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LoanSTAR Monitoring and Analysis Program

Potential Energy Savings from Optimized Schedule & Economizer Cycles in the Moody Library Building at UTMB

Submitted to the
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by the
Monitoring and Analysis Group (Task E)

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EXECUTIVE SUMMARY

This report presents the results of a study which was initiated in order to estimate the potential energy savings due to optimizing the HVAC operation schedule and using economizer cycles in the Moody Library Building located at the University of Texas Medical Branch at Galveston, Texas (UTMB). An optimized HVAC operation 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 $46,500 can be realized by optimizing the operation schedule and partially closing cold deck coils. 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 economizer cycle can reduce the annual energy costs by amount of $26,100. These two measures can reduce the annual energy costs by $71,500, or 37% of the total building energy costs.

To maintain the humidity levels, the optimized schedule should be implemented by partially closed coils. It cannot be implemented by increasing cold deck temperature.

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POTENTIAL ENERGY SAVINGS FROM OPTIMIZED SCHEDULE & ECONOMIZER CYCLES IN THE MOODY LIBRARY BUILDING AT UTMB

1. INTRODUCTION

The Moody Library houses book collections, offices, conference rooms, and necessary service facilities at the UTMB with a total gross area of 67,380 ft^2. The facility is in operation about 5,486 hours per year. This building is six-story with a core 1st floor 5th and 6th floor. The major facilities are located on the 2nd to 4th floors. The 5th and 6th floors serve as bindery operation, storage and mechanical rooms. The building has light-colored brick walls with two huge glass covers on north and south walls. The total exterior envelope is 28,000 ft^2 with a glass area of 7,364 ft^2. Light energy levels are 2.8 W/ft^2 on average and varied substantially from area to area. The building requires a strict indoor relative humidity and temperature condition (50% and 70 °F) to prevent deterioration of books.

There are two single duct constant air volume systems, which supply 73,000 CFM air to the building with about 4% outdoor air intake. The two systems have the same size and serve the south and north parts, respectively. Chilled water and steam are supplied by the central plant. Two small air handling units (AHU) have been installed in 1992 to maintain proper relative humidity and temperature condition in the rare book rooms when the major AHUs are turned off at night and weekends. However, these two small AHUs can not be used due to mis-design. Therefore, the two major AHUs are operating 24 hours a day year around. The AHUs' cold deck temperature is maintained at 56 °F in order to maintain suitable room relative humidity levels. Consequently, a substantial amount of reheat takes place.

Hourly building energy consumption data (electricity, chilled water, and steam) are being measured by the LoanSTAR program as well as by the EMCS at UTMB. According to the LoanSTAR measured results, this building consumed 1.57 million kWh, 15,100 MMBtu chilled water in 1992, and 8,500 MMBtu steam from July 1992 to June 1993. This energy consumption costs $194,900/yr or $2.89/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 (56%), followed by steam (22%), and electricity (21%).

Table 1: Summary of the Annual Energy Consumption at the Moody Library Building

	Electricity	Chilled-water	Steam	Total
Consumption	1.57 Million kWh	15,100 MMBtu	8,500 MMBtu	
Costs	$41,800	$110,000	$43,200	$194,900
% of Total Cost	21%	**56%**	22%	

Figure 1 shows the measured daily average chilled water and steam energy consumption versus the ambient temperature. It clearly shows that a substantial amount of steam is used on very hot summer days, which indicates that substantial reheat is present in this building. We also note low chilled water and steam energy consumption for a few very hot days when the hot deck was turned off and the cold deck temperature was increased. Although this operation could not be continued due to increased room relative humidity levels, it did demonstrate that reducing the amount of reheat can save substantial steam (0.5 MMBtu/hr) and chilled water energy (0.5 MMBtu/hr) if problem associated with the room relative humidity could be solved by other measures.

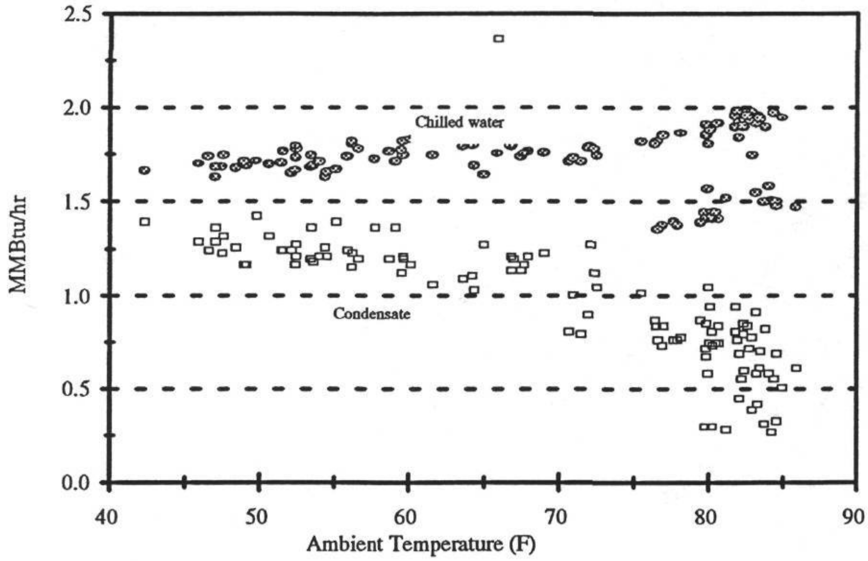

Figure 1: Measured Daily Average Chilled Water and Steam Energy Consumption Versus the Daily Average Ambient Temperature. Data Were Measured from July 14, 1992 to June 29,1993.

