ESL-TR-93/10-07

LoanSTAR Monitoring and Analysis Program

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

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

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October, 1993

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EXECUTIVE SUMMARY

The LoanSTAR Monitoring and Analysis Group, Energy Systems Laboratory at Texas A&M University, was requested by University of Texas Medical Branch at Galveston to investigate O&M measures in their five LoanSTAR program buildings. This report describes the suggested O&Ms in John Sealy North Building, a surgical building of 54,494 ft^2, which currently spends $502,100 per year on electricity, steam and chilled water. The suggested O&Ms include optimizing the outside air treatment cold deck reset schedule, the cold deck reset schedule and the hot deck reset schedule. These optimized HVAC operation schedules were 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 $67,000, or 13% of the annual costs, can be realized using the optimized operation schedules which can be implemented without additional costs. Our analysis indicates that the room comfort levels will not be degraded by these measures.

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

1. INTRODUCTION

The John Sealy North Building is a two-story structure attached to the north side of John Sealy Hospital. It houses the primary operating rooms on the second floor and associated facilities on the base floor. This 2-story structure has a total floor area of 54,500 ft^2. The building has light-colored brick walls with window area less than 5% of wall area.

The lighting, people and equipment are the major sources of internal gain in this building. Light energy levels are 3 W/ft^2. About 10% of the lights are incandescent, and 90% are fluorescent.

Four major constant air volume systems serve this building and a kitchen located in the John Sealy Hospital. Air handling unit (AHU) 211 is a dual duct constant volume (DDCV) system with outdoor air pre-treated by air handling unit 212. This system supplies a total of 44,000 CFM air to the first floor and east portion of the second floor. The return air fraction is about 50% for this system. AHU 210, 213, and 335 are single duct constant volume (SDCV) systems. AHU 210 and 335 supply about 45,600 CFM outdoor air to the second floor. AHU 213 provides 25,000 CFM outdoor air to a kitchen area which is located within the John Sealy Hospital Building. Hereafter, AHU 211 and 212 are called the DDCV system while AHUs 210, 213 and 335 are called the SDCV system. Currently, the DDCV system has pre-treatment supply air temperature of 57 °F and supply air temperature of 53 °F. The SDCV systems have an average supply air temperature of 53 °F. Note that there are three additional small AHUs which serve two operating rooms and the mechanical room. However, they are not included in this analysis due to their small capacities.

Hourly building energy consumption data (electricity, chilled water, and steam) was measured by the LoanSTAR program [1] as well as by the EMCS at UTMB. According to the LoanSTAR measured results, this building consumed 3.64 million kWh of electricity in 1992. According to EMCS measured results, this building consumed 46,420 MMBtu of chilled water, and 13,000 MMBtu of steam from July 1992 to June 1993. The energy consumption costs \$502,100/yr or \$9.21/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 (67%), followed by electricity (19%), and steam (13%).

Table 1: Summary of the Annual Energy Consumption at the John Sealy North Building

	Electricity	Chilled-water	Steam	Total
Consumption	3.64 Million kWh	46,420 MMBtu	13,135 MMBtu	
Costs	\$96,835	\$338,866	\$66,397	**\$502,098**
% of Total Cost	19%	**67%**	13%	

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

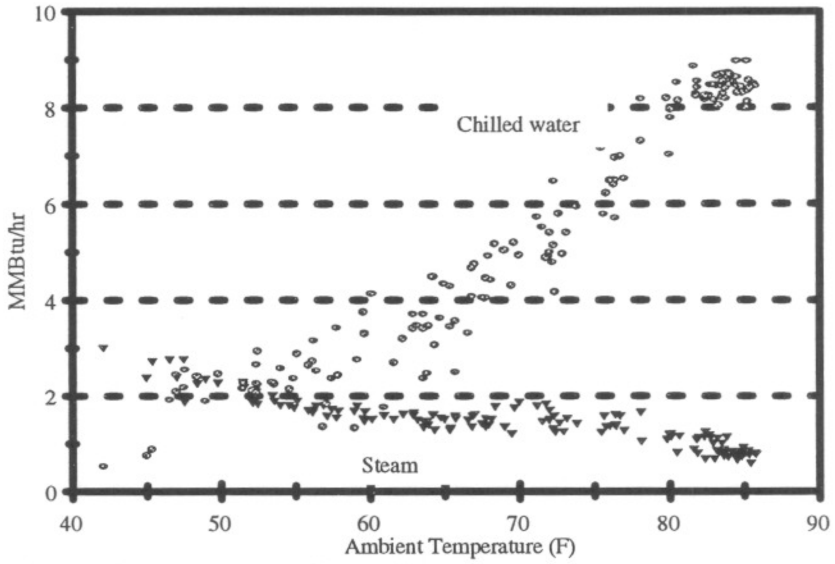


Figure 1: Measured Daily Average Chilled Water and Steam Energy Consumption vs the Daily Average Ambient Temperature, Data were measured from December, 1992 to August, 1993

All the AHUs and their associated equipment are under EMCS control, which is well operated and maintained. 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 for the John Sealy North building at UTMB. It briefly describes the methodology used to identify O&M measures at the John Sealy North Building, presents a simplified HVAC system model used for the present O&M analysis and for optimizing HVAC operation, and discusses the energy and dollar savings.

2. METHODOLOGY

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

1. LoanSTAR information data base browsing. 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 the energy performance history; and

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

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 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 potential;

(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 schedules, air flow rates, and possible nighttime setback, as well as miscellaneous information from the EMCS system, such as measured energy performance.

UTMB personnel accepted the suggestions of optimizing HVAC system but rejected the option to reduce supply air flow rate because they were unsure as to how the occupants would react.

3. Data quality check. Before using the LoanSTAR data to estimate potential O&M savings, the data set is 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 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 which are programmed into a computer simulation code. The simplified computer model uses measured daily average ambient temperature and dew point temperature 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 the 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. 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 temporarily disable the EMCS system and for a few days use the suggested settings instead. Although this test would not show the full potential of optimized schedule savings, it provides an opportunity 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 the DDCV system and the building is shown in Figure 2a where the building is modeled 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]. The Schematic of SDCV system and building is shown in Figure 2b, where the three AHUs are simplified as one and the building is idealized as one zone.

