ach (air change number of building volume in one hour) for the exterior zone and 0.2 ach for the interior zone. The interior zone receives infiltration through exterior doors and corridors.

3.2. Model Calibration

The chilled water and steam energy consumption were predicted with the simplified model using measured daily average temperature from December 1992 to June 1993. Figure 5 permits a visual comparison of the measured energy consumption with model simulated energy consumption. The horizontal axis is ambient temperature while vertical axis is daily average chilled water and steam energy consumption. It shows that the simulated data fits well with the measured data. The predicted daily average chilled water consumption was 8% lower than measured values while the predicted steam consumption was 3% lower than the measured values over a period from December 1, 1992 to June 30, 1993. The root mean square errors are 1.13 MMBtu/hr and 0.25 MMBtu/hr for chilled water and steam, respectively. The coefficient of variation are 19% and 6% for chilled water and steam, respectively.

The LoanSTAR measured steam energy consumption is also compared with EMCS measured data on hourly basis for 24 hours. Comparison of results shows that LoanSTAR measured steam consumption is within 10% of EMCS measured data (please see Appendix B for details).

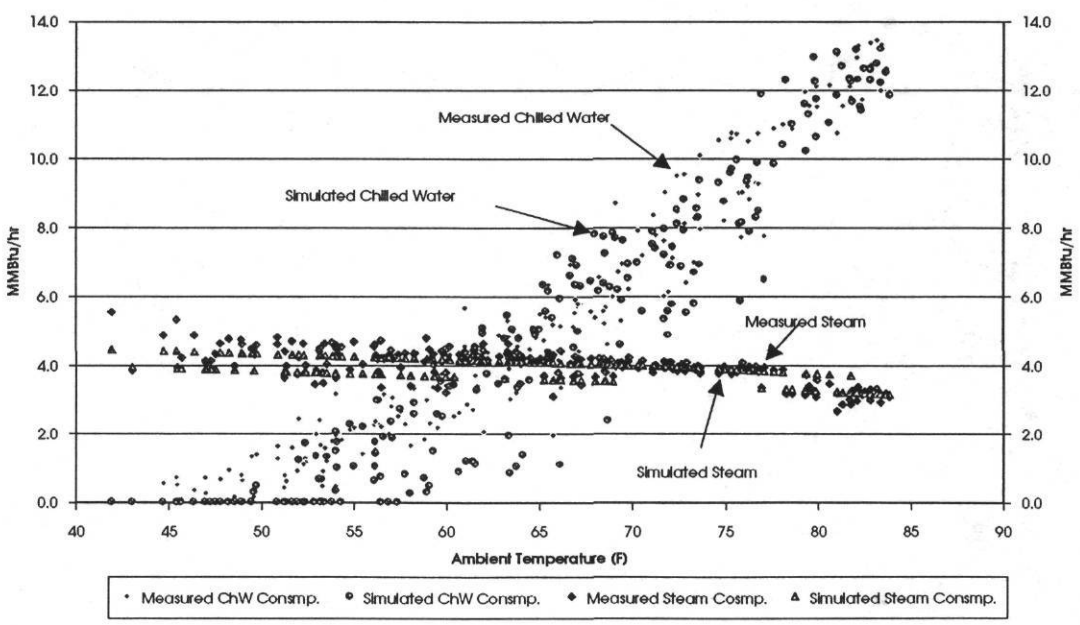

Figure 5: Comparison of Simulated and Measured Average Daily Energy Consumption (December 1992 - June 1993)

Figure 6 shows the comparison between simulated and measured energy consumption when plotted against time. It shows that the simplified model matches very well with the daily variation. The model predicted chilled water consumption, however, is lower than measured values from January 1993 to April 1993. This difference may be due to the use of average daily temperature instead of hourly data. The other reason could be the use of preheat coils (not programmed in the simulation model) at lower temperatures, which increases the air temperature at the entrance of the cold decks thus increasing the measured chilled water usage.

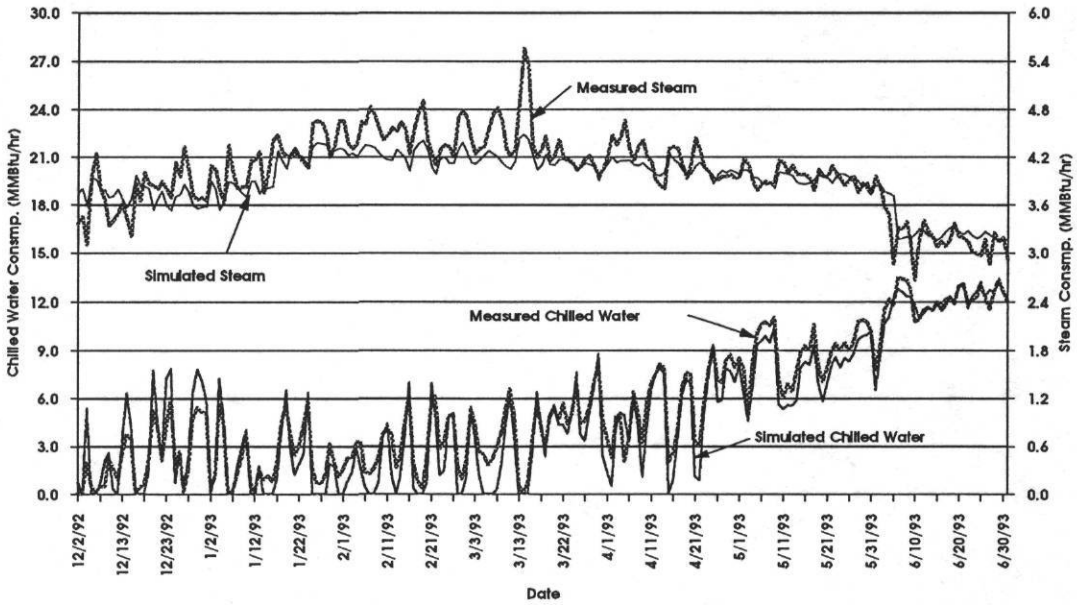

Figure 6: Comparison of Simulated & Measured Daily Average Energy Consumption (December 1992 - June 1993)

The calibrated simplified model was used to calculate annual energy consumption using bin data for outdoor temperature. Due to the lack of measured hourly dry bulb and dew point temperatures for Galveston, measured hourly data from July 1, 1992 to June 30, 1993 for Houston was used to generate bin temperatures. Figure 7 shows the number of hours in each bin. The horizontal axis is the bin temperature, where 24 bins are used with a spread of 3 °F in each bin. The number of hours under a specified temperature

during a full year are shown on the vertical axis. From the graph we can see that most of the hours are between 50 °F and 90 °F .