Both air handling units (AHU) and their associated equipment are under the 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 briefly describes a study of potential O&M improvements conducted for the Moody Library Building at UTMB. It also describes the methodology used to identify O&M measures at the Moody Library Building, presents a simplified HVAC system model used for the O&M analysis and operation optimization, and discusses the energy and cost savings.

2. METHODOLOGY

The methodology used to explore the energy conservation 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 scatter plots of all monitored channels for each week;

(iv) the Monthly Energy Consumption Report (MECR), which presents an overview of 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) the HVAC system operation could be optimized by reducing reheat, (b) the air flow rates could be reduced, and (c) economizer cycles could be cost effective, and (d) HVAC systems may be shut down during unoccupied hours.

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 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 optimizing HVAC system and economizer cycles while rejected reducing air flow rate because of concern 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 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 certain system parameter 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 thoses of the 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 temporarily disable the EMCS

system and for a few days use the suggested setting 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 air handling units (AHU) and the building is shown in Figure 2, where the two 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].

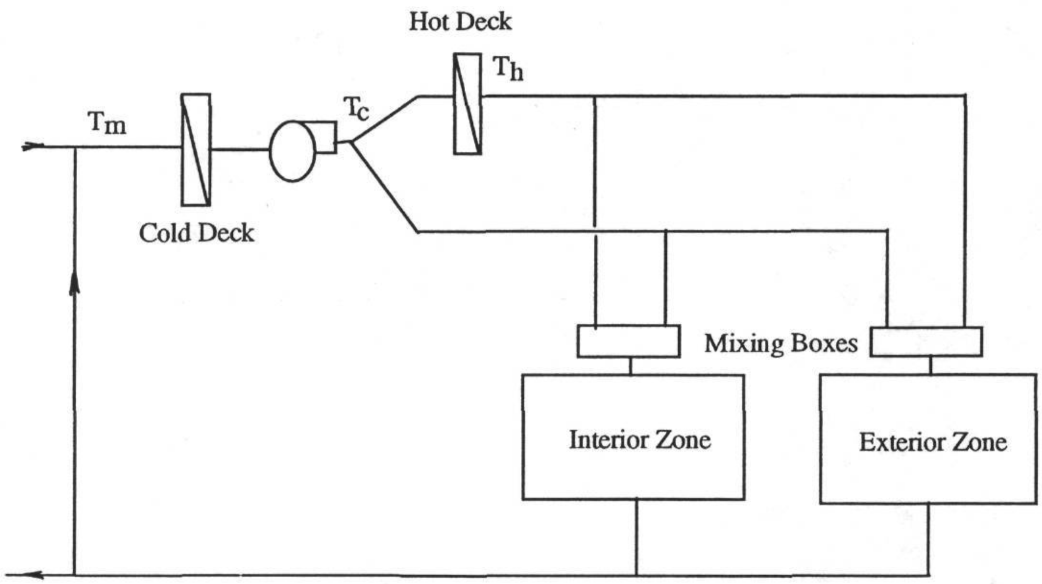

Figure 2: Schematic Chart of Air Handling Units for Moody Library Building

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

The two AHUs supply 73,000 CFM air to the building with a total outdoor air intake of 2,900 CFM while the initial design assumed 11,000 outdoor air intake (see Appendix B for detail). The EMCS was programmed for 56 °F as the cold deck supply air temperature, and a range of 80 °F to 90 °F as the hot deck supply air temperature, which was varied according to the ambient temperature. EMCS measured results show that the building has an average room temperature of about 72 °F and return air temperature of 74 °F.

The interior and exterior zones are divided according to the building plan (Figure 3). 30% 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 (67,380 ft^2). The internal gain due to lights is taken as 2.8 W/ft^2 based on the lighting capacity and other internal gain is taken as 10% of the lighting gain. 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 each 60 ft^2 of conditioned area, and the sensible and latent loads due to people are calculated by assuming that they are all normal office workers. The domestical hot water and other steam and hot water consumption are estimated as 0.217 MMBtu/hr, which was determined from the measured steam consumption when the hot decks were shut off during a few of summer days.