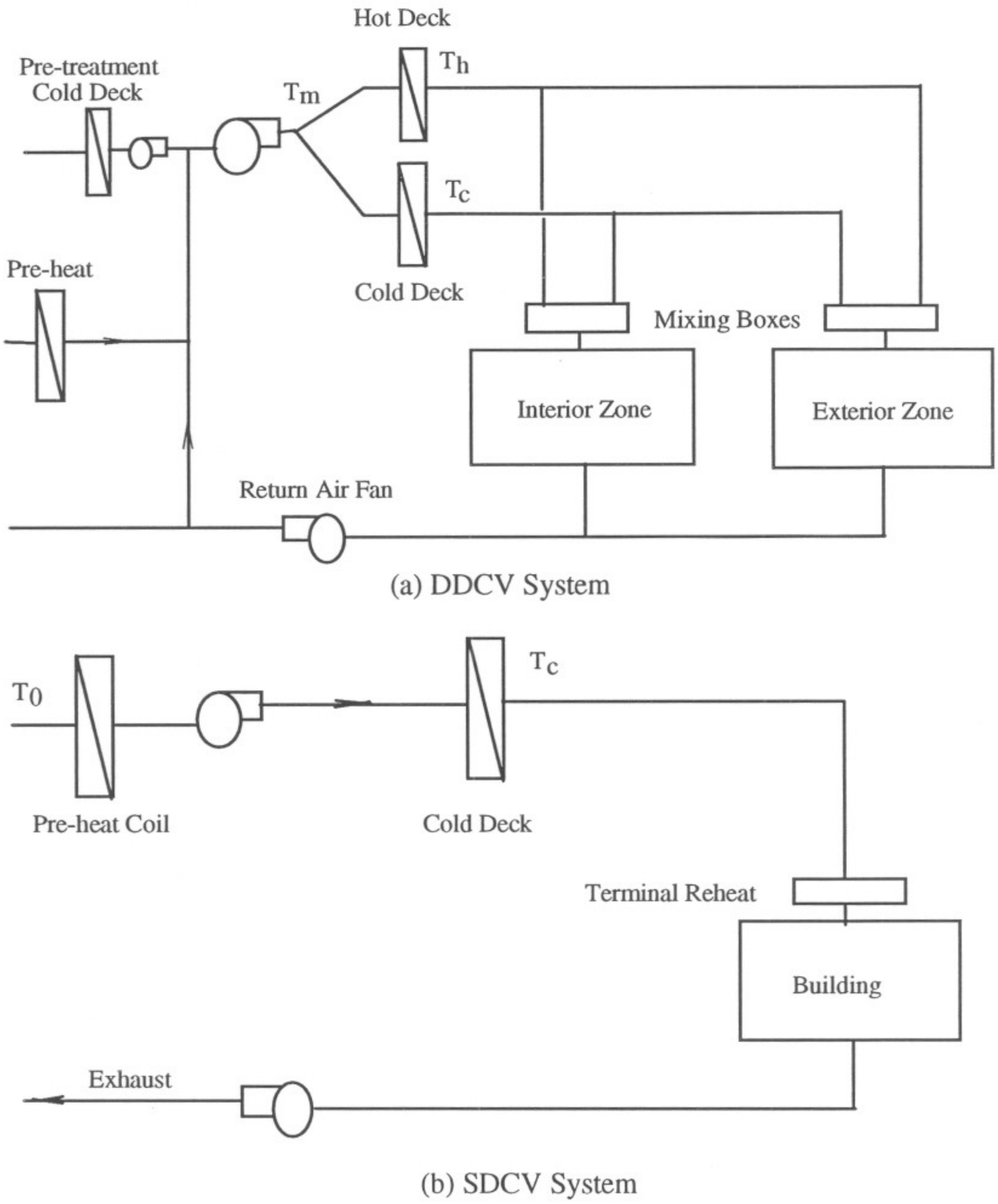


Figure 2: Schematic of HVAC System for John Sealy North Building

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

3.1.1 DDCV System:

The AHU supplies an air flow rate of 44,000 CFM to the building with a total outdoor air intake of 22,000 CFM. The EMCS was programmed for 55 °F as the average pre-treatment cold deck supply air temperature, 55 °F as the main cold deck supply air temperature, and a range of 80 °F to 90 °F range for the hot deck supply air temperature which was varied according to the ambient temperature. However, the EMCS measured 57.7 °F for the outdoor air pre-treatment cold deck and 53 °F for the main cold deck during our visit on July 15, 1993. 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 interior and exterior zones are divided according to the building plan (Figure 3). 35% of the total area (39,696 ft^2) is classified as the interior zone and the rest of the area as the exterior zone for the portion served by the DDCV system. The internal lighting gain is 3 W/ft^2. The equipment heat gain is taken as 10% of the lighting gain. A factor of 0.8 is used to account for lighting and equipment reduction at night. The number of people is estimated by assuming one person for every 40 ft^2 of floor area, and the sensible and latent loads due to people are calculated by assuming standard losses by normal office workers [3].

The building envelope area is calculated as 14,600 ft^2, which includes 275 ft^2 window area for the portion severed by the DDCV system. A heat transfer coefficient of 0.2 Btu/ft^2 °F hr was assumed for walls and 1.0 Btu/ft^2 °F for windows

Air infiltration rates are taken as 0.6 ACH (air change number of building volume in one hour) for the exterior zone and 0.4 ACH for the interior zone.

3.1.2 SDCV System:

The three AHUs supply 70,000 CFM outdoor air to the operating rooms and kitchen located in the John Sealy Hospital. The three AHUs are about the same size while EMCS programed the supply air temperature as 48 °F (AHU 210), 52 °F (AHU 213) and 55 °F (AHU 335). The model uses a temperature of 53 °F according to the EMCS measured results.

The portion served by the SDCV systems is treated as one zone with a total floor area of 35,966 ft^2 and has a total envelope area of 20,900 ft^2 (including a window area of 138 ft^2) . The internal gain from lighting and equipment is taken as 6 W/ft^2. 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 30 ft^2 of floor area, and the sensible and latent loads due to people are calculated by assuming standard losses by normal office workers [3].

A heat transfer coefficient value of 0.1 Btu/ft^2 °F hr was assumed for walls and 1 Btu/ft^2 °F for windows. Air infiltration rates are taken as 0.6 ACH (air change number of building volume in one hour). Note that a relative larger air infiltration rate has been chosen because this building is operated under negative pressure. The domestic hot water and other steam and hot water consumption are estimated as 0.41 MMBtu/hr.