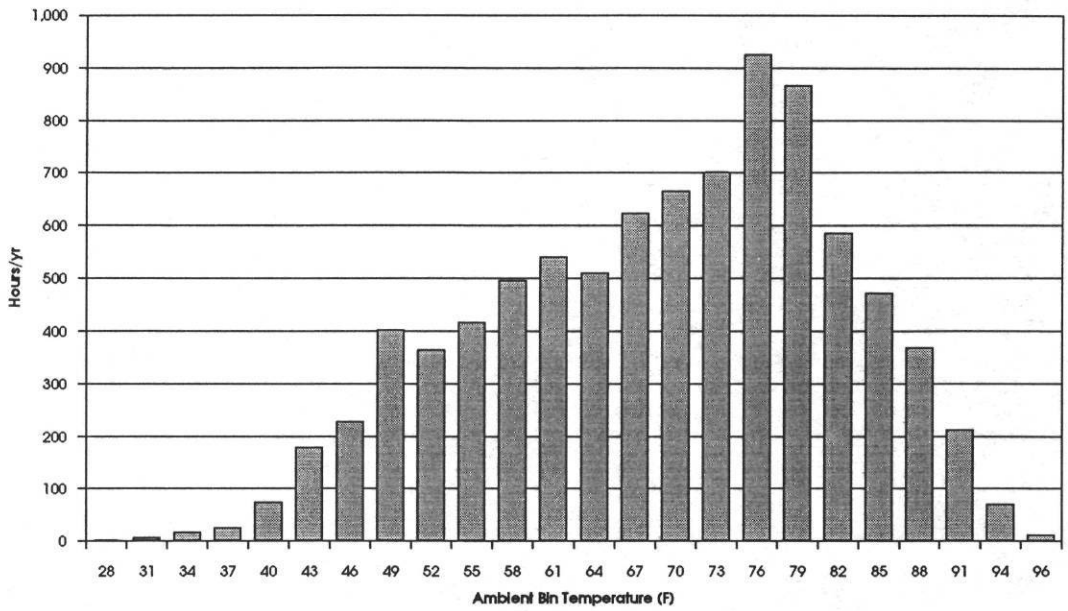

Figure 7: Houston Bin Temperature Chart generated using LoanSTAR Measured Hourly Temperature from July 1, 1992 to June 30, 1993

The coincident dew point temperatures are plotted as a function of the ambient dry bulb bin temperature in Figure 8. The figure shows that the dew point increases with the ambient temperature when the ambient temperature is lower than 80°F and then remain more or less constant when the ambient temperature is higher than 80°F. The fixed dew point temperature indicates that the absolute moisture content does not change when the ambient temperature is higher than 80 °F. Consequently, the sensible load increases with temperature while the latent load does not change when the ambient temperature is higher than 80 °F.

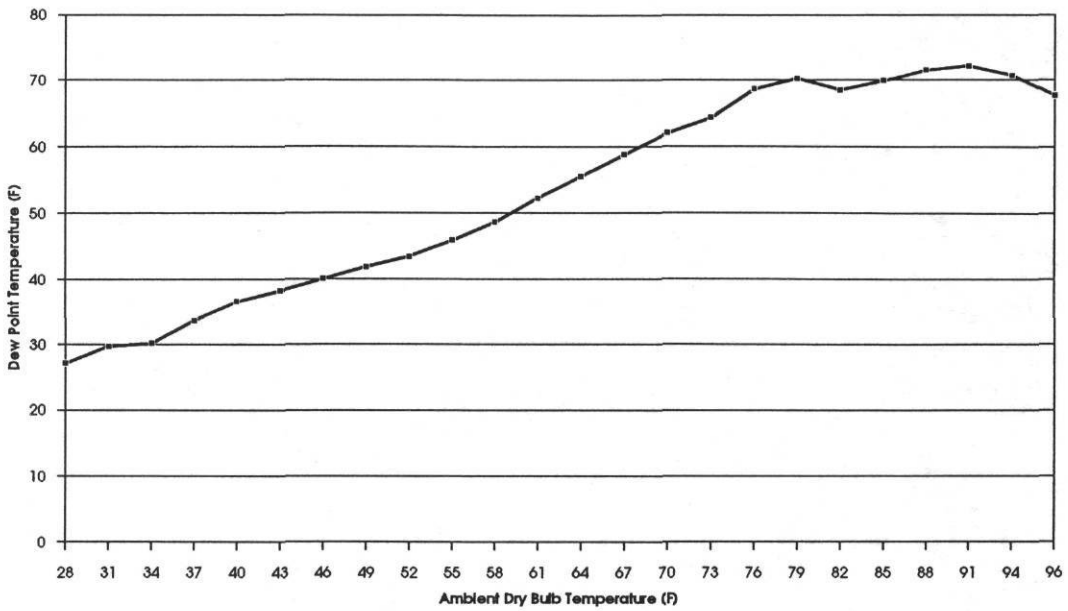

Figure 8: Coincident Dew Point Temperature vs. Ambient Dry Bulb Temperature for Houston from July 1, 1992 to June 30, 1993.

The comparison of measured and predicted annual energy consumption is given in Table 2. It shows that the calibrated model has a high level of accuracy in predicting annual energy consumption.

Table 2: Comparison of Measured and Simulated Energy Consumption (December 1992 - June 1993)

Item	Chilled Water	Steam
Measured Consumption	27,700 MMBtu	20,300 MMBtu
Simulated Consumption	25,300 MMBtu	19,850 MMBtu
Difference	2,400 MMBtu	450 MMBtu

Table 3 summarizes values of key parameters used in the calibrated simplified model and in the baseline setting of the EMC system. The fraction of return air is slightly different between the calibrated model and the value calculated from design

parameters. This value is adjusted to match chilled water consumption sensitivity to ambient temperature.