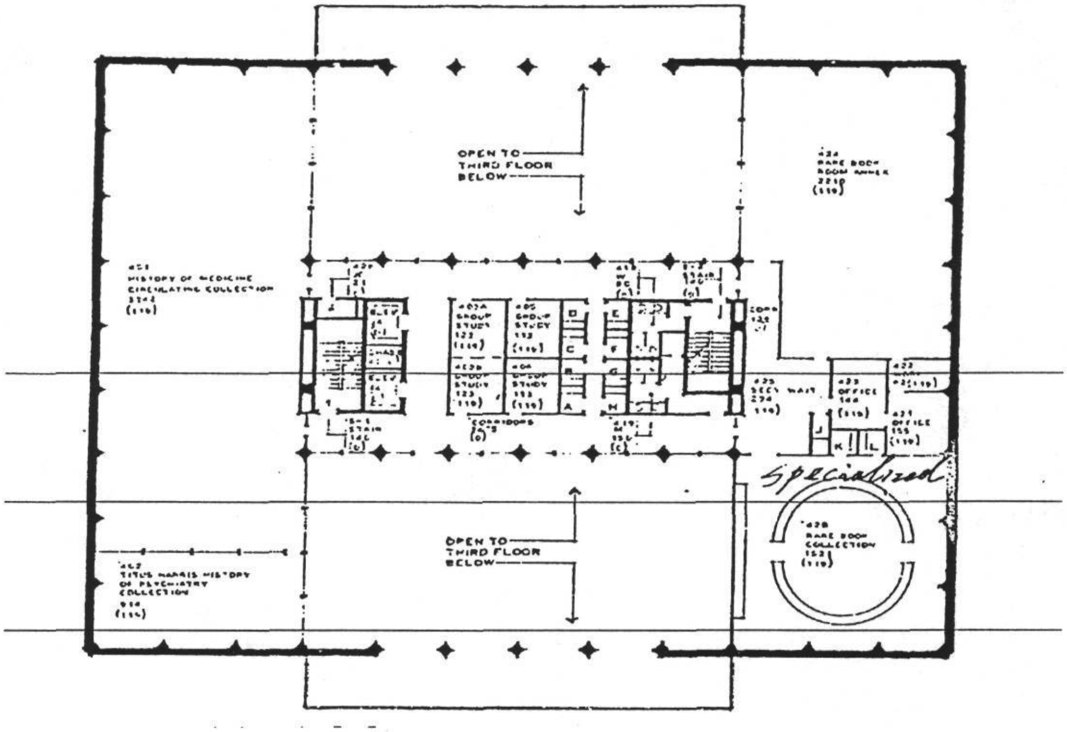

Figure 3: Typical Floor Plan in the Moody Library Building

The building envelope area is calculated as 28,000 ft^2, which includes 7,364 ft^2 window area. The walls are assigned a heat transfer coefficient value of 0.20 Btu/ft^2 °F hr . The windows are assigned a heat transfer coefficient value of 1.00 Btu/ft^2 °F.

Air infiltration rates are taken as 0.2 ACH (air change number of building volume in one hour) for the exterior zone and no infiltration for the interior zone.

3.2 Model Calibration

LoanSTAR measured 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 EMCS measured steam energy consumption is 11% higher than LoanSTAR measured consumption (See Appendix C). The chilled water consumption is not compared due to lack of the EMCS measured data.

The chilled water and steam energy consumption were predicted with the simplified model using the measured daily average temperature from July 14, 1992 to June 29, 1993. However, the chilled water and steam energy consumption are compared only for 109 days because of lack of complete 24 hours measured temperature for the rest of the days.

Figure 4 compares the measured energy consumption and the model predicted energy consumption. The horizontal axis is the ambient temperature while the vertical axis is the daily average chilled water and steam energy consumption. It shows that the model predicted chilled water and steam energy consumption fit well with the measured data. However, there are a few of days when the predicted consumption is substantially higher than the measured consumption.

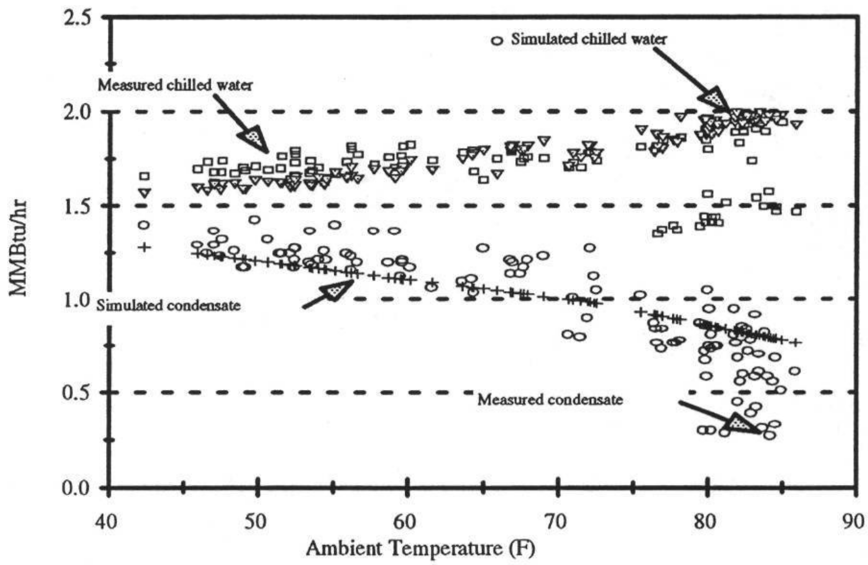

Figure 4: Comparison of Average Daily Energy Consumption Between the Model Predicted and Measured

