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LoanSTAR Monitoring and Analysis Program

Potential Operation and Maintenance (O&M) Savings in the John Sealy South Building at UTMB

Submitted to the
Texas State Energy Conservation Office
By the
Monitoring and Analysis Task E (O&M)

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DISCLAIMER

EXECUTIVE SUMMARY

The LoanSTAR Monitoring and Analysis Group was requested by UTMB to investigate O&M measures in their five LoanSTAR program buildings. This report describes the suggested O&M measures in John Sealy South Building, an out-patient building of 373,000 ft^2, currently costs $990,000 per year on electricity, steam and chilled water. The suggested O&Ms include optimizing the HVAC operation schedule, calibrating and relocating of some of the cold deck temperature sensors. The optimized schedule was determined using an analysis involving a simplified HVAC model, which was calibrated against daily data measured by the LoanSTAR program. It is estimated that annual savings of $174,000 can be realized by changing the EMCS control program and calibrating the temperature sensors of the cold deck. The majority of energy savings occur because the optimized operation schedule reduces reheat substantially. Our analysis indicates that the indoor comfort level will not be degraded by this measure.

The lighting levels in the corridor and nurses station areas are currently substantially higher than Illuminating Engineering Society (IES) suggested values. The suggested delamping can reduce annual electricity energy costs by $46,000. These two measures can reduce the building's annual energy costs by $220,000, or 22% of the annual building energy cost.

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POTENTIAL OPERATION AND MAINTENANCE SAVINGS IN THE JOHN SEALY SOUTH BUILDING AT UTMB

1. INTRODUCTION

The John Sealy South Building is a major in-patient care facility on the campus of the University of Texas Medical Branch at Galveston. This 12-story structure consists of two wings, east and south, with a total floor area of 373,000 square feet. The building has light-colored brick walls with windows recessed from direct sunlight exposure. The windows make up 7% of the wall area. The exterior zone of the building is occupied by in-patient rooms, while the interior zone is occupied by nurses' stations and other office-type rooms.

Lighting and people are the major sources of internal gain for this building. Light energy levels are 2.75 W/ft^2 and corridors are substantially overlit. About 25% of the lights are incandescent, located in the patient room clusters, and 75% are fluorescent, located in the remaining areas of the building, such as corridors and nurses' stations. At night, most lights in patient room are turned off, while interior zone lights remain on. Lighting energy consumption can be reduced substantially by delamping and turning off lights in the interior zone (corridors, nurses' stations, and vacant special treatment rooms) at night and during unoccupied hours.

There are four dual duct constant air volume systems, which supply 302,000 CFM to the building with about 30% outdoor air intake. These systems pre-cool outdoor air before mixing with return air. Chilled water and steam are supplied by the central plant. An Energy Management and Control System (EMCS) was supposed to maintain the outdoor air cold deck temperature at 60 °F and the main cold deck temperature at 55 °F. However, these cold decks were found to be operating at 52.8 °F and 51.5 °F, respectively, due to inappropriate sensor installation and calibration problems (see Appendix A for details). This actual operation schedule is called the base schedule hereafter. Correcting sensor errors can save a substantial amount of chilled water and steam energy.

Hourly building energy consumption data (electricity, chilled water, and steam) are being measured by the LoanSTAR program [1] as well as by the EMCS at UTMB. According to the LoanSTAR measured results, this building consumed 8.7 million kWh of electricity in 1992, 87,000 MMBtu of chilled water, and 25,000 MMBtu of steam from September 1992 to August 1993. This energy consumption costs $990,000/yr or $2.65/ft^{2}yr using the following unit prices: $0.02659/kWh, $7.30/MMBtu for chilled water and $5.055/MMBtu for steam. The largest energy cost is for chilled water (64%), followed by electricity (23%), and steam (13%).

Table 1: Summary of the Annual Energy Consumption at the John Sealy North Building (September 1992 to August 1993)

	Electricity	Chilled-water	Steam	Total
Consumption	8.7 Million kWh	86,976 MMBtu	24,547 MMBtu	
Costs	$231,635	$634,923	$124,085	**$990,643**
% of Total Cost	23%	**64%**	13%	

Figure 1 shows the measured daily average chilled water and steam energy consumption versus the ambient temperature. It shows clearly that a substantial amount of steam is used on very hot summer days, which indicates that substantial reheat is present in this building. Reducing the amount of reheat is likely to save substantial steam and chilled water energy.

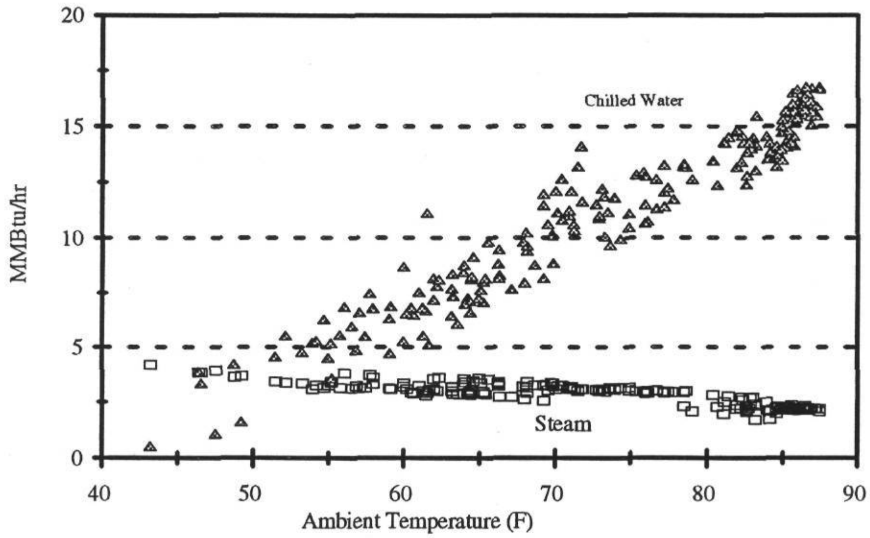

Figure 1: Measured Daily Average Chilled Water and Steam Energy Consumption vs the Daily Average Ambient Temperature. Data were measured from February 1 to August 29, 1993

All four air handling units (AHU) and their associated equipment are under EMCS control. The EMCS system can continuously regulate the hot deck and cold deck temperatures according to the ambient temperature.

This report describes a study of potential O&M improvements conducted for the John Sealy South Building at UTMB. It briefly describes the methodology used to identify O&M measures at the John Sealy South Building, presents a simplified HVAC system model used for the present O&M analysis and HVAC operation optimization, and discusses the energy and dollar savings that can be achieved.

2. METHODOLOGY

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

1. <u>LoanSTAR information data base browsing</u>. The LoanSTAR information base includes:

(i) the LoanSTAR Database (LSDB), which contains continuously measured hourly energy and weather data;

(ii) the site description note book (SDN), which contains detailed information on HVAC systems, lighting, building envelope, and occupancy schedule as well as the audit report information;

(iii) the Inspection Plot Notebook (IPN), which contains many time series and scale plots of all monitored channels for each week;

(iv) the Monthly Energy Consumption Report (MECR), which reports energy performance each month and summarizes energy performance history; and

(v) the Annual Energy Consumption Report (AECR), which summarizes one year of energy performance.

Browsing this information base led us to identify the following O&M measures (a) lighting levels could be reduced, (b) the HVAC system operation could be optimized by reducing reheat and (c) the air flow rates or CFM could be reduced.

2. Site visit/system examination. The purpose of the site visit includes:

(i) contacting personnel at the site agency and exchanging opinions on O&M potentials;

(ii) verifying information from the LoanSTAR information base by walking through the building and mechanical rooms and talking with the operator and office personnel;

3) examining the feasibility of potential O&M measures;

4) exploring new O&M measures; and

5) collecting system information, such as cold deck and hot deck temperature schedule, air flow rates, and possible nighttime setback, as well as

miscellaneous information from EMCS system, such as measured energy performance.

UTMB personnel accepted the suggestions of delamping and optimizing HVAC system while rejected reducing air flow rate because they wary that the occupants may not accept the reduction of air flow rate.