Table 3: Summary of the Model Calibration Parameter Adjustments.

Item	Schedule (EMCS)	Schedule (Model)
Air flow (cfm)	175,000 (Blue prints)	172,000
Outside air fraction	0.8 (Blue prints)	0.75
Main-cold deck temp. (^{o}F)	54	54

Hot deck (^{o}F)	If T0<80 then
	Min(90, 80-0.25*(T0-75))
	Else
	80
		If T0<80 then
		Min(90, 80-0.25*(T0-75))
		Else
		80

Return air temperature (^{o}F)	77	77
Room air temperature (^{o}F)	72	72

4. OPTIMIZING COLD DECK SCHEDULE

The goal of optimizing the cold deck by raising its temperature is to minimize the energy consumption while maintaining the require comfort levels and also avoiding costly retrofit measures. In order to maintain indoor comfort levels, the following conditions should be satisfied: 1) the cold deck supply temperature should not be greater than 61 ^{o}F during cold winter days and should be low enough to maintain room comfort during hot summer days; 2) the hot deck supply temperature should not be lower than 80 ^{o}F during hot summer days; and 3) the room relative humidity should be within the range of 25% to 60%. In order to avoid retrofit costs, the following constraints are imposed: 1) no reduction in air flow is allowed; 2) air flow rate through hot and cold ducts should not exceed design limits; and 3) no frequent manual operations should be involved.

The optimization process is currently an iteration process. A best operation schedule is first chosen based on prior experience of the O&M staff knowledge. Then,

energy (chilled water and steam) and the mechanical operation performance (air flow through cold and hot ducts) are predicted using the simplified model. These are then compared with the best operation schedule known so far. Modifications are subsequently made and a new simulation is performed. This process is repeated until the operation schedule is considered optimal.

Table 4 lists the base and the optimized operation schedules. The base and the optimized schedules are also shown in Figure 9.

Table 4: Comparison of Operation Schedules

Item	Base	Optimized

Cold deck	54 °F	If T0>58 then
		Min(61, 61-0.09*(T0-58))
		Else
		61

Hot deck	If T0<80 then
	Min(90, 80-0.25*(T0-75))
	Else
	80
		If T0<80 then
		Min(90, 80-0.25*(T0-75))
		Else
		80

Figure 9 shows the base and optimized schedule for cold deck air temperature. Obviously, this cold deck temperature increase can reduce chilled water and steam consumption substantially.

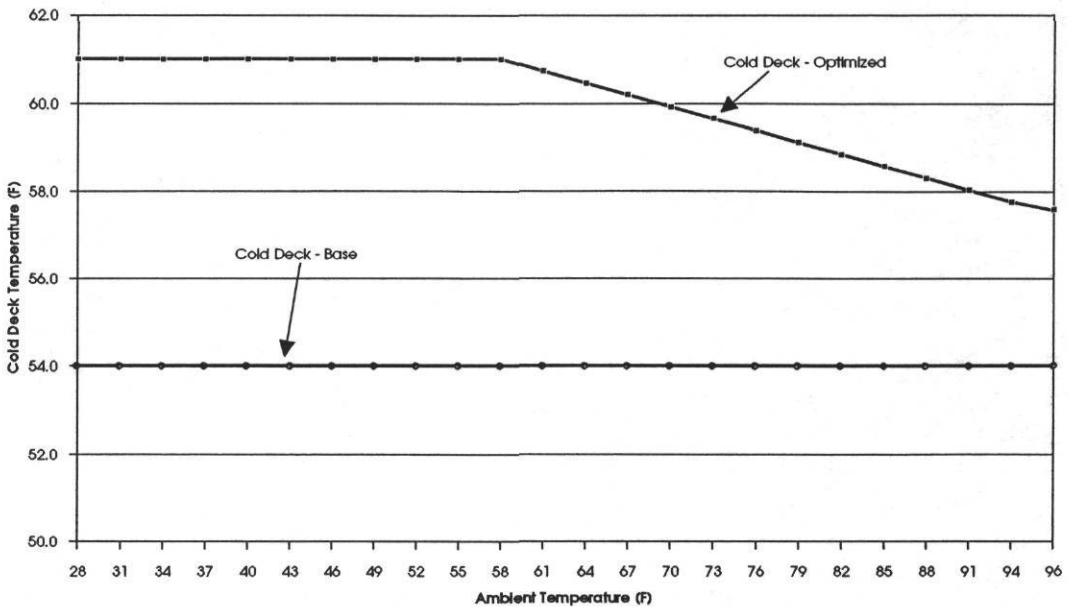

Figure 9: Base and Optimized Cold Deck Schedule

The optimized schedule changes cold deck smoothly with ambient temperature. This change can be performed by the EMCS without any major changes or difficulty.

The energy and the mechanical performance under an optimized operation schedule are compared with the base performance in the next section.

5. SIMULATED RESULTS AND DISCUSSIONS

5.1 Thermal Energy Savings Potential:

The calibrated simplified model has been used to calculate the chilled water consumption, steam consumption, room relative humidity, and air flow rate through cold and hot ducts at each bin temperature and its coincident dew point for both the base and optimized schedules. The annual energy consumption is calculated by summing the products of energy consumption and number of hours at each bin temperature over all bin temperatures.

Figure 10 compares the optimized energy performance with the base energy performance. The horizontal axis is the ambient bin temperature. The vertical axes are the energy consumption for chilled water and steam in MMBtu/hr. It shows that the optimized schedule can reduce chilled water consumption by 1.9 MMBtu/hr and steam consumption by 1.2 MMBtu/hr regardless of the ambient temperature. The simultaneous reductions of the chilled water and the steam consumption indicate that the major part of the savings are due to elimination of simultaneous cooling & heating. The relative larger chilled water savings indicate that the optimized schedule will remove less moisture, which can cause a higher room relative humidity.