Figure 5 compares the predicted and measured energy consumption using a time series chart. It shows that the simplified model captures daily variation very well. However, obvious differences between the measured and predicted energy consumption are observed from July, 1992 to September 20, 1992. These differences are very likely due to the changes of deck settings

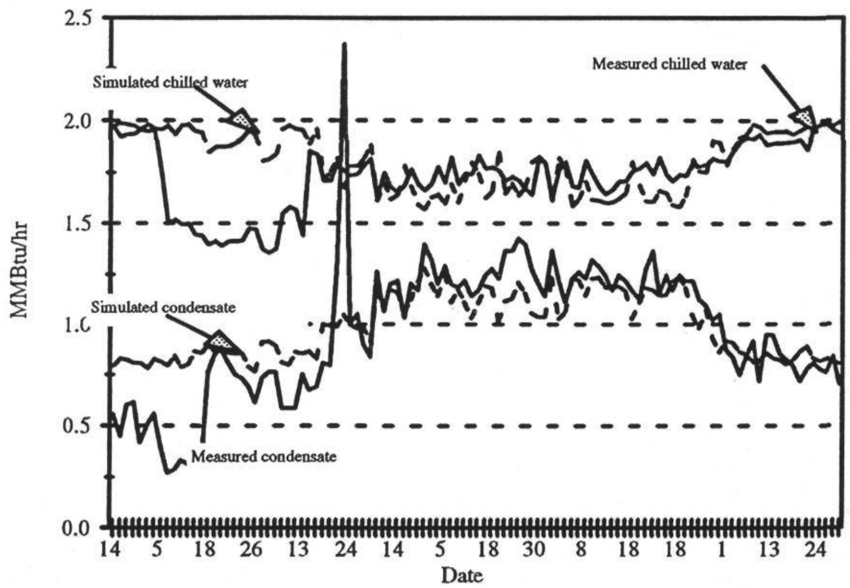

Figure 5: Comparisons of Daily Average Energy Consumption in time Series (From July 4, 1992 to June 29, 1993)

The simplified model is carefully examined by comparing measured consumption with predicted consumption from 10/24/92 to 06/29/93. There is a total of 73 days' data during this period. The measured daily average chilled water and steam energy consumption are 1.78 MMBtu/hr and 1.10 MMBtu/hr, respectively. The simulated daily average chilled water and steam energy consumption are 1.75 MMBtu/hr and 1.05 MMBtu/hr, respectively. Therefore, the model has an error less than 5% for the total energy prediction. The mean square root errors are 0.10 MMBtu/hr and 0.14 MMBtu/hr for predicted chilled water and steam energy consumption, respectively. The coefficients of variation are 6% and 13% for chilled water and steam consumption, respectively.

The calibrated simplified model was used to calculate annual energy consumption using bin data. 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, which are 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 the year that the bin temperature was measured. It was assumed that Galveston has the same weather condition as Houston.

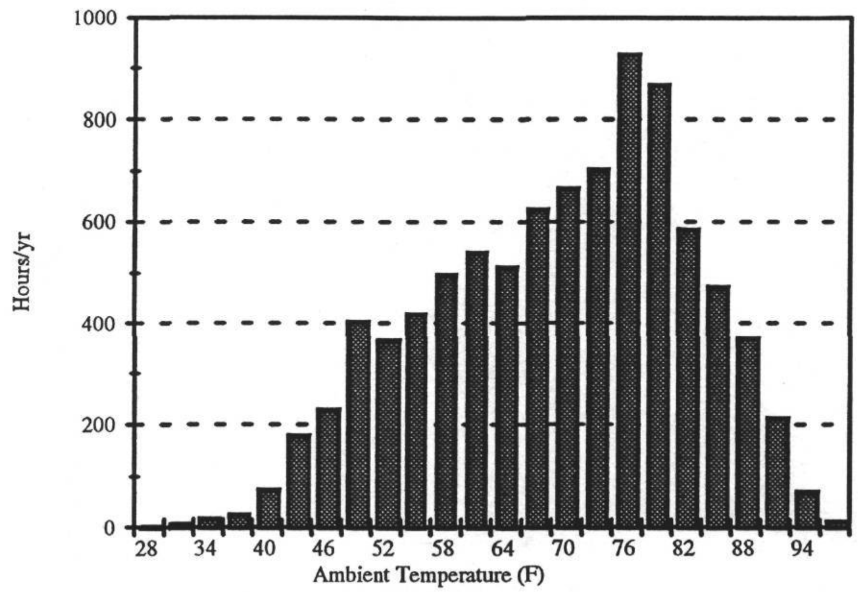

Figure 6: Houston Bin Temperature Chart Generated Using LoanSTAR Measured Hourly Temperature Data from July 1, 1992 to June 30, 1993.