3. Data quality check. Before using the LoanSTAR data to estimate potential O&M savings, they are compared with EMCS measured data. If the two sets of data are fairly consistent, the LoanSTAR data will be used in the analysis without correction. If the LoanSTAR measured data and EMCS measured data are seemed unacceptably different, the LoanSTAR data will be checked using other methods. This data 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 and programmed into a computer code. The simplified computer model uses measured daily average ambient temperature and dew point to predict daily average chilled water and hot water energy consumption. Finally, the predicted energy consumption is compared with the measured consumption. If the predicted consumption matches the measured energy consumption, then the simplified computer model and its associated parameters, such as air flow rate, cold deck and hot deck settings, and internal gains, are considered to be realistic estimates. Otherwise, calibration is required which involves adjusting parameter estimates such that better agreement with monitored data is achieved.

The preliminary model analysis showed that the EMCS's cold deck setting is higher than actual value. The measurement performed later proved that the actual cold deck settings in the four AHUs are lower than EMCS settings by 1 °F to 6 °F.

5. O&M simulation & savings calculations. The cold deck and hot deck schedules are optimized such that energy consumption is minimized 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 the ducts should be less than their capacities or design values; and

(v) there should be no extra implementation cost involved.

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

6. Feedback from UTMB physical plant personnel. UTMB personnel comment on the proposed optimized schedule and provide information necessary to modify the proposed schedule if needed. The simplified model simulation might suggest 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 the potential savings recalculated.

8. Short-term test of optimized schedule and implementation. The fixed temperature settings for the cold deck and hot deck are derived from the optimized schedule under certain ambient temperature conditions. UTMB personnel disable the EMCS system temporarily and use the suggested setting instead for a few days. Although this test would not show the full potential of optimized schedule savings, it provides the opportunities to

expose 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 & ITS CALIBRATION

3.1. Simplified Model and Input Data

The schematic of air handling units (AHU) and the building is shown in Figure 2, where the four AHUs are treated as one AHU and the building is idealized as two zones: an interior zone and an exterior zone. This modification is consistent with previous studies, for example, that of Katipamula and Claridge [2].

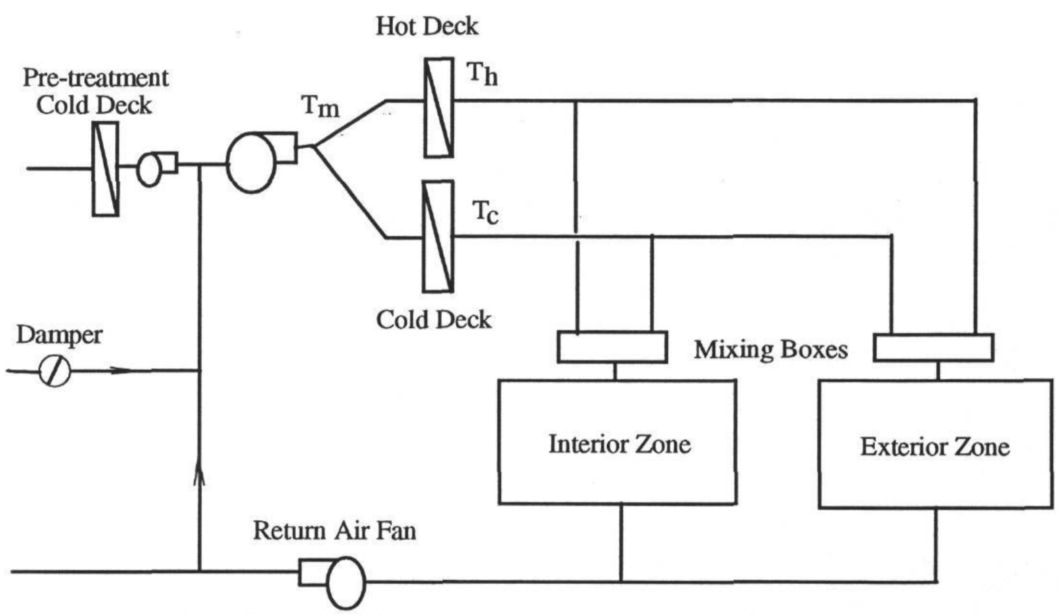

Figure 2: Schematic of HVAC System for John Sealy South Building

This is a dual-duct constant volume system. Currently, the system gets outdoor directly without through O. A. cold deck when the ambient temperature is lower than 65 °F. If the ambient temperature is higher than 65 °F, the outdoor air is forced through pre-treatment cold deck and be cooled down to 60 °F. The main cold deck sends 55 °F air to the mixing box, where it mixes with hot air to maintain room temperature at suitable levels. Although the air flow rates of cold duct and hot duct may change, the air flow to each zone and the total air flow keep constant.

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

The four AHUs supply 302,000 CFM air flow to the building with a total outdoor air intake of 69,000 CFM according to the initial design. The outdoor air cold deck was set at 60 °F, the main cold deck supply air temperature at 55 °F and 56 °F at daytime and nighttime, respectively, the hot deck supply air temperature at 80 °F to 90 °F, which was varied with ambient temperature. EMCS measured results show that the building has an average room temperature of about 72 °F and return air temperature of 77 °F after return fans.

The exterior zone is taken as the sum of areas which are directly connected with the exterior envelopes according to the building plan (Figure 3). 35% of the total area is classified as the interior zone and the rest of the area as the exterior zone. The total conditioned floor area is taken as 80% of the gross floor area (373,085 ft^2). The internal gain is taken as 2.75 W/ft^2 based on the measured lighting capacity, while a factor of 0.8 is used to account for gain reduction at night. The number of people is estimated by assuming one person for every 120 ft^2 of conditioned area, and the sensible and latent loads due to people are calculated by assuming standard losses by normal office workers [3]. The domestical hot water and other steam and hot water consumption are estimated as 1.2 MMBtu/hr, which was determined as the measured steam consumption when all the hot decks were shut down.

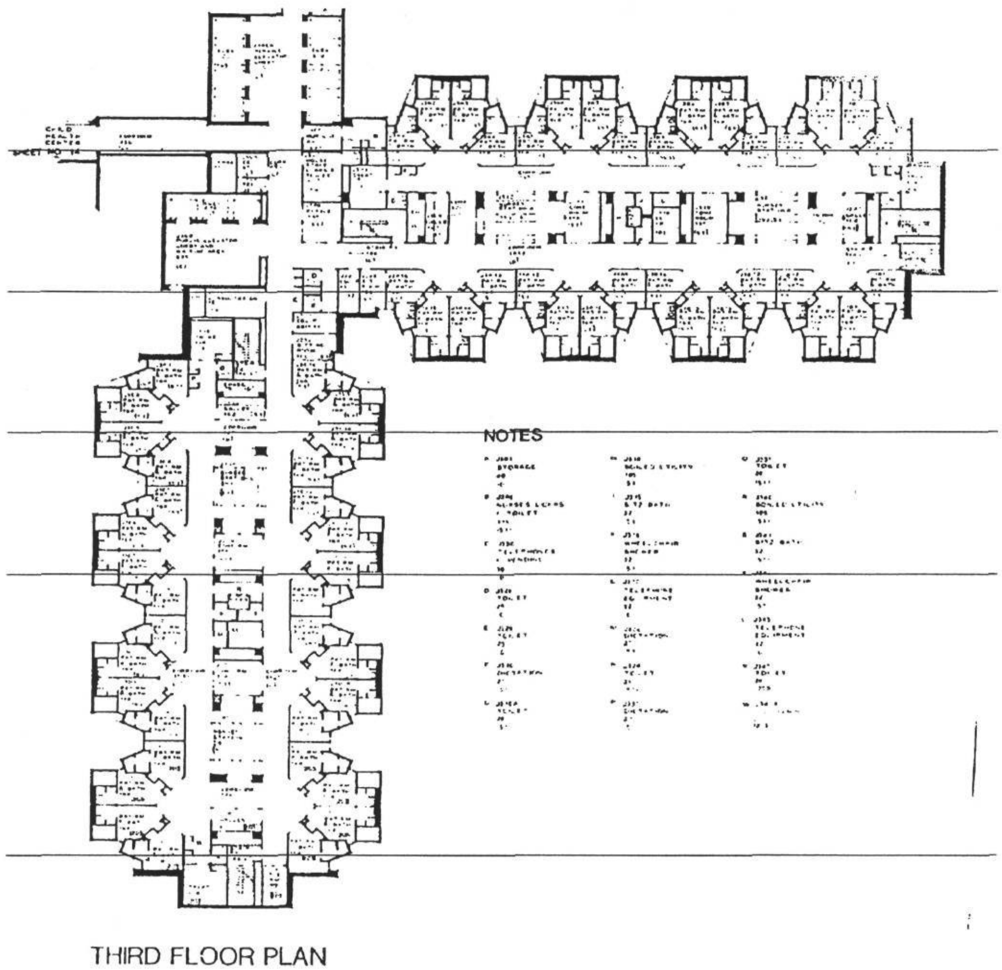

Figure 3: Typical Floor Plan in the John Sealy South Building

The building envelope area is calculated as 243,584 ft^2, which includes 15,967 ft^2 window area. A heat transfer coefficient value of 0.2 Btu/ft^2 °F hr was assumed for walls, 1 Btu/ft^2 °F for windows.