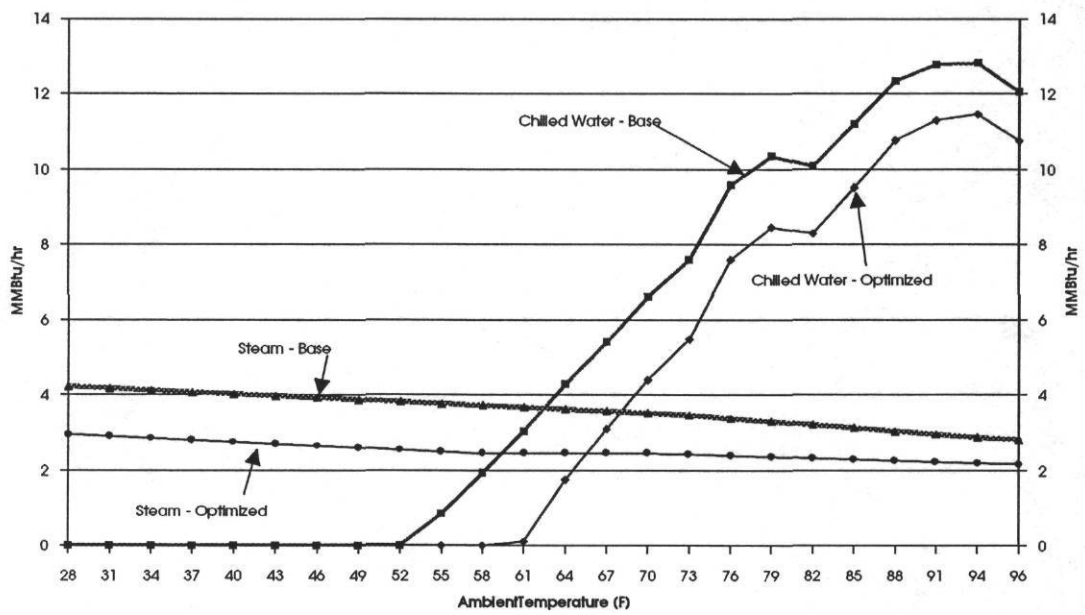

Figure 10: Comparison of the Predicted Chilled Water and Steam Energy Consumption Under Base and Optimized Operation Schedule

Figure 11 compares the predicted room relative humidity levels under the optimized and the base schedules. The predicted room relative humidity under the base schedule was consistent with the EMCS measured values. The optimized schedule can increase the room relative humidity to 61%, which is about 15% higher than the base schedule value. Recent studies [4] have found that room relative humidity levels have

less impact on comfort levels than it was thought and there is now a tendency to enlarge the relative humidity comfort zone from 25% ~ 60% to 25% ~ 70%.

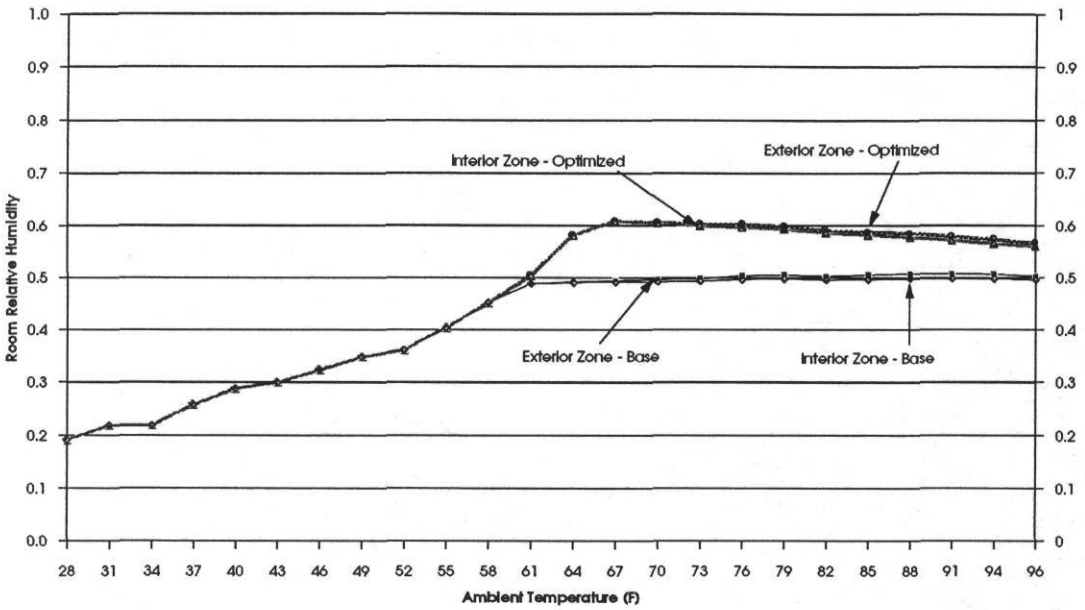

Figure 11: Comparison of the Predicted Room Relative Humidity under Base and Optimized Operation Schedules

Figure 12 compares the predicted air flow rates through cold and hot air ducts under both the base and the optimized schedules. The base schedule has a cold air flow range of 91,600 cfm to 113,500 cfm and a hot air flow range of 76,300 to 80,500 cfm, while the optimized schedule has a cold air flow range of 115,200 cfm to 131,600 cfm and a hot air flow rate range of 40,400 cfm to 56,800 cfm. The optimized schedule result in a relatively larger flow range than the base schedule. However, this flow range increase can be accommodated by the existing system, which has a capacity of 120,000 cfm for cold air and of 90,000 cfm for hot air.