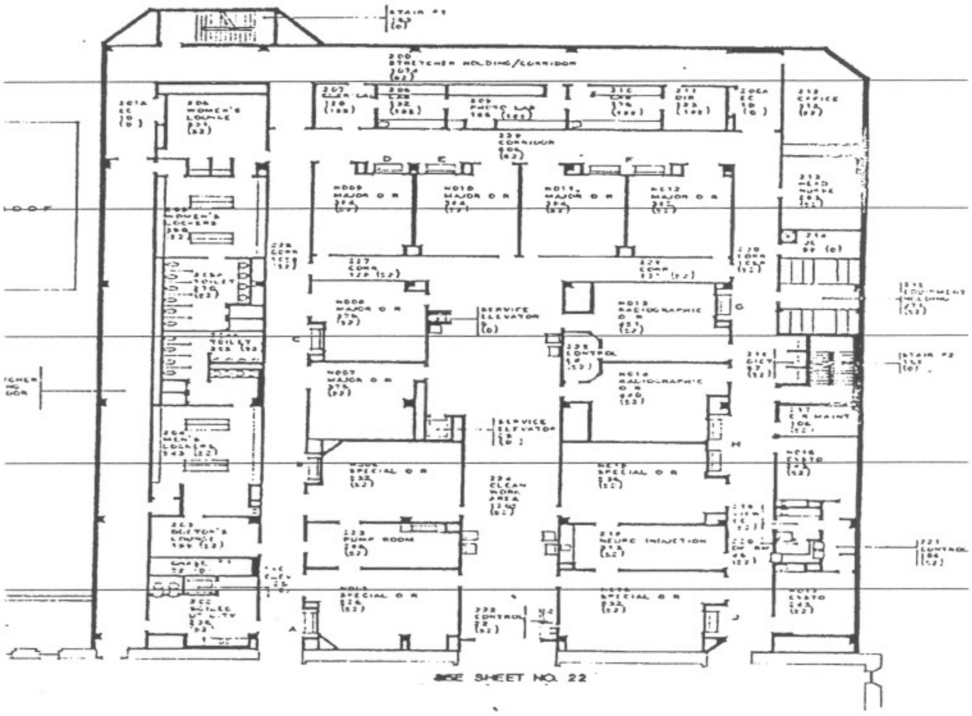


Figure 3: Typical Floor Layout of John Sealy South Building

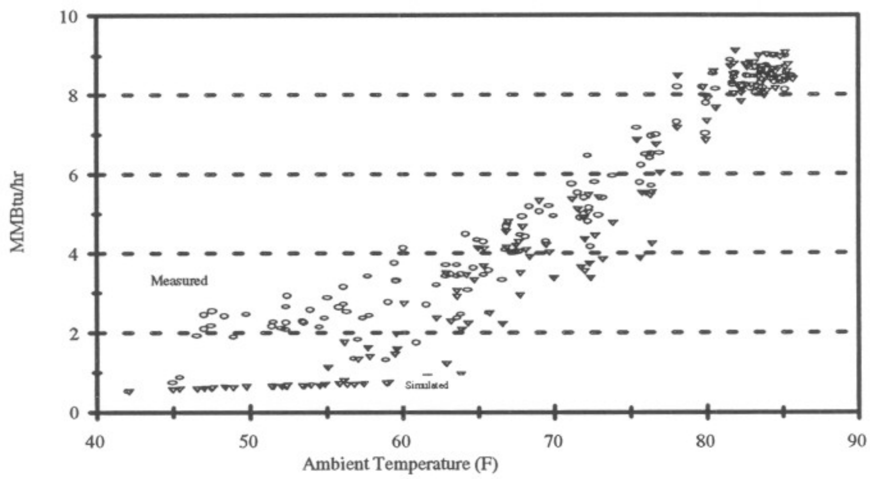
3.2 Model Calibration

Chilled water and steam data measured by the LoanSTAR are compared with EMCS measured data on a daily basis for a month. The comparisons show that the LoanSTAR measured results are consistent with EMCS measured results. Appendix B contains further details.

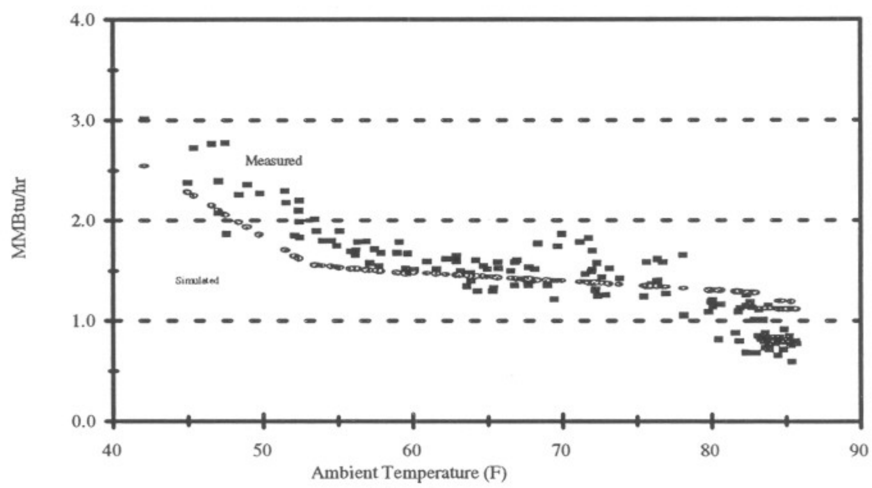
The chilled water and steam energy consumption were predicted with the simplified model using the measured daily average temperature from December 1992 to August 1993. The predicted average chilled water consumption was 12% less than the measured value, while the predicted steam consumption over a period from December 1992 to August 1993 was identical. The standard mean root square errors of the predictions are

1.02 MMBtu/hr and 0.28 MMBtu/hr for chilled water and steam, respectively. The coefficients of variation are 0.19 and 0.20 for chilled water and steam, respectively.

Figures 4 and 5 permit comparison of measured energy consumption and model predicted energy consumption. Figure 4 is a scatter plot of consumption versus temperature, while Figure 5 shows consumption in time series form.



(4a: Chilled Water)



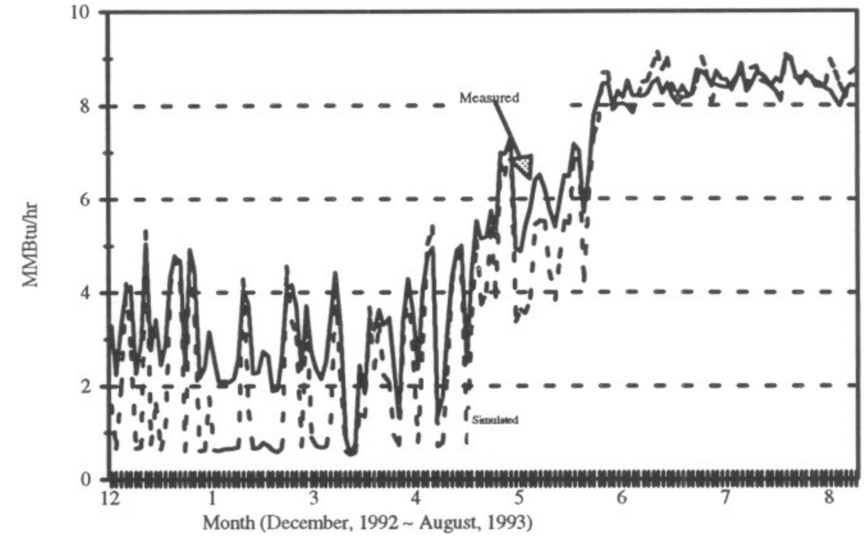
(4b: Steam)

Figure 4: Comparison of the Daily Average Energy Consumption Between Model Predicted and Measured Data from December 1992 to August 1993