Air infiltration rates are 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 door and corridors.

3.2. Model Calibration

LoanSTAR measured chilled water and steam data are compared with EMCS measured energy data on a monthly basis for a year and on a daily basis for a month. The comparisons show that the LoanSTAR measured steam consumption is about 91% of the EMCS measured values while the LoanSTAR measured chilled water consumption is 9% higher than the EMCS measured consumption. The difference between the LoanSTAR and the EMCS measured steam Btu consumption is due to the difference in return water enthalpy used. The difference between the LoanSTAR and the EMCS measured chilled water consumption is likely due to meter inaccuracies. Appendix B contains further details.

The chilled water and steam energy consumption were predicted with the simplified model using the measured daily average temperature from February 1993 to August 1993. However, the model predictions were far less measured chilled water and steam consumption. The O&M staff speculated that the cold deck temperatures were not correct. This issue was discussed during the feedback meeting and measurements were performed after the meeting. Measurement showed a main cold deck supply air temperature of 51.5 °F and a outdoor air treatment cold deck temperature of 52.8 °F , while EMCS readings were 53.9 °F and 56 °F respectively. More details are given in Appendix A. After these new cold deck settings were used in the simplified model, the predicted daily average chilled water consumption was 3% less than the measured value while the predicted steam consumption was 0.3% lower than the measured value for the period from February 1, 1993 to July 12, 1993. The standard root mean square errors of the predictions are 1.12 MMBtu/hr and 0.3 MMBtu/hr for chilled water and steam, respectively. The coefficients of variation are 0.11 and 0.10 for chilled water and steam, respectively.

Figures 4 and 5 permits a visual comparison of the measured energy consumption and the model predicted energy consumption. Figure 4 is a scatter plot shown consumption versus temperature while Figure 5 shows consumption as time series data.

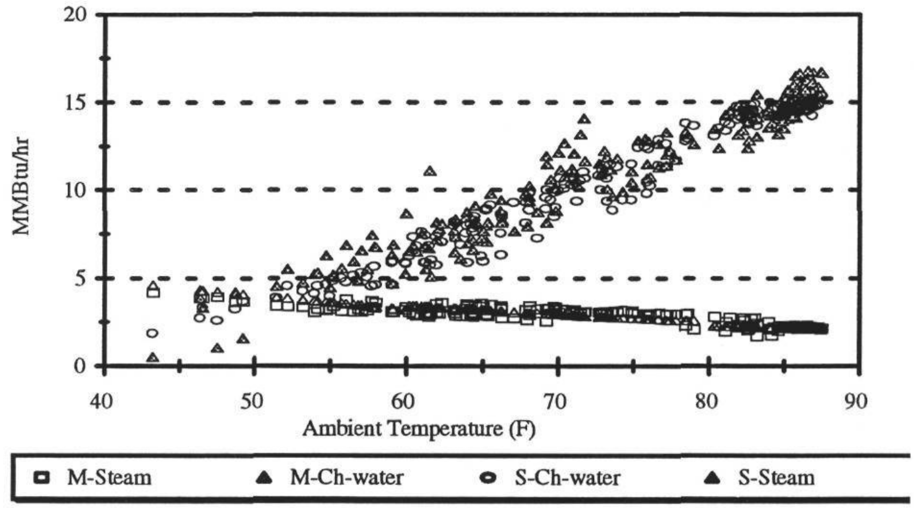

Figure 4: Comparison of Daily Average Energy Consumption Between the Model Predicted and Measured. Data were measured from February 1, 1993 to August 29, 1993. (M-steam : measured steam consumption; M-Ch-water : measured chilled water consumption; S-Ch-water : simulated chilled water energy consumption; and S-steam : simulated steam consumption)

Figure 5 shows that the model predicted chilled water consumption is lower than measured values from the middle of July to the middle of August. This difference may be due to a decreased cold deck setting during this hot period according to the operating staff.

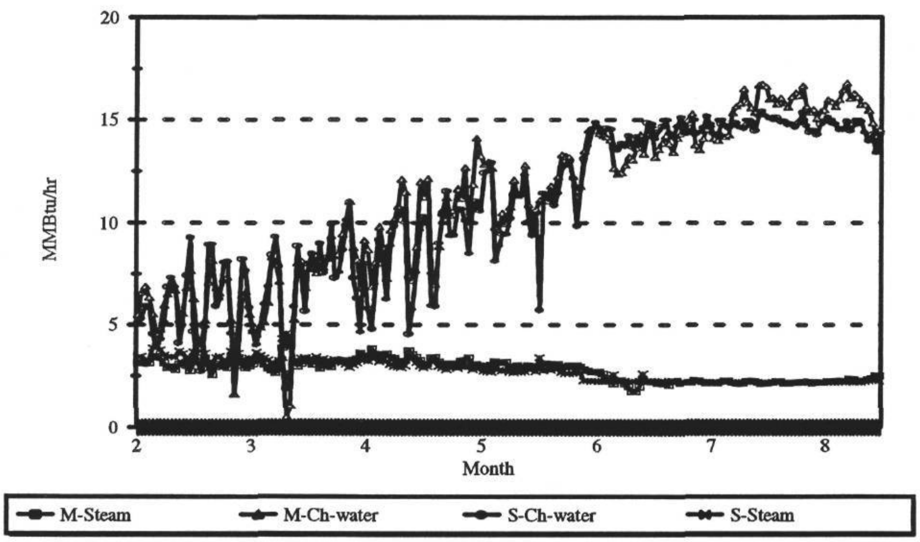

Figure 5: Comparisons of Daily Average Energy Consumption as time Series. Data from February 1, 1993 to August 29, 1993
(M-steam : measured steam consumption; M-Ch-water : measured chilled water consumption; S-Ch-water : simulated chilled water energy consumption; and S-steam : simulated steam consumption)

The calibrated simplified model was used to calculate annual energy consumption using bin data for outdoor temperature. Due to lack of measured hourly dry bulb and dew point temperatures in Galveston for a complete year during 1992-93, the measured hourly data from July 1, 1992 to June 30, 1993 for Houston were used to generate bin temperatures, as shown in Figure 6. The horizontal axis is the bin temperature, where 24-bin is used with 3 °F width for each bin. The vertical axis shows the number of hours during this year for each bin temperature. It was assumed that Galveston has the same weather conditions.

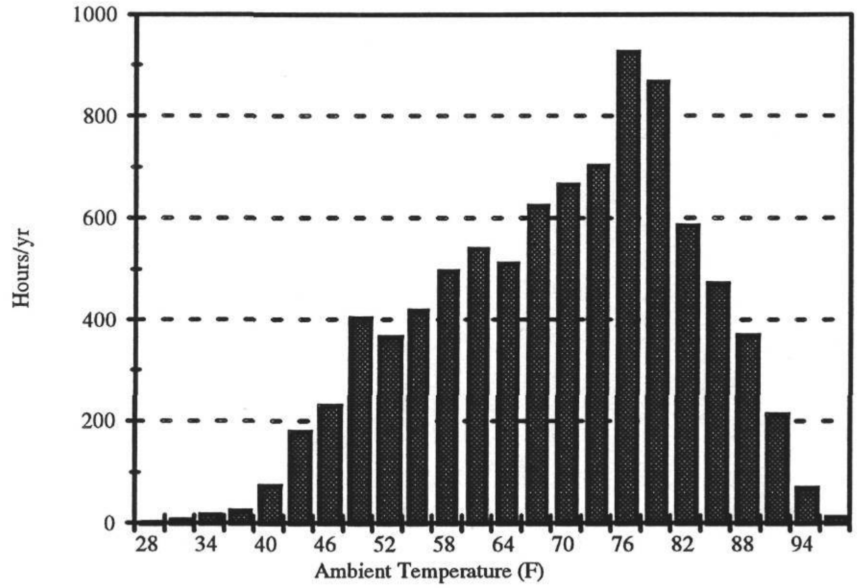

Figure 6: Houston Bin Temperature Chart Generated Using LoanSTAR Measured Hourly Temperature Data from July 1, 1992 to June 30, 1993.