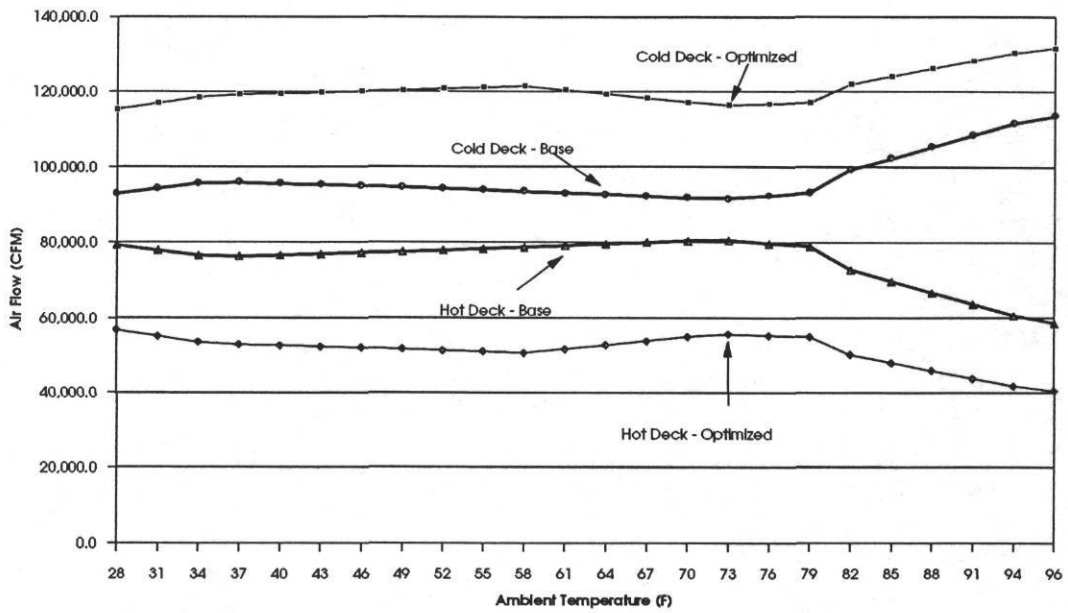

Figure 12: Comparison of Air Flow Rates through the Cold Deck and the Hot Deck under Base and Optimized Schedules

The annual energy consumption has been predicted for both the base schedule and the optimized schedule and are compared in Figure 13. The horizontal axis is the ambient bin temperature and the vertical axis is the annual energy consumption for each bin year. The potential chilled water savings can be calculated as the areas enclosed by two chilled water consumption curves, and the potential steam savings can be calculated as the area enclosed by two steam curves.

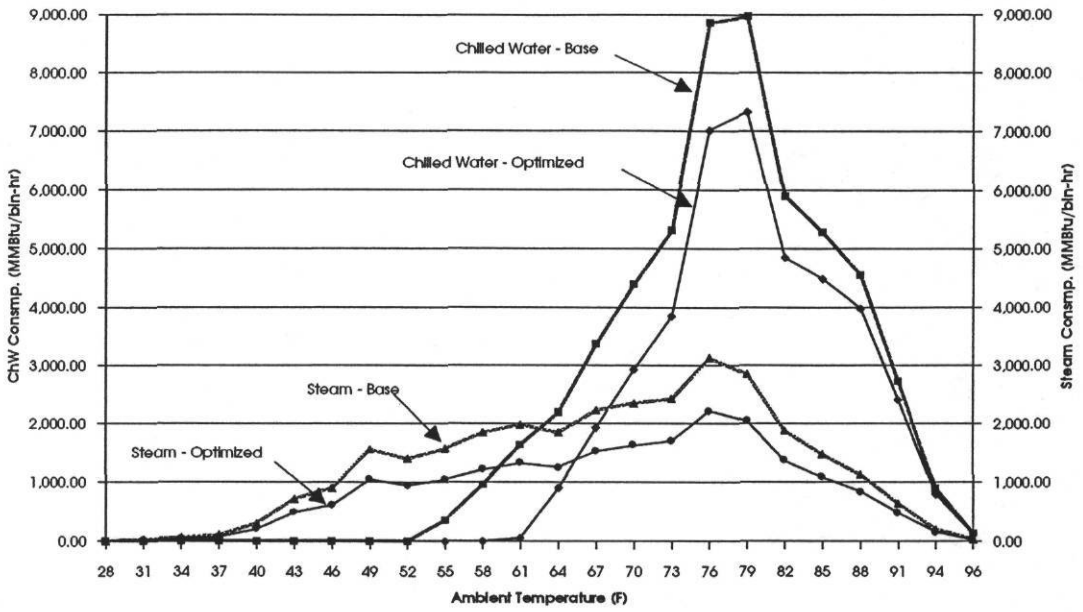

Figure 13: Comparison of the Predicted Annual Chilled Water and the Steam Energy Consumption under Base and Optimized Operation Schedules

The overall energy performance and the potential savings are summarized in Table 5. It shows that the optimized schedule can reduce annual chilled water consumption from 55,500 MMBtu to 40,600 MMBtu, with a savings of 14,900 MMBtu/yr and reduce the annual steam energy consumption from 30,600 MMBtu to 21,200 MMBtu with a savings of 9,400 MMBtu/yr. These energy savings reduce the annual cost by $108,700 for chilled water and $47,300 for steam. The total potential savings are $156,000/yr, which are 23% of the current annual building energy cost, or 27% of the current thermal energy costs.

Table 5: Summary of Potential O&M Savings at Basic Science Building

Description	Consumption		Savings					
	MMBtu		MMBtu		Dollars		Total	
	Ch-Water	Steam	Ch-Water	Steam	Ch-Water	Steam	Dollars	%
Base	55,000	30,600						
Optimized Case	40,600	21,200	14,900	9,400	108,700	47,300	156,000	23%

Note:

* The annual energy costs were $670,864, which includes $96,987 for electricity costs (1992, Basic Science Building, LoanSTAR measured energy consumption data), $405,069 for chilled water costs, and $168,808 for steam. The chilled water and steam consumption were calculated using a simplified model which was calibrated using the measured chilled water and steam consumption.

* The energy costs were calculated according to the following unit energy prices: $0.02679/kWh for electricity, $7.30/MMBtu for chilled water and $5.055/MMBtu for steam.

Table 6 summarizes the energy indices of Basic Science Building based on gross floor area (137,856 ft^2). The optimized schedule can reduce annual chilled water consumption per unit floor area from 0.4 to 0.3 MMBtu/ft^2-yr, reduce steam energy index from 0.22 to 0.15 MMBtu/ft^2-yr. The potential chilled water and steam combination savings are \$1.13/ft^2-yr.

Table 6: Summary of Thermal Energy Indices

Item	Chilled Water	Steam	Savings (MMBtu/ft^2)		Total Savings
	MMBtu/ft^2	MMBtu/ft^2	Chilled water	Steam	(\$/ft^2)
Base	0.4	0.22			
Optimized	0.3	0.15	0.79	0.34	1.13

6. MEASURED SAVINGS

Cold deck temperature for both the air handling units was raised from 54 ^{o}F to 59 ^{o}F on July 2, 1993. Reduction in chilled water and steam consumption was immediately noticed. Data from July 2, 1993 to October 25, 1993 were used to calculate the savings for 117 days by using a single linear regression model. Figure 14 shows the pre-and the post-chilled water consumption and Figure 15 shows the pre-and post-steam consumption. The drop in energy consumption is distinctly much noticeable. As of October 25, 1993 Basic Science Building has saved 5,840 MMBtu in chilled water energy and 3,100 MMBtu in steam energy, which translates into \$42,600 and \$15,600, respectively. The total savings in 117 days comes out to be \$58,200.