The mean coincident dew point temperatures are plotted as a function of the ambient 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, then, remains more or less as constant at 70 °F 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.

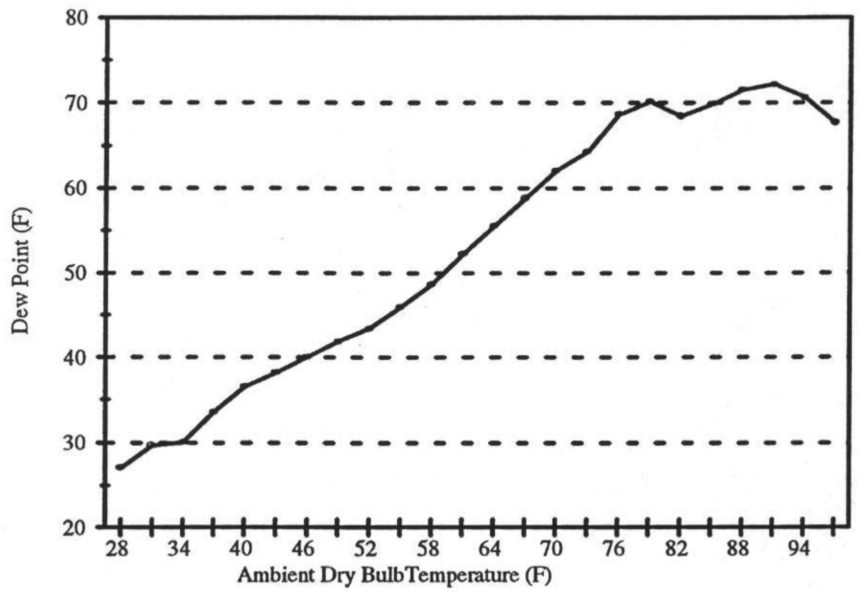

Figure 7: Mean Coincident Dew Point Temperature as a Function of Dry Bulb Temperature in Houston for July 1992 to June 1993

Table 2 summaries the values of the key parameters used in the calibrated simplified model and the baseline settings of the EMCS system. The calibrated return air fraction was determined from measured chilled water consumption (see Appendix B) which is substantially higher than the design value. The calibrated air flow rate was determined from the site measurement.

Table 2: Summary of the Model Calibration Parameter Adjustment

Item	Schedule (EMS)	Schedule (Model)
Supply air flow rate	86,015 (Blue prints)	73,113
Return air fraction	0.85 (Blue prints)	0.96
Cold deck temp.	56	56

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

Return air temperature	74	74
Room air temperature	72	72

4. OPTIMIZING SCHEDULES & ECONOMIZER CYCLE

The goal of optimizing cold deck and hot deck schedules is to minimize the energy consumption while maintaining comfort levels with minimum implementation costs.

The optimization process is currently an iterative process. A best operation schedule is first chosen. Then, energy and mechanical operation performances are predicted using the simplified model. The energy and mechanical performances are compared with the best so far obtained, modifications to the operation schedule are made and a new simulation performed. This process is repeated until the operation schedule is considered the optimal.

The base and the optimized operation schedules are listed in Table 3, and are also shown in Figure 8 versus the ambient temperature.

Figure 8 shows that the optimized schedule has the cold deck supply air temperature set higher than that of the base schedule settings. Obviously, these cold deck temperature increases can reduce chilled water and steam consumption substantially while the room relative humidity levels are maintained by using the partially closed coils.

Table 3: Comparison of Operation Schedules

Item	Base	Optimized
Cold deck	56 °F	Min(62, 60-0.2*(T0-85))

Hot deck	If T0<80 then
	Min(90, 80-0.25*(T0-75))
	Else
	80
		If T0<80 then
		Min(90, 80-0.25*(T0-80))
		Else
		70