The mean coincident dew point temperatures are plotted as a function of the ambient bin temperature in Figure 7. The Figure shows that the dew point increases with the ambient temperature when the ambient temperature is lower than 80 °F, and remains more or less a 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 loads do not change when the ambient temperature is higher than 80 °F.

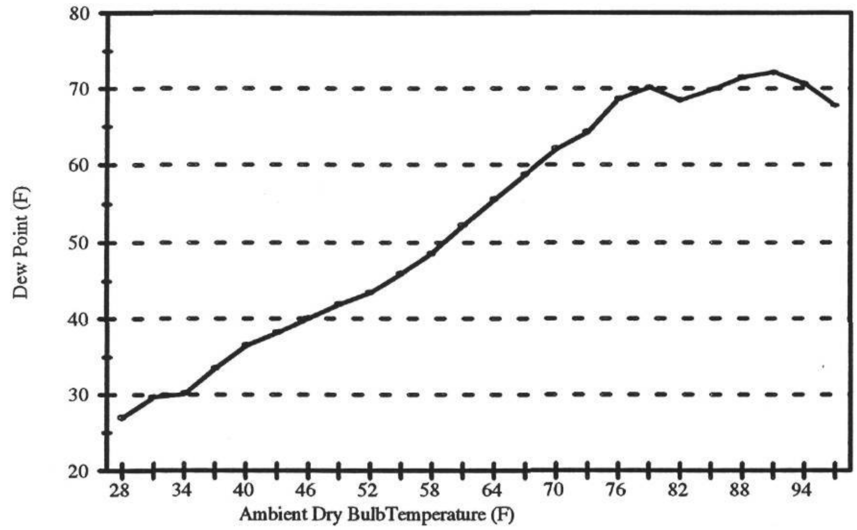

Figure 7: Mean Coincident Dew Point Temperature as a Function of Dry Bulb Temperature in Houston for July 1, 1992 through June 30, 1993

Table 2 summarizes the values of the key parameters used in the calibrated simplified model and the baseline settings of the EMCS system. The obvious difference between the EMCS measured supply air temperature and the actual measured temperature indicates that **relocating** and calibrating temperature sensors are necessary.

The calibrated fraction of return air is slightly different when the value is calculated from the design parameters. This calibrated fraction of return air makes the predicted chilled water consumption best match the measured chilled water consumption.

Table 2: Summary of the Model Calibration Parameter Adjustment

Item	Schedule (EMS)	Schedule (Model)
CFM	302,300 (Blue prints)	302,300
Return air fraction	0.77 (Blue prints)	0.7
Pre-cold deck temperature °F	60.0	52.8*
Main-cold deck temp. °F	55.0	51.5*

Hot deck °F	If T0<80 then
	Min(90, 80-0.25*(T0-75))
	Else
	80
		If T0<80 then
		Min(95, 90-0.25*(T0-75))
		Else
		85

Return air temperature °F	77	77
Room air temperature °F	72	72

4. OPTIMIZED COLD DECK & HOT DECK SCHEDULES

The goal of optimizing cold deck and hot deck schedules is to minimize the energy consumption while maintaining comfort levels and avoiding retrofit costs. In order to maintain indoor comfort levels, the following conditions should be satisfied: 1) the cold deck supply air temperature should be low enough to maintain interior zone comfort condition during cold winter days and the supply air temperature should be low enough to maintain exterior room comfort during hot summer days; 2) the hot deck supply air temperature should not be lower than 75 °F during hot summer day; 3) the room relative humidity should be within the range of 30% to 60%. In order to avoid retrofit costs, the following constraints exist: 1) no CFM reduction is allowed; 2) air flow rates 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 O&M staff knowledge. Then, energy (chilled water and steam) and mechanical operation performances (air flow through cold and hot ducts) are predicted using the simplified model. After the energy and mechanical performances are compared with the so far best known, modification of the operation schedule is made and a new simulation performed. This process is repeated until the operation schedule is considered adequate by O&M staff.

Table 3 lists the base and the optimized operation schedules. The base and the optimized schedules are also shown in Figure 8. We note that the optimized schedule has both the O. A. cold deck and the main cold deck supply air temperatures higher than those of the base schedule settings. Obviously, these cold deck temperature increases can reduce chilled water and steam consumption substantially.

Table 3: Comparison of Operation Schedules

Item	Base	Optimized

O. A. treatment coil	If T0>60 °F then
	52.8 °F,
	else
	Off
		if T0>60 °F then
		Min(54, 54-0.05*(T0-60))
		else
		off

Main cold deck	51.5 °F	Min(59, 59-0.05*(T0-50))

Hot deck	If T0<80 then
	Min(95, 90-0.25*(T0-75))
	Else
	85
		If T0<80 then
		Min(85, 85-0.25*(T0-60))
		Else
		75

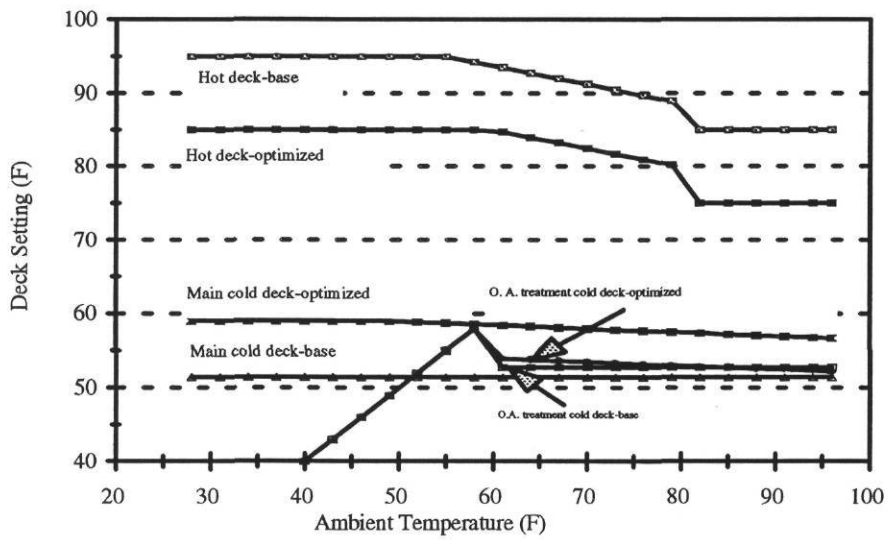

Figure 8: Base and Optimized Cold & Hot Deck Schedules

The optimized schedule changes in cold deck temperature with ambient temperature, can be performed by the EMCS without extra work. The hot deck supply air temperature is decreased compared to the base schedule, while still high enough to satisfy heating requirement.

The optimized schedule has the O. A. cold deck supply air temperature lower than the main cold deck supply air temperature. This arrangement removes one of the two duties of the main cold deck: to remove moisture and to remove sensible heat. If the O. A. cold deck can remove enough moisture, then the main cold deck supply air temperature can be regulated solely based on sensible load. Consequently, cold deck supply air temperature can be increased, which can result in substantial energy savings.

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

5. RESULTS AND DISCUSSIONS

5.1 Thermal Energy Saving 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 the energy consumption and number of hours at each bin temperature over all bin temperatures.