The other major concern was the rise in room relative humidity. In the previous section it has been stated that the optimized schedule can increase the room relative humidity to 61%. According the EMCS data for July 16, 1993 the room relative humidity was 63% which is consistent with the predicted value.

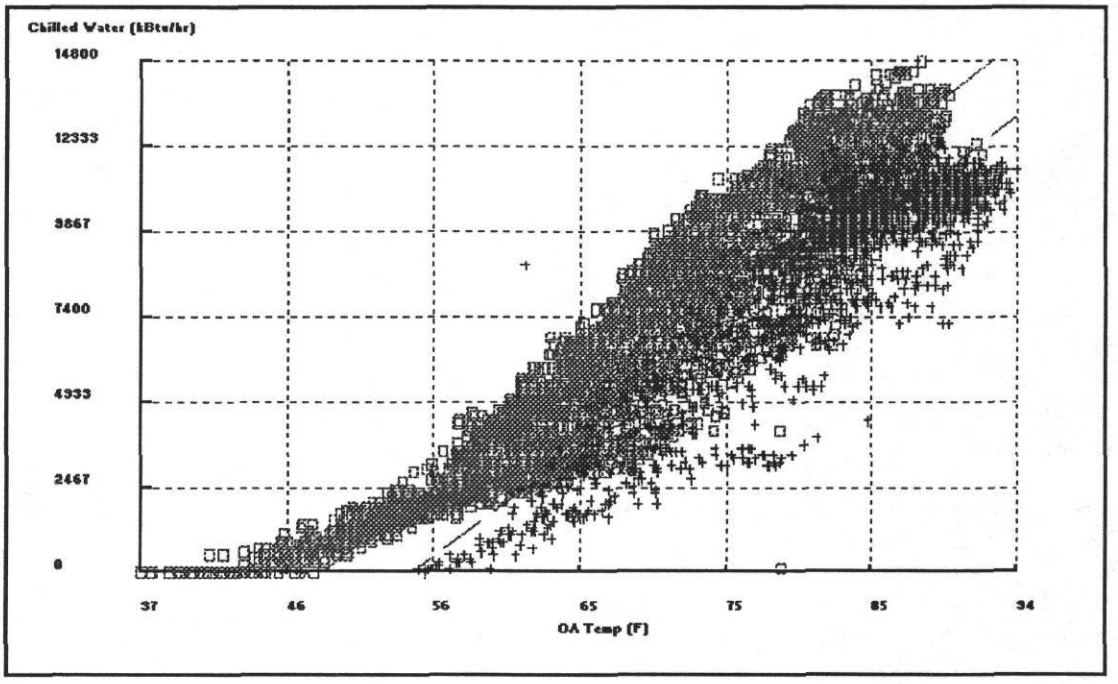

Figure 14: Pre-and Post-Chilled Water Consumption (January 1993 to October 1993) after raising Cold Deck Temperature to 59 ^{O}F on July 2, 1993.

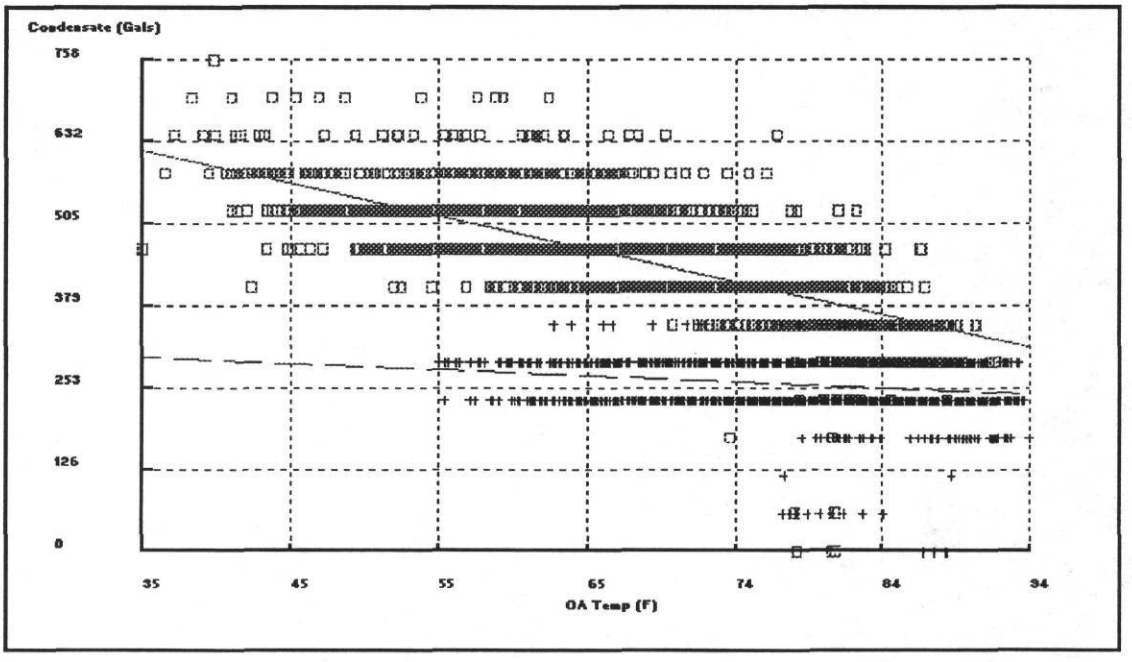

Figure 15: Pre-and Post-Steam Consumption (January 1993 to October 1993) after raising Cold Deck Temperature to 59 ^{O}F on July 2, 1993.

7. CONCLUSIONS

Our study finds that the annual building energy costs can be reduced by $156,000. The optimized operation schedules, developed by minimizing thermal energy consumption in the building, can be implemented by changing the EMCS program. The optimized operation schedule does not degrade the room comfort levels.