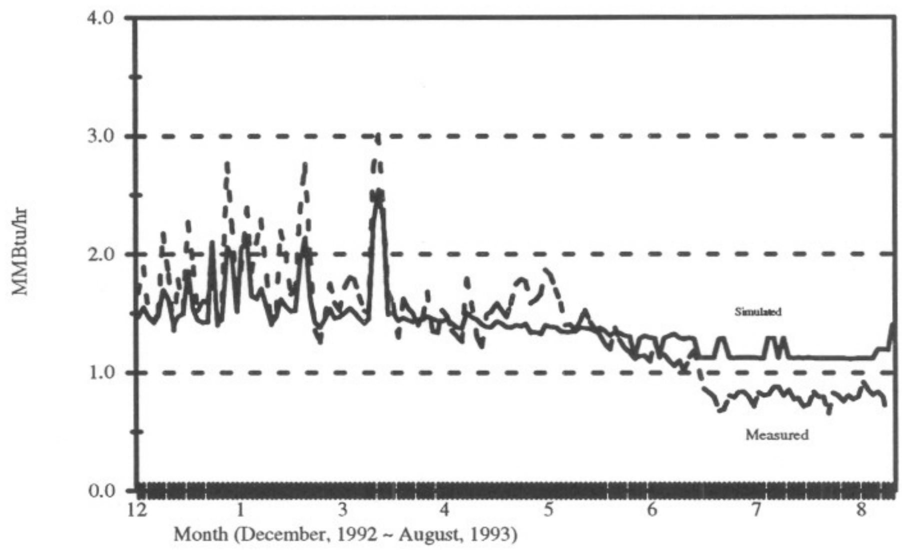
Figure 4a compares the model predicted chilled water energy consumption with the measured chilled water energy consumption. The horizontal axis is the daily average ambient temperature. This figure shows that the predicted consumption matches the measured values very well when the daily average ambient temperature is higher than 60 °F. However, the measured chilled water energy consumption is substantially higher than the predicted chilled water consumption when the daily average ambient temperature is lower than 60 °F. These differences can be explained as follows. When the daily average ambient temperature is lower than 60 °F, the daytime temperature may be higher than 50 °F and nighttime temperature may be lower than 50 °F. Consequently, the pre-heat coil is off at daytime and is on at night. However, the model assumed that the pre-heat coil is off when the daily average ambient temperature is higher than 53 °F. Therefore, the predicted chilled water and steam consumption are lower than the measured values when the ambient temperature is lower than 60 °F. These differences can be reduced by using hourly temperatures. Since these differences have little impact on the saving estimate, the daily average ambient temperature is used in this analysis.

Figure 4b compares the measured steam consumption with the predicted steam consumption. The relative large difference between measured and predicted values are to be noted. The reasons of these differences have been explained above.

Figure 5 compares measured energy consumption with predicted energy consumption in time series (5a for chilled water and 5b for steam). Figure 5 shows again that the relative large differences between measured and predicted values occurred during winter months (December to March) when the daily ambient temperature is lower than 60 °F. It is also noted that the measured steam energy consumption is lower than the predicted value from July to August 1993. The steam consumption is low due to hot deck being shut off, while the model simulation assumes hot deck to be on.



(5a: Chilled Water)



(5b: Steam)

Figure 5: Comparisons of the Daily Average Energy Consumption in Time Serie Form, Data were measured from December, 1992 to August, 1993

The calibrated simplified model was used to calculate annual energy consumption using bin data for outdoor temperature. Lacking 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-bins with 3 °F width for each bin are used. 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.

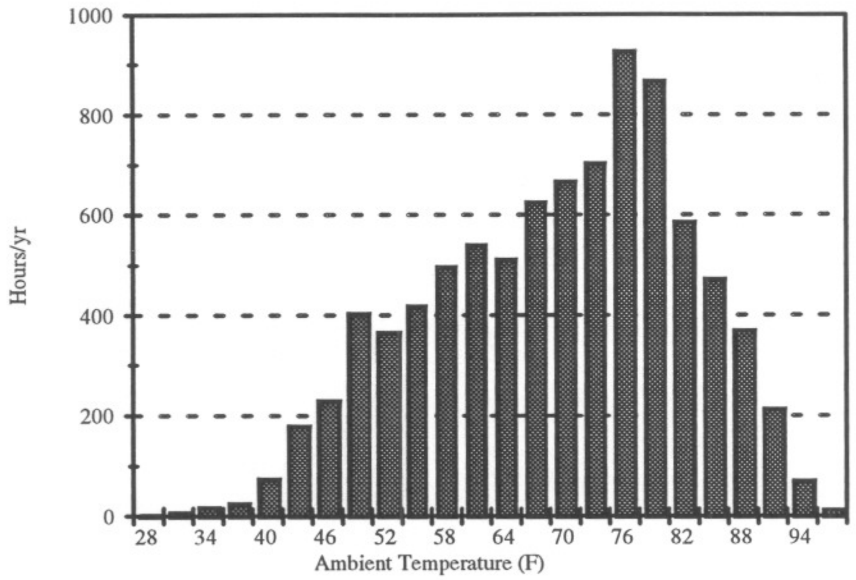


Figure 6: Houston Bin Temperature Chart Generated Using LoanSTAR Measured Hourly Temperature Data from July 1, 1992 to June 30, 1993.

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.

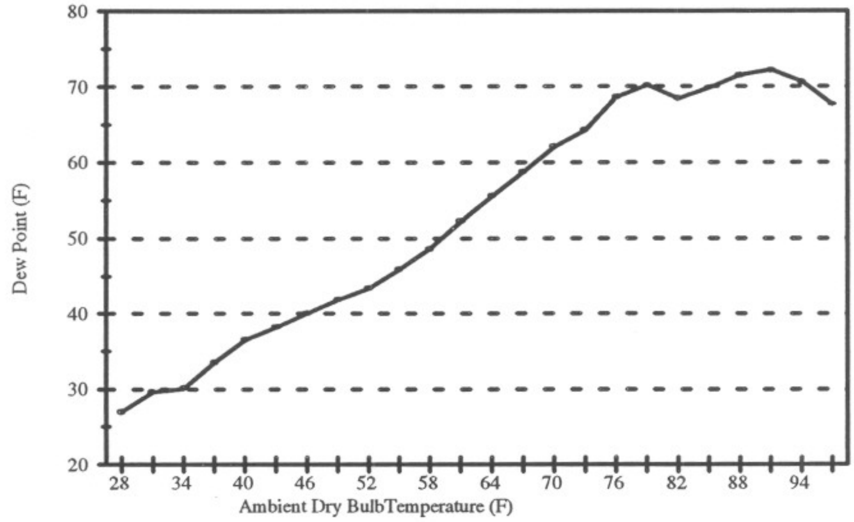


Figure 7: Mean Coincident Dew Point Temperature as a Function of Dry Bulb Temperature in Houston for July 1, 1992 through June 30, 1993

Table 2 summaries the values of the key parameters used in the calibrated simplified model and the baseline settings of the EMCS system. The model assumes the pre-heat coil to become active when the ambient temperature is lower than 53 °F, while the EMCS system starts it when the ambient temperature is lower than 46 °F. This change would reduce the effects of diurnal may relieve the impact of daily temperature variation when the daily average temperature is used in the model simulation.

Table 2: Summary of the Model Calibration Parameter Adjustment

Item	Schedule (EMS)	Schedule (Model)
DDCV		
Supply air flow rate (CFM)	43,900	44,000
Return air fraction	0.51 (Blue prints)	0.51
Pre-cold deck temperature °F	57	57
Main-cold deck temp. °F	55	53
Hot deck °F	Min(95, 85+0.2*(90-T0))	Min(95, 85+0.2*(90-T0))
Return air temperature °F	77	77
Room air temperature °F	72	72
SDCV		
CFM	70,680	70,680
Cold deck temperature	53	53

Pre-heat deck	If T0<46 then
	51
	else
	off
		If T0<53 then
		53
		else
		off

Terminal reheat	Thermal balance	Thermal balance

4. OPTIMIZING 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 are imposed: 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 iterative process. A best operation schedule is first chosen based on prior experience. Then, energy (chilled water and steam) and mechanical operation performance (air flow through cold and hot ducts) are predicted using the simplified model. Predicted energy and mechanical performance results are compared with the best results so far obtained, modification to the operation schedule is made and a new simulation performed. This process is repeated until the operation schedule is considered optimal 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 a lower pre-treatment supply air temperature but a higher main coil supply air temperature for DDCV system. This change will maintain the dehumidification capacity of the system but reduce re-heat substantially. The optimized operation schedule increased cold deck supply air temperature for the SDCV systems to account for the variation of seasonal load.