Figure 9 compares the optimized energy performance with the base energy performance. The horizontal axis is the ambient bin temperature. The vertical axis is the energy consumption in MMBtu/hr for both the chilled water and the steam. It shows that the optimized schedule can reduce chilled water consumption by 1.95 MMBtu/hr and steam consumption by 1.13 MMBtu/hr regardless of the ambient temperature. The simultaneous reductions of the chilled water and the steam consumption indicate that the majority of savings, which are about 1.13 MMBtu for chilled water and 1.13 MMBtu for steam, come from the reduction of reheat. The relatively larger chilled water savings (0.82 MMBtu/hr) indicate that the optimized schedule will remove less moisture, which can cause a higher room relative humidity. It was also noted that there are sudden decreases of both the chilled water and the steam consumption when the ambient temperature is around 80 °F which is due to the schedule change of the hot deck.

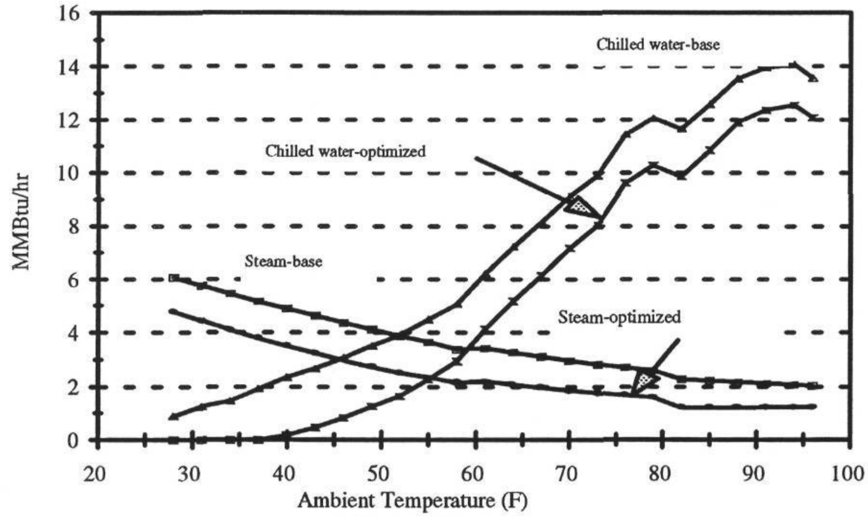

Figure 9: Comparison of the Predicted Chilled Water and the Steam Energy Consumption Under Both the Base and the Optimized Operation Schedules

Figure 10 compares the predicted room relative humidity levels under the optimized schedule and under the base schedule. 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 57%, which is about 8% higher than the base schedule value. Recent studies [4] have found that room relative humidity levels have less impact on comfort levels than was thought earlier and there is a tendency to enlarge the relative humidity comfort zone from 30% ~ 60% to 30% ~ 70%. One of the UTMB buildings has had a room relative humidity of about 60% since July, 1993 with no complaints.

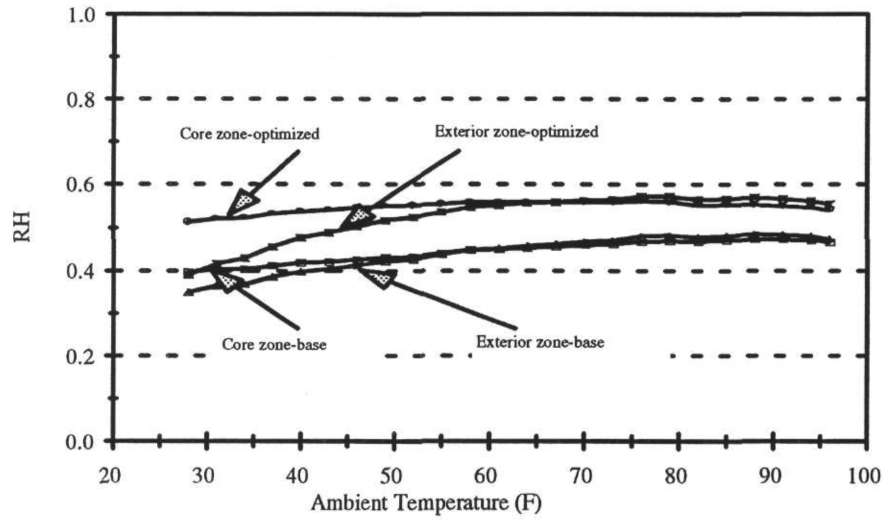

Figure 10: Comparison of the Predicted Room Relative Humidity under Both the Base and the Optimized Operation Schedules

Figure 11 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 130,000 CFM to 220,000 CFM and a hot air flow range of 75,000 to 170,000 CFM, while the optimized schedule has a cold air flow range of 110,000 CFM to 250,000 CFM and a hot air flow rate range of 60,000 to 190,000 CFM. The optimized schedule causes a relatively larger flow range compared with the base schedule. However, this flow range increase can be accommodated by the existing system, which has a capacity of 270,000 CFM for cold air and 220,000 CFM for hot air.

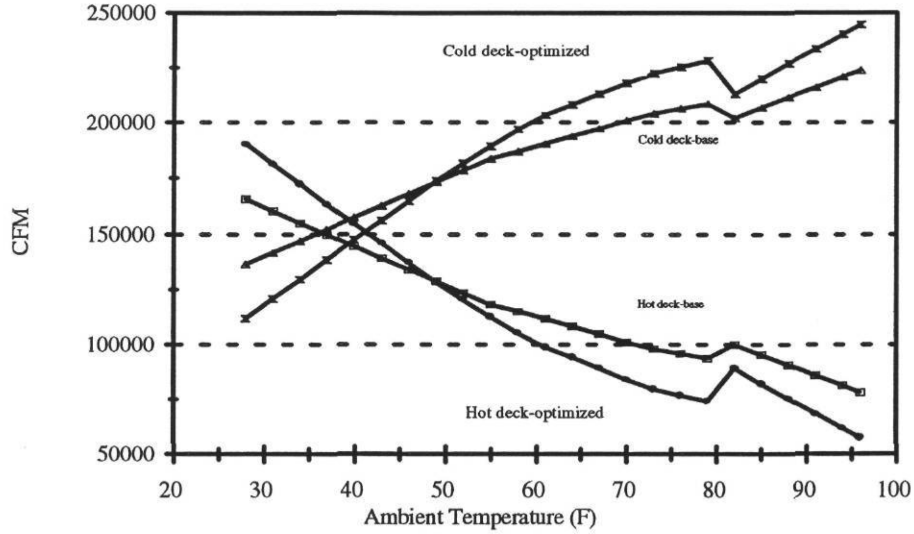

Figure 11: Comparison of Air Flow Rates Through the Cold Deck and the Hot Deck under Both the Base and the Optimized Schedules

The potential annual energy savings was calculated as the difference in energy consumption under the base and optimized schedules. The results are shown in Figure 12. The horizontal axis is the ambient bin temperature and the vertical axis is the potential annual energy savings for each bin.

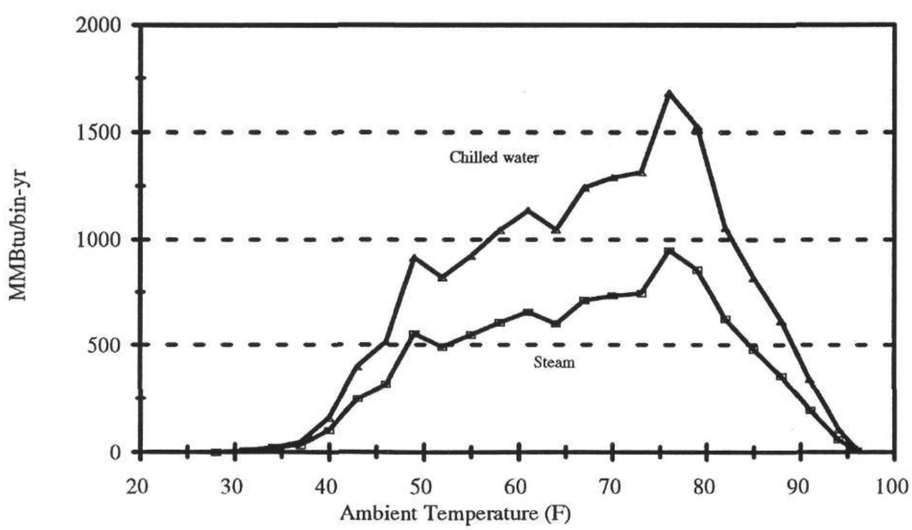

Figure 12: Predicted Potential Annual Chilled Water and the Steam Energy Savings

Table 4: Summary of Potential O&M Savings at the John Sealy North Building

No	Description	Consumption		Savings					
		MMBtu		MMBtu		Dollars		Total	
		Ch-Water	Steam	Ch-Water	Steam	Ch-Water	Steam	Dollars	%
0	Base	76,760	26,758						
1	Optimized Case	59,699	16,847	17,061	9,911	124,544	50,098	174,642	18%

Note:

The annual energy costs were $990,643, including $231,635 electricity costs (1992, John Sealy South, LoanSTAR measured energy consumption data), $634,923 chilled water costs, and $124,085 steam costs. 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 using the following unit energy prices: $0.02679/kWh for electricity, $7.30/MMBtu for chilled water and $5.055/MMBtu for steam.