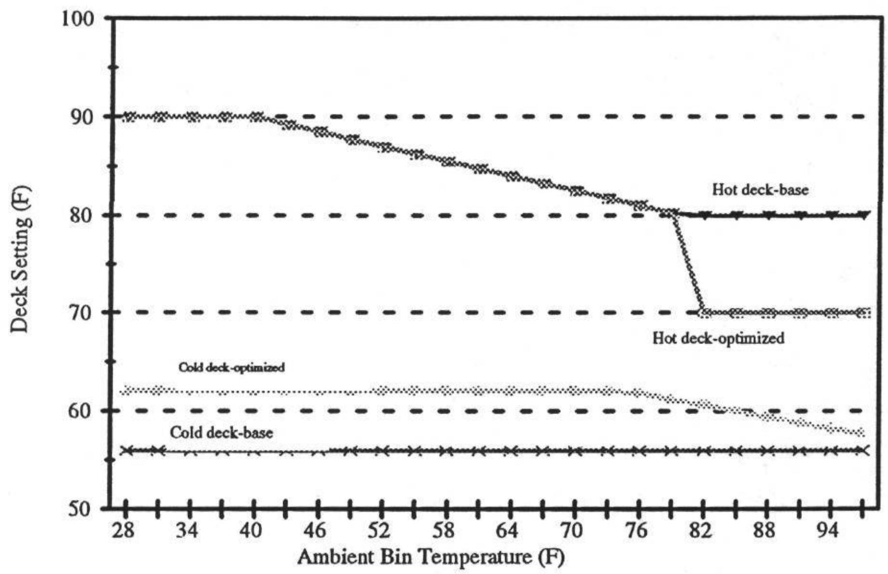

Figure 8: Base and Optimized Cold & Hot Deck Schedule

Figure 9 shows the schematic diagram of the cold deck which consists of three parallel coils. If an automatic valve is installed at the exit of coil 3, the optimized supply air temperature is maintained by mixing the warm air from coil 3 and the cold air from coils 1 and 2. Since coils 1 and 2 maintain low supply air temperature, the cold deck can remove a substantial amount of moisture. Consequently, the cold deck can provide high temperature supply air.

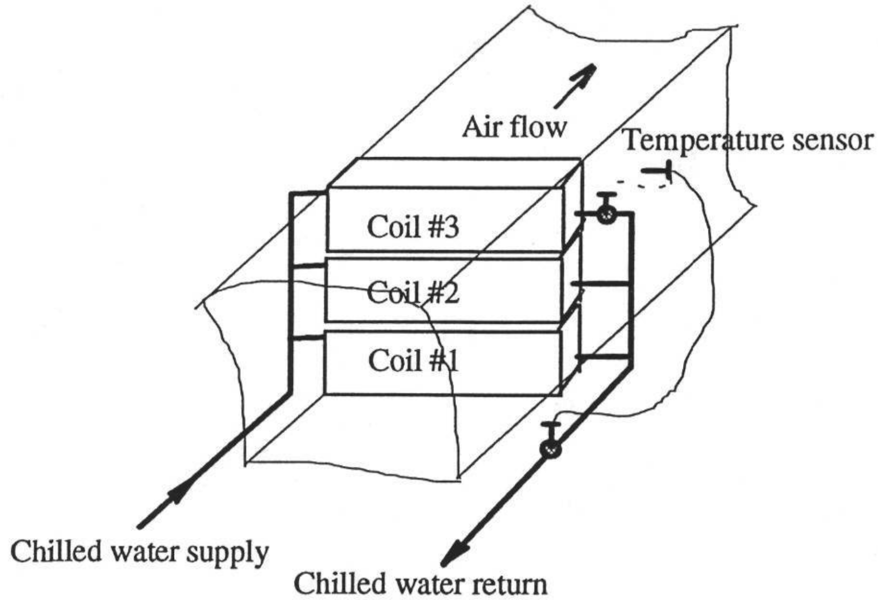

Figure 9: Schematic of Cold Deck Coil Connections

Figure 10 shows the supply air temperature from each coil. It is observed that coil 1 can be simply closed during winter when the ambient temperature is lower than 75 °F, and that the supply air temperature of coil 3 decreased with the increase of the ambient temperature when the ambient temperature is higher than 75 °F. Therefore, manual shut-off coil 3 is suggested for this winter.

The optimized schedule can be implemented by installing a temperature sensor for the supply air temperature of coil 3 and an automatic valve at the chilled water exit. The optimized schedule of coil 3 is maintained by the new installed sensor and valve. Then, the overall optimized schedule can be maintained by the existing sensor and valve for each AHU.