Table 3: Comparison of Operation Schedules

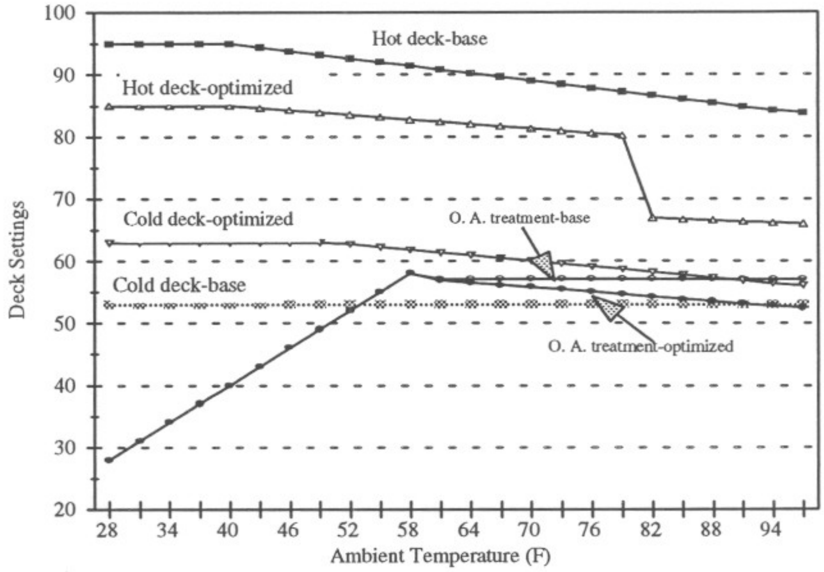
Item	Base	Optimized
	DDCV	

O. A. treatment coil	If T0>60 °F then
	57 °F,
	else
	Off
		if T0>60 °F then
		Min(57, 57-0.125*(T0-60))
		else
		off

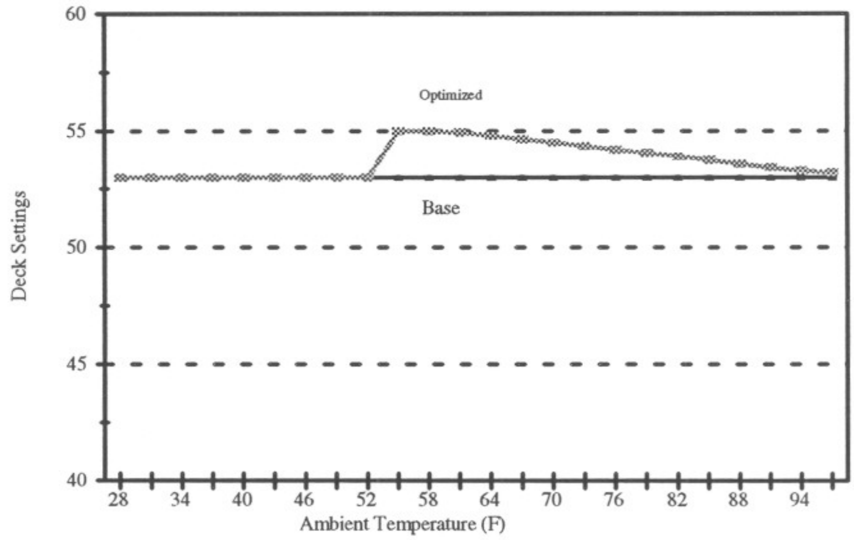
Main cold deck	53 °F	Min(63, 63-0.15*(T0-50))

Hot deck	Min(95, 85+0.2*(90-T0))	If T0<80 then
		Min(85, 85-0.125*(T0-40))
		Else
		off

	SDCV	
Cold deck	53	Min(55, 55-0.05*(T0-60))



(8a: DDCV System)



(8b: SDCV Systems)

Figure 8: Base and Optimized Cold & Hot Deck Schedules

The optimized schedule changes of cold deck temperature with ambient temperature can be performed by the EMCS without additional expense. The hot deck supply air temperature is lower compared to the base schedule, while still being high enough to satisfy heating requirements.

The optimized schedule requires that the pre-treatment cold deck supply air temperature be 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 pre-treatment 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.

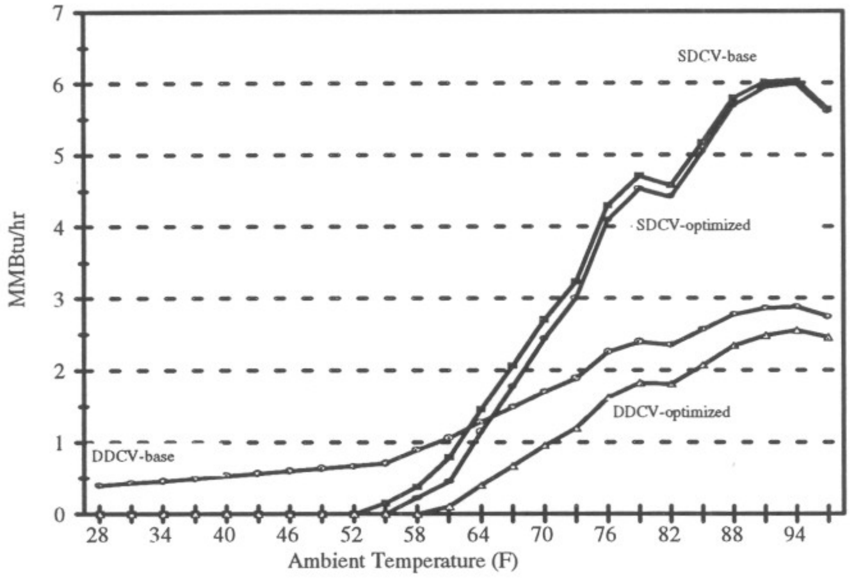
5. RESULTS AND DISCUSSIONS

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 product of the energy consumption and number of hours at each bin temperature.

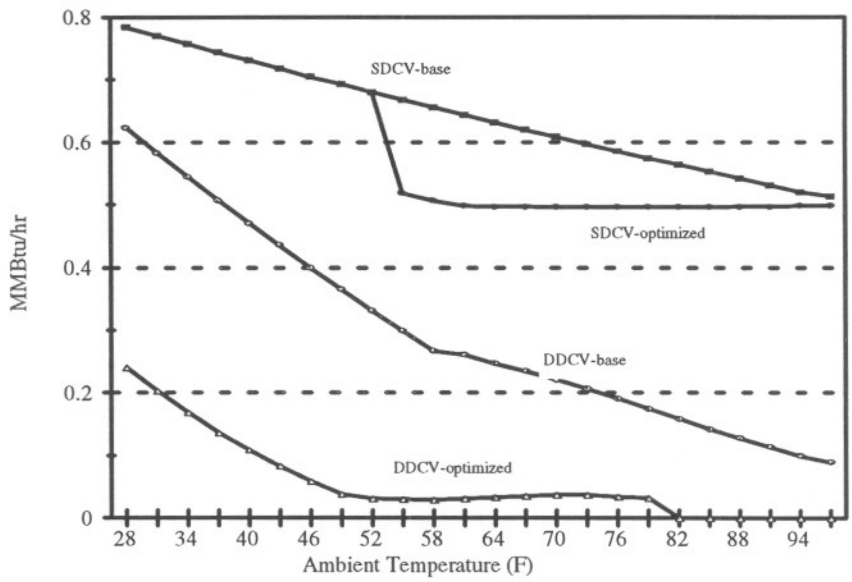
Figure 9 compares the optimized energy performance with the base energy performance. The horizontal axis is the ambient bin temperature. The vertical axis is the energy consumption in MMBtu/hr. Figure 9a shows results for chilled water and Figure 9b for steam.