The overall optimized energy performance and the potential savings are summarized in Table 4. It shows that the optimized schedule can reduce annual chilled water consumption from 76,760 MMBtu to 59,699 MMBtu with a savings of 17,061 MMBtu/yr, and reduce the annual steam energy consumption from 26,758 MMBtu to 16,847 MMBtu with a savings of 9,911 MMBtu/yr. These energy savings reduce the annual cost by $124,544 for chilled water and $50,098 for steam. The total potential savings is $174,642/yr, which is 18% of the annual building energy cost, or 23% of the chilled water and steam energy costs.

Table 5 summarizes the energy indices of the John Sealy South building based on gross floor area. The optimized schedule can reduce chilled water consumption per unit floor area from 0.206 to 0.160 MMBtu/ft^2-yr and reduce steam energy index from 0.072 to 0.045 MMBtu/ft^2-yr. The potential chilled water and steam combination savings are $0.47/ft^2-yr.

Table 5: Summary of Thermal Energy Indices

Item	Ch-water	Steam	Savings MMBtu/ft^2		Savings $
Base	0.206	0.072	Ch-water	Steam	Total
Optimized	0.160	0.045	0.046	0.027	$0.47

5.2 Nighttime Setback

It should be pointed out that the simplified model analysis did not investigate the potential savings of nighttime set back. However, it is suggested that nighttime set back be incorporated into the optimized schedule to achieve extra energy savings. This may be done by increasing the cold deck setting by 2 °F over the optimized schedule or may be done by trial and error by the operators.

The room relative humidity can be controlled to a lower level than that under the optimized schedule by partially closing the cold deck[5], which would require installation of automatic valves at each entrance of parallel cold deck coils. Other retrofit measures may also exist, such as economizer cycles and VAV conversion. However, these retrofit measures are beyond the scope of the current study.

5.3 Lighting Energy Savings Potential:

During the evening of July 15, from approximately 8:00 p.m. until 9:30 p.m., a nighttime walk-through was conducted to determine if O&M savings potential existed by turning off lights in unoccupied areas.

The O&M staff observed that lighting levels in elevator lobbies and corridors appeared to be excessive. From interviews of nurses at both the third floor and fifth floor nurses' stations, it was learned that the nurses turn off the corridor lights in some areas after 9:30 p.m., but not in all areas. The lights in patient rooms were regularly turned off at 9:30 p.m., but it was up to the individuals at the nursing stations to turn off corridor lights. Illumination levels appeared to be very high in elevator lobbies, corridors, and nursing stations.

On Friday, July 16, a more detailed lighting survey was conducted using a calibrated foot-candle meter. This survey was conducted by John Houcek of the LoanSTAR staff and George Thomas, summer intern at the UTMB-Galveston facility.

It can be seen in Table 6 that most of the areas surveyed had illumination levels in excess of IES recommendations. Since the fixtures in these areas are 2ft. × 4ft., 4 lamp, lay-in troffers, they would easily lend themselves to delamping which would decrease the electrical load by 88 watts per fixture if two lamps and a ballast are disconnected. In addition, a formal program of turning-off lights in corridors and elevator lobbies after hours would contribute to less energy consumption.

Table 6: Summary of Lighting Survey in John Sealy South Building

Floor	Area	Measured Footcandles	Recommended IES Footcandles Day	Night
3	Elevator Lobby	132	50	20
3	Day Surgery Corridor	85	30	30
5	Elevator Lobby	149	50	20
5	Main Corridor	133	20	3
5	Patient Rooms Corridor	39	20	3
5	Nurses' Station	156	70	30
7	Elevator Lobby	150	50	20
7	Main Corridor	127	20	3
7	Nurses' Station #1	150	70	30
7	Nurses' Station #2	175	70	30
7	Patient's Room Corridor	80	20	3

Based on both the nighttime walk-through and the daytime lighting survey, it is recommended that priority be given to delamping fixtures in corridors and nursing stations in the John Sealy South building. Due to time constraints we were not able to count all the fixtures on all 12 floors. However, the audit report states that a delamping project in hallways and nursing stations would involve 3554 bulbs and 1777 ballasts. The removal of 3554 bulbs, or two from each of 1777 fixtures, at 40 watts each would cause a reduction in kWh cost of $33,113/yr. based on the following formula:

3554 lamps x 40W/lamp x 8760 hr/yr x $.02659/kWh = $33,113/yr

The resultant demand reduction would allow for additional savings of $12,828/yr. based on the following formula:

3554 lamps x 40W/lamp x $7.52/kva/mo x 12 months = $12,828/yr

Total potential annual electric savings, excluding chilled water savings, is projected at $45,940/yr. Additional savings of $4,592/yr would be expected if the ballasts were also removed, but the labor factor would be much higher compared with delamping only.

6. CONCLUSIONS

Our study finds that the annual building energy costs can be reduced by $220,582 of which $174,642/yr savings is from optimizing HVAC operation schedules and $45,940/yr savings from delamping corridor and nurses' station areas. 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.

Extra savings may be achieved by incorporating nighttime set back into the optimized schedule which can be implemented by increasing the cold deck temperature by 2 °F.

Other energy conservation measures are also possible, such as the use of economizer cycles and a partially closed cold deck. However, these retrofit measures require implementation costs and are beyond the scope of the current study.

REFERENCES

1. D. E. Claridge, August, 1993, "Monthly Energy Consumption Report," Energy Systems Laboratory, Texas A&M University, College Station, TX.

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

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

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

5. M. Liu, A. Athar, D. E. Claridge, J. Haberl, T. A. Reddy, & E. White, 1994, "Potential Energy Savings from Partially Closing 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.

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APPENDIX A: COLD DECK SETTING MEASUREMENT

On September 17 (11:50 a.m. to 3:00 p.m.), 1993, Mr. Bullacher (HVAC operator) and Dr. Liu Mingsheng measured the cold deck supply air temperature, the hot deck supply air temperature, the return air temperature and the supply air temperature of pre-treat coils for four AHUs in the John Sealy Buildings using an Omega Thermometer, which is used as a calibration tool at UTMB. The measurement took about three hours. The EMCS readings were taken simultaneously. This method makes it possible to identify the possible EMCS measurement bias.

The measurement results are summarized in the following table. The first column lists the name of AHUs, the second column lists the supply air temperature of outdoor air treatment coils, the third column lists the cold deck supply air temperature, the fourth column lists the hot deck supply air temperature, and the fifth column lists the return air temperature.

Table A1. Comparison of On-site Measured and EMCS Measured temperatures

AHU		Pre-coils (°F)	Cold Deck (°F)	Hot Deck (°F)	Return (°F)
250	On-site	54	48	83	74
	EMCS	57.3	54.2	86.8	78.3
251	On-Site	59.7*	52.4	82.7	75
	EMCS	57.9	53.5	85	78
252	On-site	51.5	51.4	88.4	73.9
	EMCS	52.9	Down	Down	Down
253	On-site	72.7*	54.1	86.4	75
	EMCS	Down	Down	Down	Down

Outdoor Air (O. A.) Coil Supply Air Temperature:

The measurement results showed that EMCS readings are 3.3 °F higher than the on-site measured temperature for AHU 250, and 1.4 °F higher for AHU 252. For these two cases, the thermometer probe was inserted into a duct through the soft connection, which is about 1 foot away from the EMCS thermometers.

The on-site readings show very high temperatures 59.7 °F and 72.7 °F for AHU 251 and 253 respectively, where the thermometer probe was inserted into the same place of the EMCS temperature sensors. It seems that the temperature sensors are put into vortex zones; therefore, they are unable to measure the true supply air temperature in these two cases.