Other energy conservation measure, such as partially closed chilled water coils were also considered. The total savings from O&M option # 2 comes out to be $176,000/yr. However, this retrofit measure would require investment costs and exceeded the current scope of this study.

REFERENCES

1. D. E. Claridge, "Monthly Energy Consumption Reports" Energy Systems Laboratory, 1992 and 1993, College Station, TX.

2. ASHRAE, "1993 ASHRAE Handbook, Fundamentals," 1993.

3. S. Katipamula and D. E. Claridge, "Use of Simplified System Models to measure Retrofit Energy Savings," Proceedings of the Solar Engineering 1992, Maui, Hawaii, April, 1992.

4. B. W. Olesen, "Standards for Design and Evaluation of Thermal Indoor Environment," ASHRAE Journal, August, 1993, Vol. 35, No. 8.

5. M. Liu and A. Athar, D. E. Claridge, J. Haberl, T. A. Reddy, E. White, 1994, "Potential Energy Savings from Partially Closed Cold Deck Coils & Optimal Use of Economizer Cycles in Hot & Humid Climates---Case Study", Draft Paper Submitted to the Ninth Symposium on Improving Building System in Hot and Humid Climates, Fort Worth, TX.

ACKNOWLEDGMENTS

Contributions from the Physical Plant Department at UTMB are appreciated. The EMCS measured energy data that was provided and mutual discussions of the optimized operation schedules were invaluable to this study. In particular, we would like to thank Mr. Ed White, the Executive Director of the Physical Plant Department (now the Director of Energy Conservation and Management Department), Mr. John Windham, the HVAC Foreman and Mr. Steve Hodgson. We would also like to thank Ms. Susan Swanson for her editing assistance.

APPENDIX A: SIMPLIFIED SYSTEM MODELS

The simplified schematic of air handling unit (AHU) is shown in Figure A1. The building is idealized as two zones: interior zone and exterior zone.

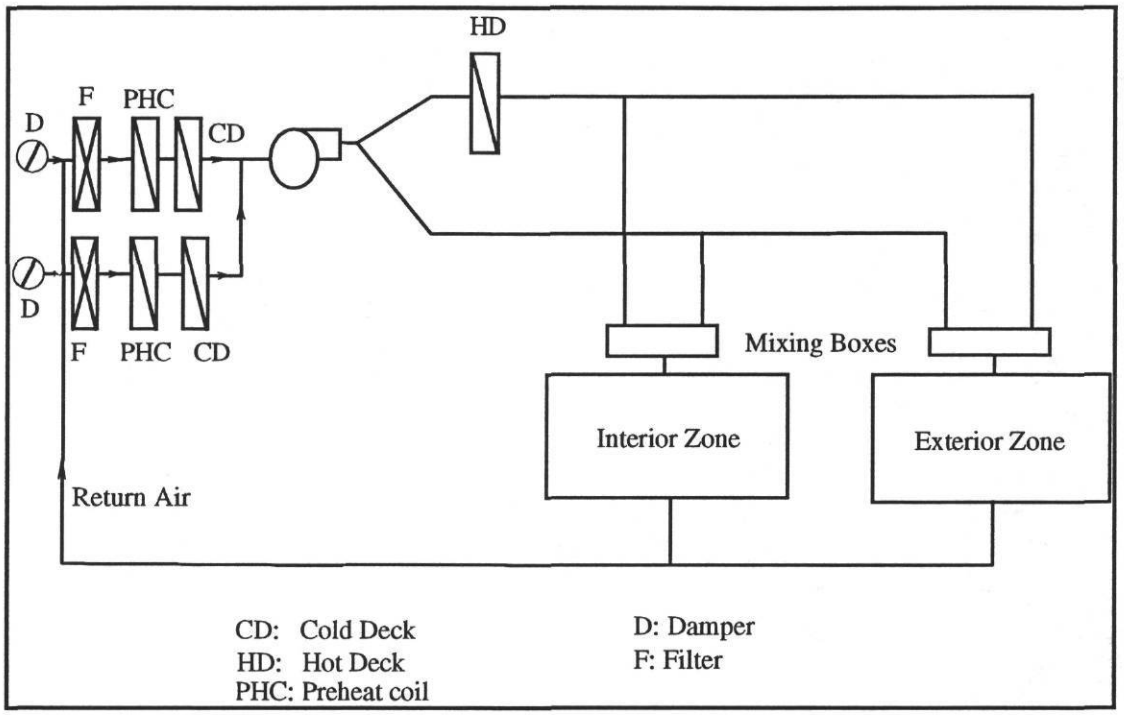

Figure A1: Schematic of HVAC System for Basic Science Building

The chilled water consumption of the main cold deck is calculated by the formula:

$$E_c = \dot{m}_c (h_m - h_c)$$

where, E_c is the chilled water energy consumption of the main cold deck, $\dot{m}_c$ is the mass flow rate through the cold deck, h_m is the specific air enthalpy at the entrance of the cold deck and h_c is the cold deck supply air specific enthalpy.

The steam energy consumption of the hot deck is calculated by the formula:

$$E_h = \dot{m}_h \times C_p (T_m - T_h)$$

where, E_h is the steam energy consumption of hot deck, $\dot{m}_h$ is the mass air flow rate through the hot deck, T_m is the air temperature at the entrance of the hot deck, T_h is the hot deck supply air temperature and C_p is the air specific heat.

The air specific enthalpy and temperature at the entrance of the cold deck and hot deck are calculated using energy balance principles.

$$h_m = f_o \times h_{pre} + (1 - f_o) \times h_r + \frac{E_{fan}}{\dot{m}}$$

where, h_r is the air specific enthalpy after the return air fan, E_{fan} is the energy consumption of the supply air fan and other symbols are as defined earlier.

The air temperature at the entrance of the cold deck and hot deck is also calculated using energy balance principles.

$$T_m = f_o \times T_{pre} + (1 - f_o) \times T_r + \frac{E_{fan}}{\dot{m} \times C_p}$$

where, T_{pre} is the pre-treatment cold deck supply air temperature, T_r is the return air temperature after the return fan, and other symbols are defined earlier.