Figure 9 shows that the optimized DDCV schedule can reduce chilled water energy consumption by 0.7 MMBtu/hr and steam energy consumption by 0.4 MMBtu/hr. The optimized SDCV system schedule can reduce chilled water energy consumption by 0.3

MMBtu/hr and steam energy consumption by 0.1 MMBtu/hr on average although the savings is temperature dependent.



(9a: Chilled Water)



(9b: Steam)

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 DDCV schedule maintains the current room relative humidity levels. The optimized SDCV schedule may increase room relative humidity levels by 3%. It should be pointed out that the model analysis did not simulate the humidifier. Therefore, the low relative humidity levels for SDCV conditioned zone does not represent realistic room relative humidity levels when the ambient temperature is lower than 60 °F.

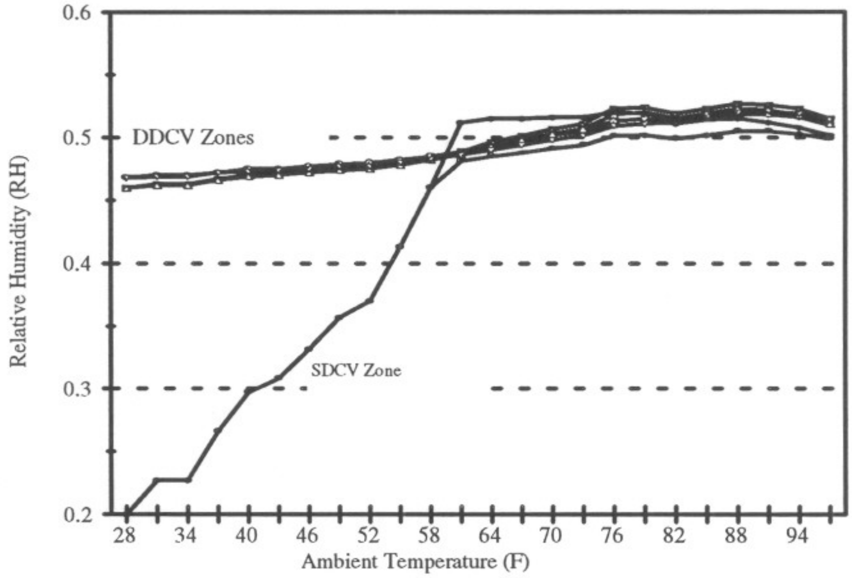


Figure 10: Comparison of the Predicted Room Relative Humidity under Both the Base and the Optimized Operation Schedules

Figure 11 compares the predicted air flow rates through cold and hot air ducts under both the base and the optimized schedules. This figure shows that the optimized schedule can reduce hot duct air flow significantly. However, the maximum air flow rate through the cold deck is only slightly higher than the maximum air flow rate under the base operation schedule.

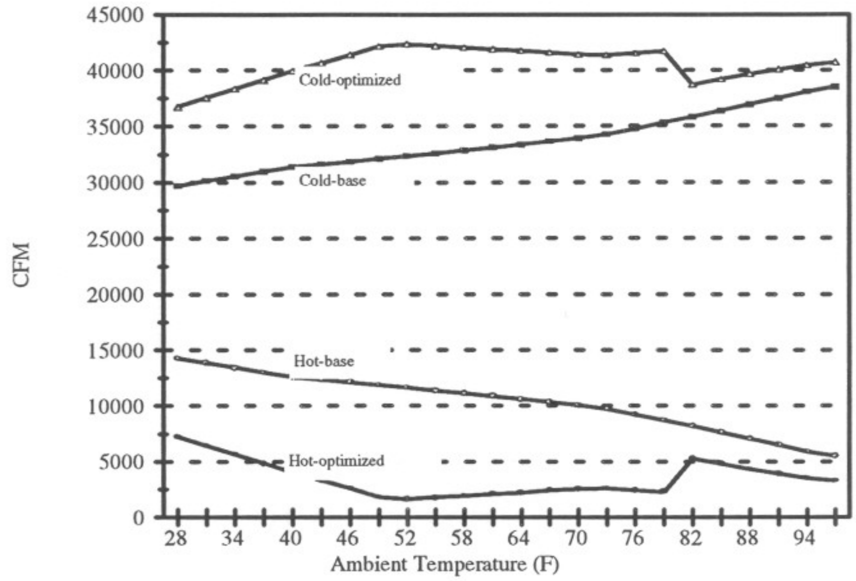


Figure 11: Comparison of Air Flow Rates Through the Cold Deck and the Hot Deck under Both the Base and the Optimized Schedules

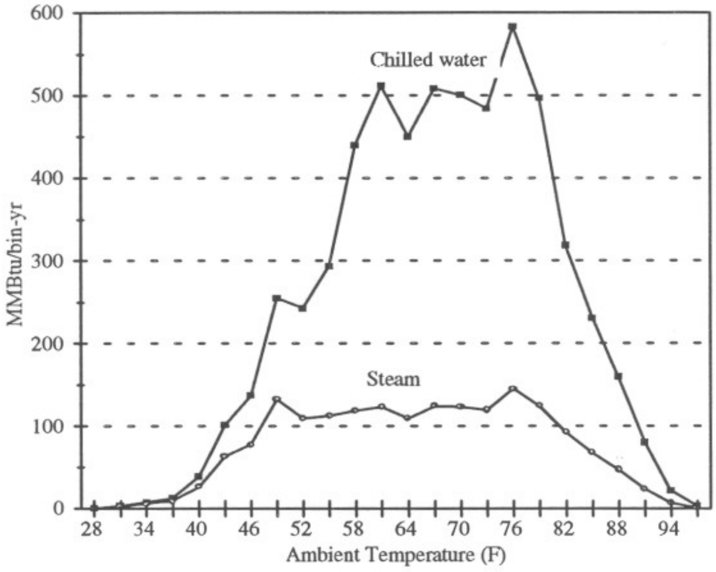
The potential annual energy saving was calculated as the difference in energy consumption under the base and optimized schedules. The results are shown in Figure 12 (12a for DDCV system and 12b for SDCV systems). The horizontal axis is the ambient bin temperature and the vertical axis is the potential annual energy savings for each bin.