In general, it is believed that the pre-coils had a supply air temperature of less than 55 °F. It is suggested that EMCS sensors be relocated.

Cold Deck Supply Air Temperature:

The on-site readings were 6.2 °F and 1.1 °F lower than EMCS readings for AHUs 250 and 251, respectively. These differences may indicate a consistent bias of the EMCS thermometer readings.

In general, the cold deck supply temperature varied from 48 °F to 54.1, with an average of 51.5 °F.

Hot Deck Supply Air Temperature:

The on-site readings were slightly lower than EMCS readings. It should be pointed out that the hot deck temperatures varied from 82.7 °F to 88.4 °F (average of 85.1 °F) while the EMCS setting is 80 °F.

Return Air Temperature:

The on-site readings were about 3 to 4 °F lower than EMCS's reading. However, EMCS takes readings behind the fans, which generates heat, while on-site measurement takes data in front of the fans. To take into account the temperature increase due to the fan, it seems that EMCS readings may be 1 °F higher than the on-site readings. On the basis of on-site measurement, the return air temperature behind the fans is about 77 °F, which is used in the model simulation.

Summary:

The following table summarizes the measured results, design/EMCS settings and the simulation values of the hot deck temperature, the cold deck temperature, the return air temperature, and the pre-coil temperature.

Table A2. Summary of Measured, Model Used and Design Set Temperatures

	Pre-coil (°F)	Cold Deck (°F)	Hot Deck (°F)	Return (°F)
On-site Measured	52.8	51.5	85.1	77
Model Used	55	50	85	77
Design/EMCS Value	60	54.5	80	

The table shows clearly that the model simulation used right values. The temperature settings should be changed and, consequently, $200,000 annual savings are expected at this building.

It is suggested that the pre-coil thermometers for AHU 251 and 253 be relocated. The thermometer readings can be used as control signals for AHUS 250 and 251 after bias corrections (3.3 for AHU 250, 1.45 for AHU 252).

APPENDIX B: DATA QUALITY CHECK

Steam:

The LoanSTAR program measures the number of pumps on. It was converted to GPH by multiplying a factor of 34.91. This factor was determined by comparing the number of pump run and the GPH measured by EMS in UTMB within a short term test.

Figure B-1 compares the LoanSTAR and EMCS measured steam flow rate. The Figure shows that the LoanSTAR measured average flow rate of 4.615 gallon per minute while EMCS measured a value of 4.477 gallon per minute. The relative difference is 3%.

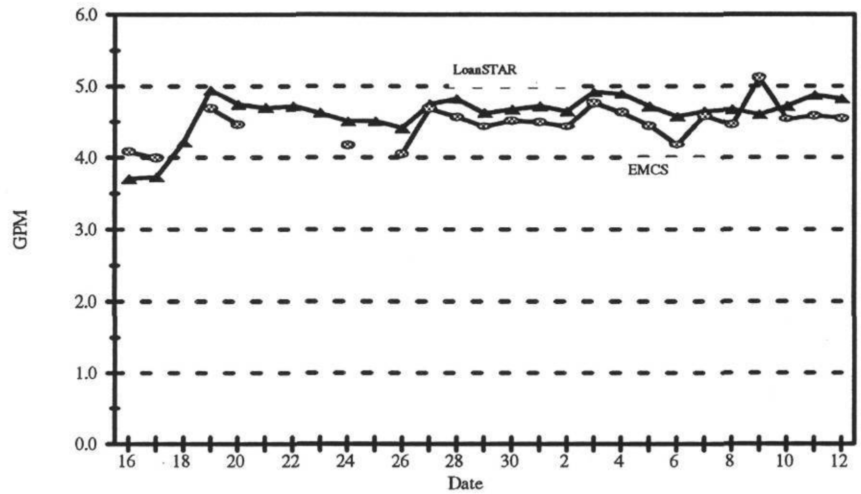

Figure B1: Comparison of LoanSTAR and EMCS Measured Daily Average Steam Consumption from June 16 to July 14, 1993

The LoanSTAR converts GPM to MMBtu/hr by multiplying a factor of 0.4719 (8.667*0.9075/1000*60). Note the water density is taken as 8.667 lb/gal according to the EMS program at UTMB, and the latent heat is taken as 0.9075 kBtu/lb according to LoanSTAR. The EMCS at UTMB converts GPM to MMBtu/hr by multiplying a factor of 0.5533. EMCS assumes that the steam has enthalpy contribution of 1.064 kBtu/lb to the system.

Figure B2 compares LoanSTAR measured daily average steam energy consumption with EMCS measured data from June 16 to July 14, 1993. Figure B2 shows that EMCS measured consumption is about 13% higher than LoanSTAR measured data due to that EMCS uses the enthalpy difference value as 1.064 while LoanSTAR uses 0.9075 ((1.065-0.9075)/0.9075=17%).

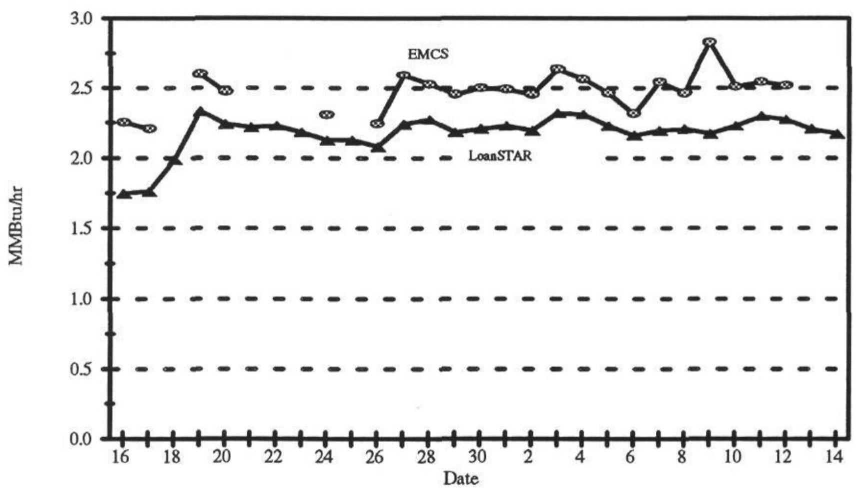

Figure B2: Comparison of LoanSTAR and EMCS measured Daily Average Steam Consumption from June 16 to July 14, 1993

Figure B3 compares LoanSTAR measured monthly steam energy consumption with EMCS measured data. The Figure shows again that the EMCS measured higher steam Btu consumption. However, this difference is due to that EMCS take the energy contribution per pound steam as 1.064 while LoanSTAR uses 0.9075.

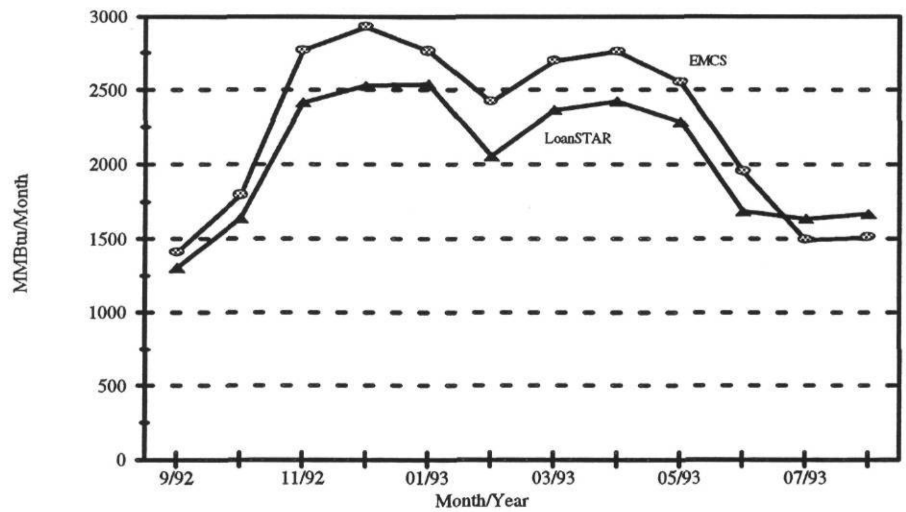

Figure B3: Comparison of LoanSTAR and EMCS Measured Monthly Steam Consumption from September, 1992 to August, 1993

Note that the steam flow rate data was not available for this building.