The constant air flow terminal boxes are used in this building, therefore, the air flow rate through each box should not be changed. Consequently, the simplified model requires constant air flow rate to each zone although the ratio of cold air to the hot air changes with zone load and ambient, ambient condition and the cold deck and hot deck settings. The air flow rate to each zone is calculated according to the zone area.

$$\dot{m}_{ext} = \dot{m} \times \frac{A_{ext}}{A}$$

$$\dot{m}_{\text{int}} = \dot{m} \times \frac{A_{\text{int}}}{A}$$

where, $\dot{m}_{ext}$ and $\dot{m}_{\text{int}}$ are the air flow rate to exterior and interior zones respectively, A_{ext} and A_{int} are the conditioned floor areas in exterior and interior zones respectively and A is the total conditioned area.

The air flow through cold deck and hot deck can be solved through the following energy and mass balance equations:

$$\dot{m}_{c,\text{int}} \times (T_{room} - T_c) + \dot{m}_{h,\text{int}} \times (T_{room} - T_h) + \dot{m}_{\text{inf},\text{int}} \times (T_{room} - T_o) = \frac{Q_{\text{int}}}{C_p}$$

$$\dot{m}_{c,ext} \times (T_{room} - T_c) + \dot{m}_{h,ext} \times (T_{room} - T_h) + \dot{m}_{\inf,ext} \times (T_{room} - T_o) = \frac{Q_{ext}}{C_p}$$

$$\dot{m}_c = \dot{m}_{c,\text{int}} + \dot{m}_{c,ext}$$
$$\dot{m}_h = \dot{m}_{h,\text{int}} + \dot{m}_{h,ext}$$
$$\dot{m}_{ext} = \dot{m}_{c,ext} + \dot{m}_{h,ext}$$
$$\dot{m}_{\text{int}} = \dot{m}_{c,\text{int}} + \dot{m}_{h,\text{int}}$$

where, T_{room} is the room temperature, Q_{int} and Q_{ext} are the sensible loads at the interior zone and exterior zone respectively, $\dot{m}_{c,\text{int}}$ and $\dot{m}_{c,\text{ext}}$ are the cold deck air supply to the interior and exterior zones respectively, $\dot{m}_{h,\text{int}}$ and $\dot{m}_{h,\text{ext}}$ are the hot deck air supply to the interior and exterior zones respectively, $\dot{m}_c$ and $\dot{m}_h$ are the cold deck and hot deck air flow rate.

The room air specific humidity can be calculated using the following formula:

$$\omega_{\text{int}} = \frac{W_{\text{int}} + \dot{m}_{c,\text{int}} \times \omega_c + \dot{m}_{h,\text{int}} \times \omega_h + \dot{m}_{\inf,\text{int}} \times \omega_o}{\dot{m}_{c,\text{int}} + \dot{m}_{h,\text{int}} + \dot{m}_{\inf,\text{int}}}$$
$$\omega_{ext} = \frac{W_{ext} + \dot{m}_{c,\text{int}} \times \omega_c + \dot{m}_{h,ext} \times \omega_h + \dot{m}_{\inf,ext} \times \omega_o}{\dot{m}_{c,ext} + \dot{m}_{h,ext} + \dot{m}_{\inf,ext}}$$

where ω_{int} and ω_{ext}, are the room air specific humidity at the interior and exterior zones, respectively, W_{int} and W_{ext} are the moisture productions in the interior and exterior zones, respectively, ω_c and ω_h are the specific moisture levels at the exit of the cold deck and hot deck, respectively and other symbols are as defined earlier.

APPENDIX B: DATA QUALITY CHECK

The steam energy consumption measured by LoanSTAR is also compared with EMCS measured data for 24 hours from July 15, 1993 to July 16, 1993. Figure B1 shows the comparison results. The LoanSTAR measured steam consumption is within 10% of measured data by the EMCS.

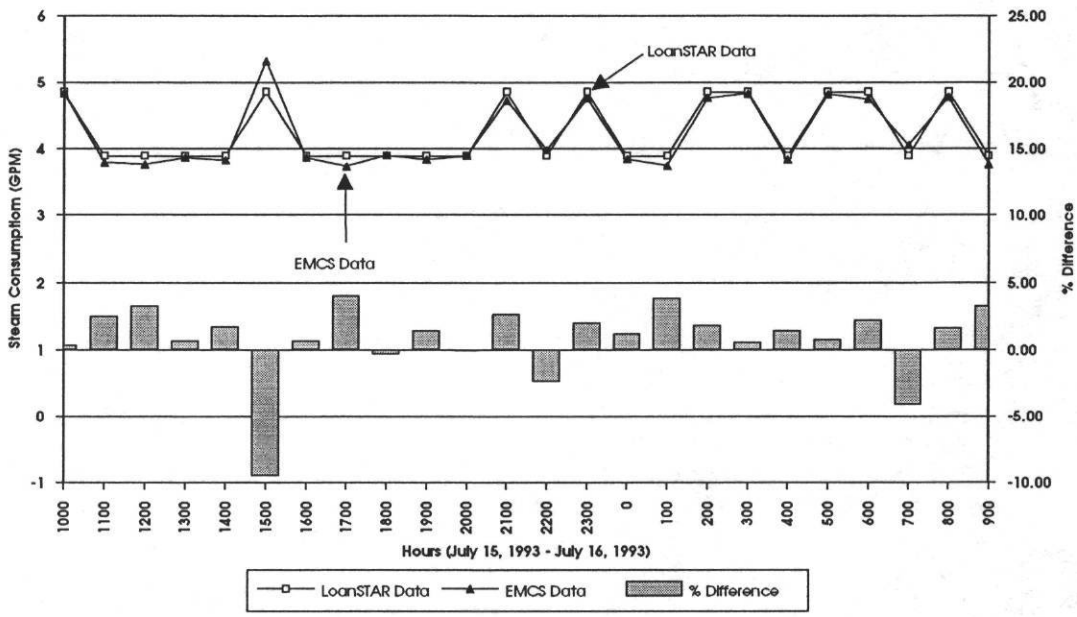

Figure B1: Comparison of LoanSTAR and EMCS measured hourly steam consumption from July 15, 1993 to July 16, 1993.

Chilled water consumption data could not be compared with EMCS measured data due to the lack of EMCS data for the same period.