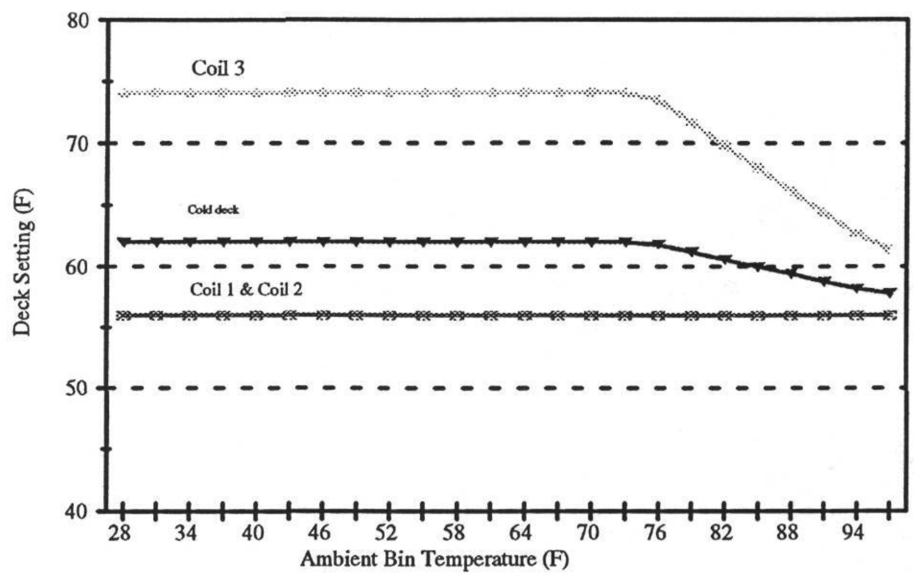

Figure 10: Supply Air Temperature for Each Coil Under Optimized Schedules

The potential savings of the economizer cycles was also investigated since it may be implemented with low costs. The economizer uses the following control strategy: if the ambient temperature is lower than cold deck supply air temperature, then the return air fraction is calculated by the formula:

$$\beta = \frac{T_c - T_o - \delta T}{T_r - T_o}$$

where β is the return air fraction, T_c is the cold deck supply air temperature, T_o is the ambient temperature, T_r is the return air temperature, and δT is the temperature rise due to supply air fan.

If the ambient temperature is higher than the cold supply air temperature but lower than a critical temperature (T_r -10°F), then the return air should be eliminated. If the ambient temperature is higher than the critical temperature, then return air fraction is taken as 0.96. Note that the critical temperature is chosen according to the relationship between ambient dry bulb and dew point temperature at Galveston. This critical temperature allows the temperature economizer to simulate enthalpy economizer.

Figure 11 shows the return air fraction versus the ambient temperature. The return air fraction reduces as the ambient temperature increases when the ambient temperature is lower than cold deck supply air temperature. Then it remains zero within a narrow temperature band and finally goes up to 0.96. The economizer has relatively higher return air fraction values under optimized cold deck operation schedule due to the increase of the supply air temperature.

In the next section, the energy performance and mechanical performance under the optimized operation schedule and economizer cycles are compared with those of the base performance.

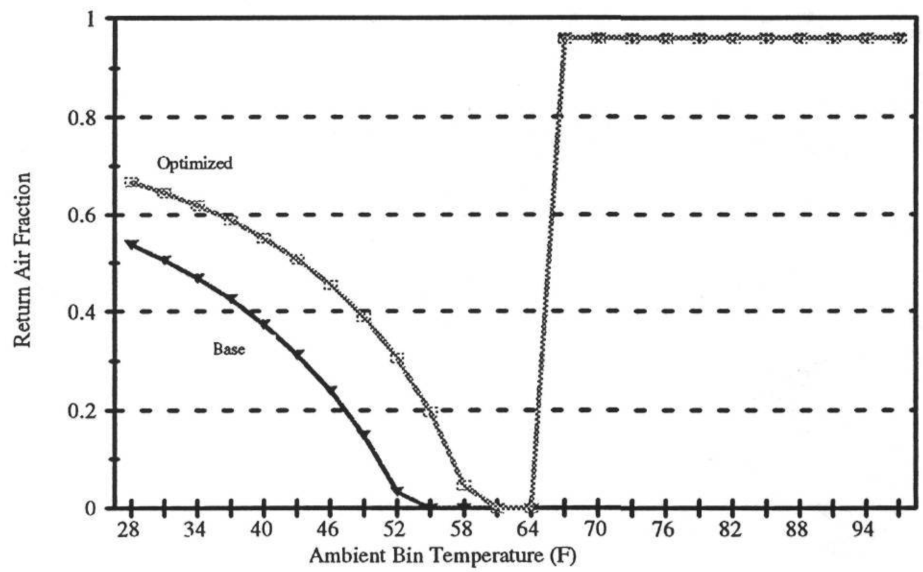

Figure 11: Return Air Fraction Under Both the Base and Optimized Schedule with Economizer Cycle

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 rates through cold and hot ducts under each bin temperature and its coincident dew point for the base, optimized schedules, economizer cycle, and combination of economizer and optimized schedule. The same calculations are also performed by assuming that the optimized schedule is implemented by resetting the cold deck temperature. The annual energy consumption is calculated by summing the products of energy consumption and the number of hours at each bin temperature. It should be pointed out here that the optimized schedule invalves implementing the option of partially closed coils, while the cold deck reset refers that the optimized schedule involves the option of increasing the cold deck temperature.

Figure 12 compares the chilled water consumption under different operation schedules. The horizontal axis is the ambient bin temperature. The vertical axis is the energy consumption MMBtu/hr for both the chilled water and the steam. It shows that the economizer cycle can eliminate chilled water consumption when ambient temperature is lower than 52 °F. The economizer also reduces chilled water consumption when the ambient temperature is within 52 °F to 67 °F. However, it does not change the chilled water consumption as soon as the ambient temperature is higher than 67 °F. The optimized schedule can reduce chilled water consumption by 0.5 MMBtu/hr when the ambient temperature is lower than 76 °F when coil 3 is closed. The reduction decreased as the ambient temperature increase further. The combination of optimized schedule and economizer can eliminate chilled water consumption when the ambient temperature is lower than 58 °F.