The overall optimized energy performance and the potential savings are summarized in Table 4. It shows that the optimized DDCV system schedule can reduce:

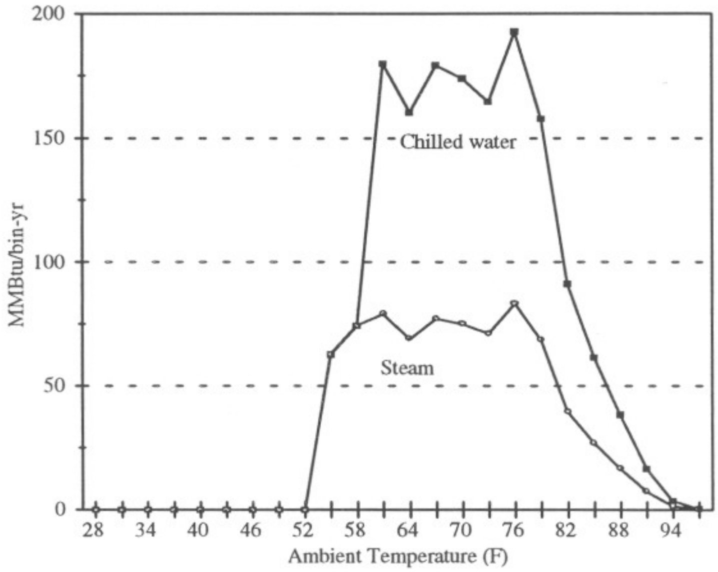
(i) annual chilled water consumption from 38,480 MMBtu to 31,050 MMBtu with a savings of 7,430 MMBtu/yr (including 5,880 MMBtu/yr from DDCV system and 1,560 from SDCV systems); and

(ii) reduce annual steam energy consumption from 7,910 MMBtu to 5,400 MMBtu with a savings of 2,520 MMBtu/yr (including 1,760 MMBtu/yr from DDCV system and 750 MMBtu/yr from SDCV systems).

These energy savings reduce the annual cost by $51,830 for chilled water and $15,150 for steam. The total potential savings are $67,000 which is 13% of the building annual energy cost, or 15% of the chilled water and steam energy costs.



(12a: DDCV System)



(12b: SDCV Systems)

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

Description	Consumption		Savings					
	MMBtu		MMBtu		Dollars		Total	
	Ch-Water	Steam	Ch-Water	Steam	Ch-Water	Steam	Dollars	%
Base	38,480	7,910						
Optimized DDCV	32,600	6,150	5,880	1,760	42,900	8,900	51,800	10%
Optimized SDCV	36,930	7,160	1,560	750	11,400	3,800	15,100	3%
Total	31,050	5,400	7,430	2,520	54,300	12,700	67,000	13%

Note:

The annual energy costs were $502,100, which include $96,800 electricity costs (1992, John Sealy South, LoanSTAR measured energy consumption data), $338,900 chilled water costs, and $66,400 steam costs (according to EMCS measured consumption from July 1992 to June 1993).

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.

Table 5 summarizes the energy indices of the John Sealy North building based on gross floor area (75,662 ft^2, which includes 18,000 ft^2 kitchen area in John Sealy Hospital). The optimized schedule can reduce chilled water consumption per unit floor area from 0.510 to 0.410 MMBtu/ft^2-yr and reduce steam energy index from 0.105 to 0.071 MMBtu/ft^2-yr. The potential chilled water plus steam savings are \$0.89/ft^2-yr.

Table 5: Summary of Thermal Energy Indices

Item	Ch-water MMBtu/ft^2yr	Steam MMBtu/ft^2 yr	Savings MMBtu/ft^2 yr		Savings Dollar
Base	0.560	0.115	Ch-water	Steam	Total
Optimized	0.470	0.079	0.090	0.036	\$0.84

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.

Other retrofit measures may also exist, such as economizer cycles and reducing air flow rate. However, these retrofit measures are outside the scope of the current study.

6. CONCLUSIONS

The annual building energy costs can be reduced by \$67,000 (i. e. 13% of the building annual cost) by using optimized operation schedules. The optimized operation schedules, developed by minimizing thermal energy consumption in the building, can be implemented by changing the EMCS program. The optimized operation schedules do not degrade the room comfort levels.

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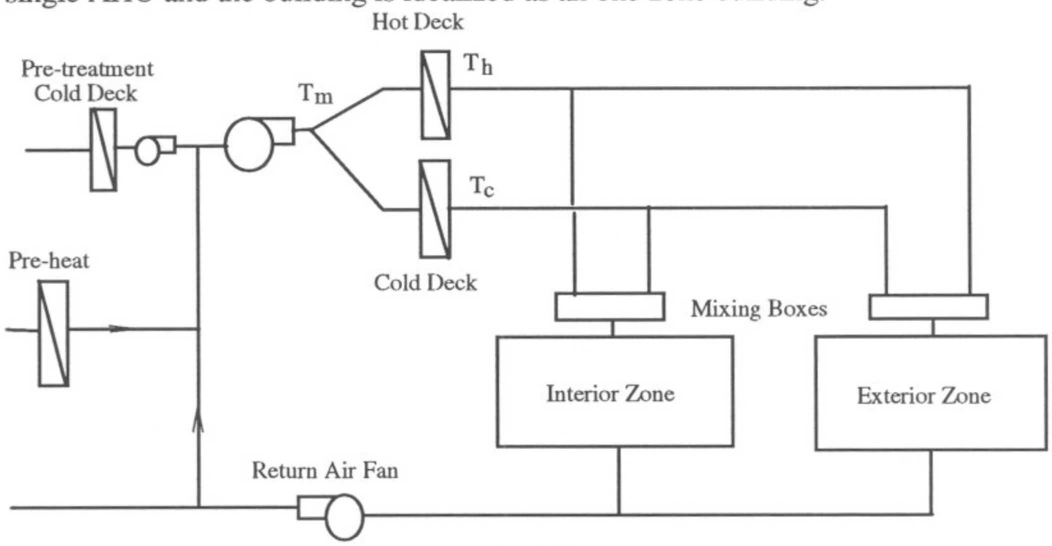
4. B. W. Olesen, August, 1993 "Standards for Design and Evaluation of the Thermal Indoor Environment," ASHRAE Journal, Vol. 35, No. 8.

ACKNOWLEDGMENTS

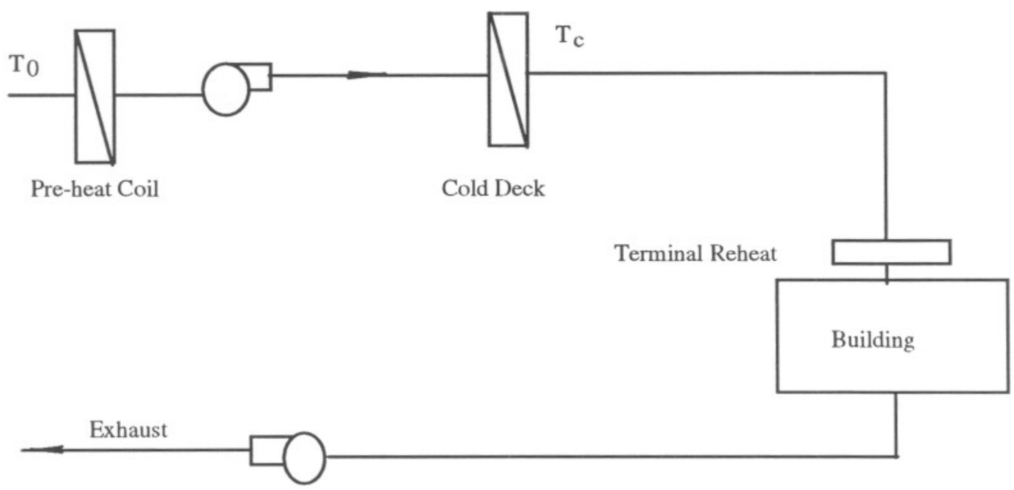
Contributions from the Department of Physical Plant at UTMB are greatly appreciated. The EMCS measured energy data that was provided and discussion regard optimized operation schedules were invaluable to this study. In particular, we would like to thank Mr. Ed. White, the executive director of the Department of Physical Plant, Mr. John Windham, the HVAC foreman, and Mr. Steven Hodgson. We would also like to thank Susan Swanson for her editing assistance.