Chilled Water:

Chilled water energy consumption: LoanSTAR and EMCS at UTMB measured chilled water consumption using different meters. Figure B4 compares daily average chilled water energy consumption measured by LoanSTAR and EMCS from June 16 to July 14, 1993. The LoanSTAR measured energy consumption is about 9% higher than EMCS data.

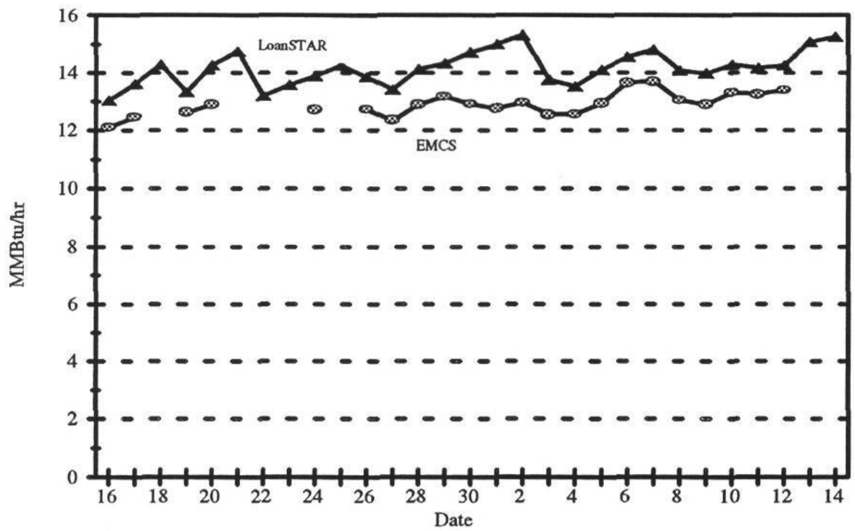

Figure B4: Comparison of LoanSTAR and EMCS Measured Daily Average Chilled Water Consumption from June 16 to July 14, 1993

Figure B5 compares LoanSTAR measured monthly chilled water consumption with that measured by EMCS. It shows that LoanSTAR measured chilled water consumption and EMCS measured results agree very well from October 1992 to March 1993. However, there are relatively big differences during the summer months. It is very likely that one of the meter does not have the correct conversion coefficients for high flow rate. The overall annual energy difference is about 9%. Certainly, the HVAC model with LoanSTAR data is likely to yield a rather accurate estimate of the subsequent energy savings due to O&M changes in the HVAC system.

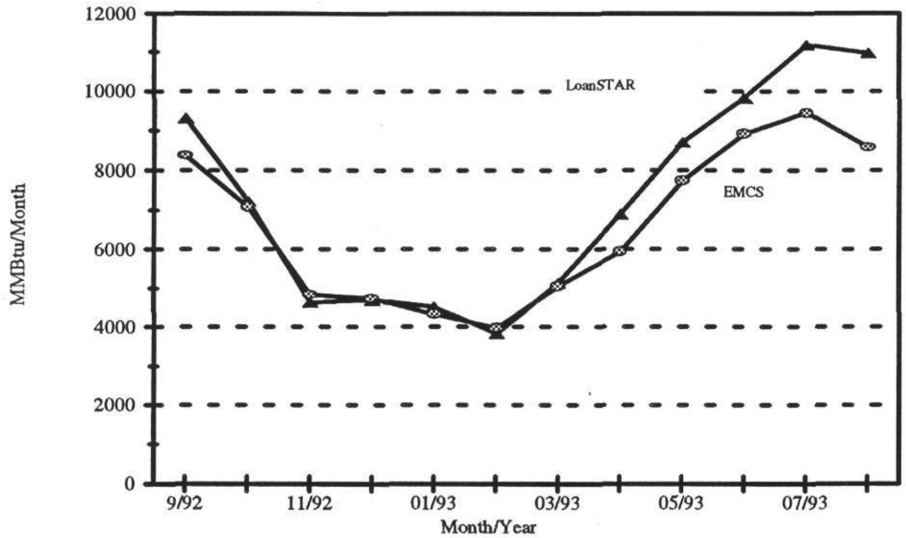

Figure B4: Comparison of LoanSTAR and EMCS Measured Monthly Chilled Water Consumption from September, 1992 to August, 1993

APPENDIX C: SIMPLIFIED SYSTEM MODELS

The simplified schematic of air handling units (AHU) and the building is shown in Figure C1, with the four AHUs assumed as one unit and the building idealized as two zones: interior zone and exterior zone.

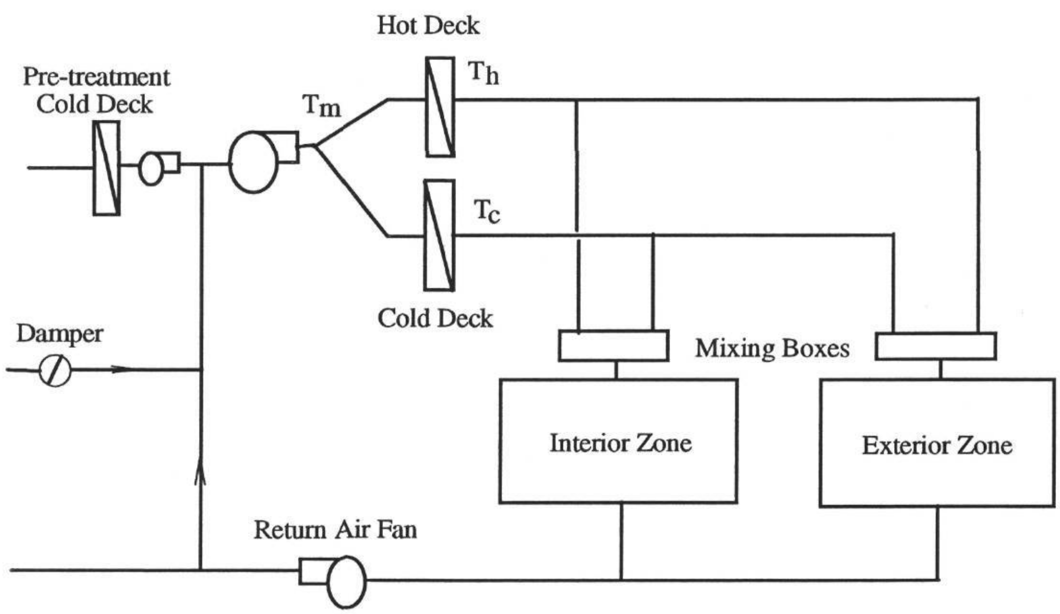

Figure C1: Schematic of HVAC System for John Sealy South Building

The O. A. cold deck is turned on only when ambient temperature is higher than 60 °F. The chilled water consumption due to this cold deck is calculated as:

$$E_{pre} = \dot{m} \times f_o (h_o - h_{pre}) + E_{fan-pre}$$

where E_{pre} is the chilled water consumption of O. A. cold deck, $\dot{m}$ is the total supply air mass flow rate, f_o is the outdoor air intake faction, h_o is the outdoor air specific enthalpy, h_{pre} is the pre-treated supply air specific enthalpy, and $E_{fan-pre}$ is the energy consumed by the fan at the exit of the O. A. cold deck. It is assumed that the fan power is converted to thermal energy completely.

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 the 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 O. A. cold deck supply air temperature, T_r is the return air temperature after the return fan, and other symbols are as defined earlier.

Since constant air flow terminal boxes are used in this building, the air flow rate through each box should not be changed regardless of operation schedules. Consequently, the simplified model requires a constant air flow rate to each zone, although the ratio of the cold air to the hot air changes with zone load, 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 $\dot{m}_{\text{ext}}$ and $\dot{m}_{\text{int}}$ are the air flow rates 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.

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

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

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

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

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

$$\dot{m}_{ext} = \dot{m}_{c,ext} + \dot{m}_{h,ext}$$

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

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

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

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

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

where ω_{int} and $\omega_{ext,}$ are the room air specific humidity in 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 at the exit of cold deck and hot deck respectively, and other symbols are as defined earlier.