APPENDIX A: SIMPLIFIED SYSTEM MODELS

The schematic of the DDCV system and the building is shown in Figure A1a where the building is treated as two zones: an interior zone and an exterior zone. This modification has been justified by Katipamula and Claridge [2]. The schematic of the SDCV system and the building is shown in Figure A1b, where three AHUs are treated as one single AHU and the building is idealized as an one zone building.



(A1a) DDCV System



(A1b) SDCV System

Figure 1A: Schematic of HVAC System for John Sealy North Building

DDCV System:

The Outside Air treatment cold deck is turned on only when ambient temperature is higher than 60 °F. The chilled water consumption of 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 outside air treatment cold deck, $\dot{m}$ is the total supply air mass flow rate, f_o is the outdoor air intake fraction, h_o is the outdoor air specific enthalpy, h_{pre} is the pre-treated supply air specific enthalpy, and $E_{fan\text{-}pre}$ is the energy consumed by the fan at the exit of the pre-treatment cold deck.

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 pre-treatment 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.

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

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

where $\dot{m}_{\mathrm{ext}}$ and $\dot{m}_{\mathrm{int}}$ 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:

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

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

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

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

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

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

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

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

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

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

where ω_{int} and $\omega_{ext,}$ 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.

SDCV Systems:

The pre-heat deck is turned on only when ambient temperature is lower than 46 °F. The steam consumption due to pre-heating is calculated as:

$$E_{pre} = \dot{m} \times C_p (T_{pre} - T_0) - E_{fan-pre}$$

where E_{pre} is the steam consumption of pre-heat coil, $\dot{m}$ is the total supply air mass flow rate, C_p is the specific heat of air, T_o is the outdoor air temperature, T_{pre} is the pre-heat supply air temperature, and $E_{fan-pre}$ is the energy consumed by the fan.

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

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

where E_c is the chilled water energy consumption of the main cold deck, $\dot{m}$ is the mass flow rate, 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 terminal re-heat box is calculated by the following formula:

$$E_h + Q_{\text{int}} + \dot{m}_{\text{inf}} c_p (T_0 - T_{room}) = \dot{m} c_p (T_{room} - T_c)$$

where E_h is the steam energy consumption of the re-heat terminal boxes, M is the mass air flow rate, T_{room} is the room temperature, Q_{int} is the sensible load, T_o is the outdoor temperature, M_{inf} is the air infiltration rate.

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

$$\omega = \frac{W + \dot{m}_c \times \omega_c + \dot{m}_{\text{inf}} \times \omega_o}{\dot{m}_c + \dot{m}_{\text{inf}}}$$

where ω is the room air specific humidity, W is the moisture production, ω_c is the specific moisture at the exit of cold deck, and other symbols are as defined earlier.

APPENDIX B: DATA QUALITY CHECK

Steam:

Figure B1 compares LoanSTAR measured daily average steam energy consumption data with EMCS measured data from June 16 to July 14, 1993. Figure B1 shows that both data are very close.

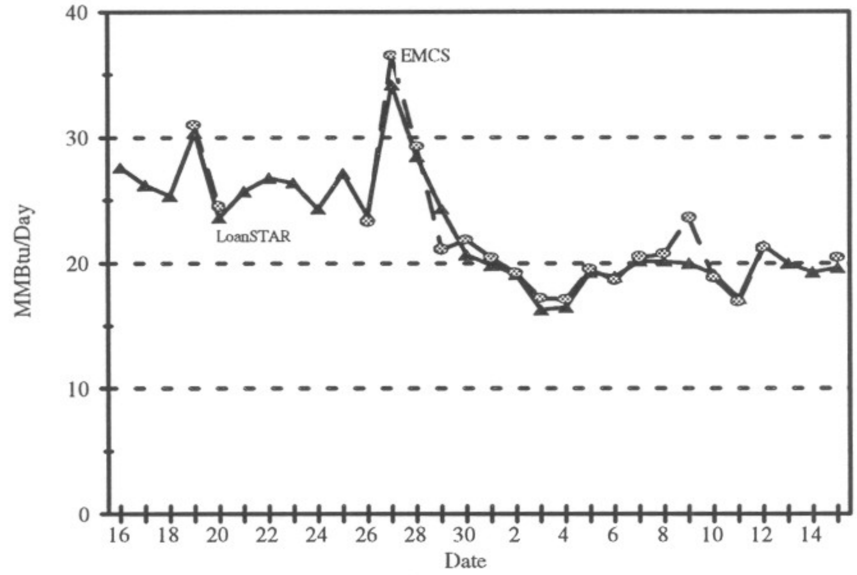


Figure B1: Comparison of LoanSTAR and EMCS measured Daily Average Steam Consumption from June 16 to July 14, 1993

Chilled Water:

Figure B2 compares daily average chilled water energy consumption measured by LoanSTAR and EMCS from June 16 to July 14, 1993. Again both data are very close.

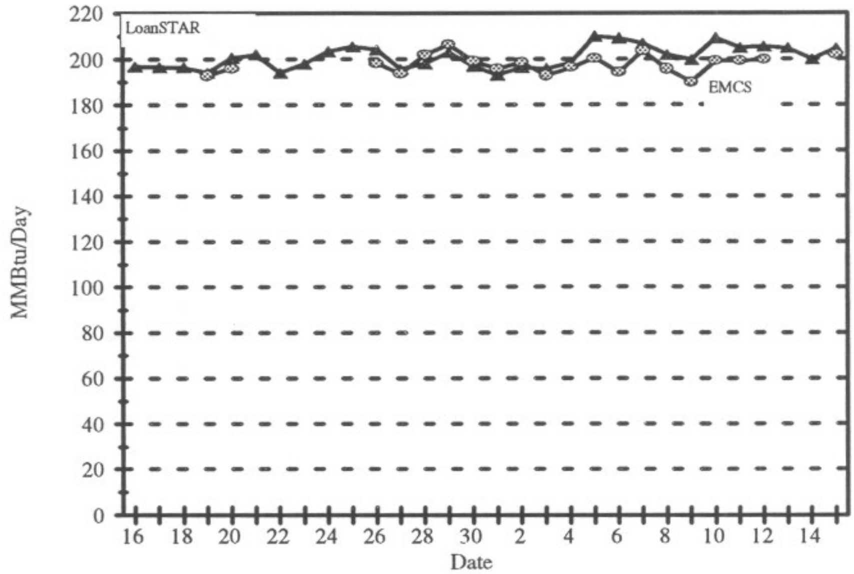


Figure B2: Comparison of LoanSTAR and EMCS Measured Daily Average Chilled Water Consumption from June 16 to July 14, 1993