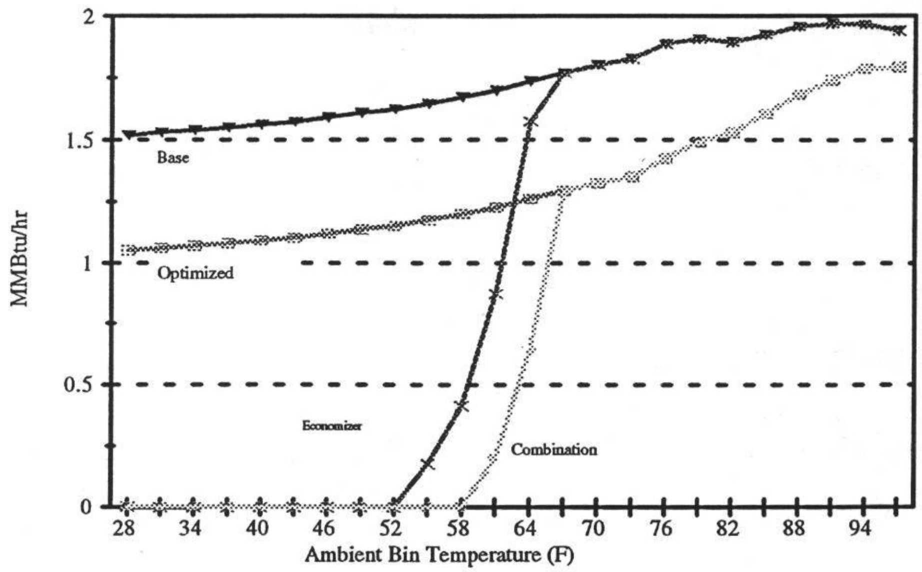

Figure 12: Comparison of the Predicted Chilled Water Energy Consumption Under the Different Operation Schedules

Figure 13 compares the steam energy consumption under the different operation schedules. The horizontal axis is the ambient bin temperature. The vertical axis is the steam energy consumption in unit of MMBtu/hr. It shows that the optimized schedule reduced steam energy consumption by 0.5 MMBtu/hr when the ambient temperature is lower than 76 °F. The reduction decreased as the ambient temperature increases further. The economizer cycle does not influence the steam energy consumption.

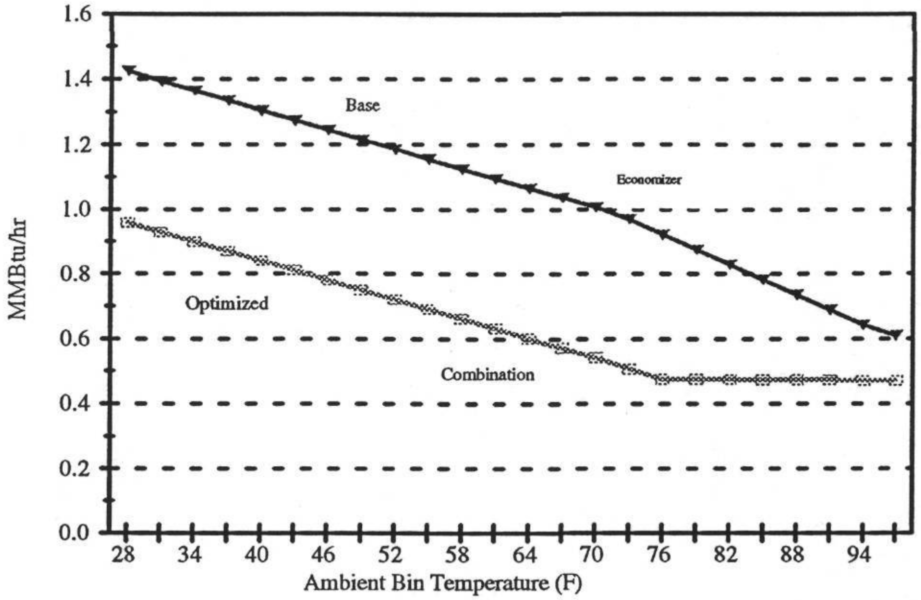

Figure 13: Comparison of the Predicted Steam Energy Consumption Under the Different Operation Schedules

Figure 14 compares the predicted room relative humidity under the different operation schedules. There are two lines for each operation schedule. One line is for the interior zone and another line is for the exterior zone. The exterior zone has slightly higher relative humidity levels than the interior zone. Since both the interior and the exterior zones have very similar humidity levels they are not indicated separately in Figure 14.

The predicted room relative humidity levels under the base schedule were consistent with the EMCS measured values. The optimized schedule can increase the room relative humidity to 50%, which is about 2% higher than the base schedule value. The economizer decreased the room relative humidity levels substantially when the ambient temperature is lower than 55 °F. Therefore, humidification is necessary during very cold days if the economizer is used.

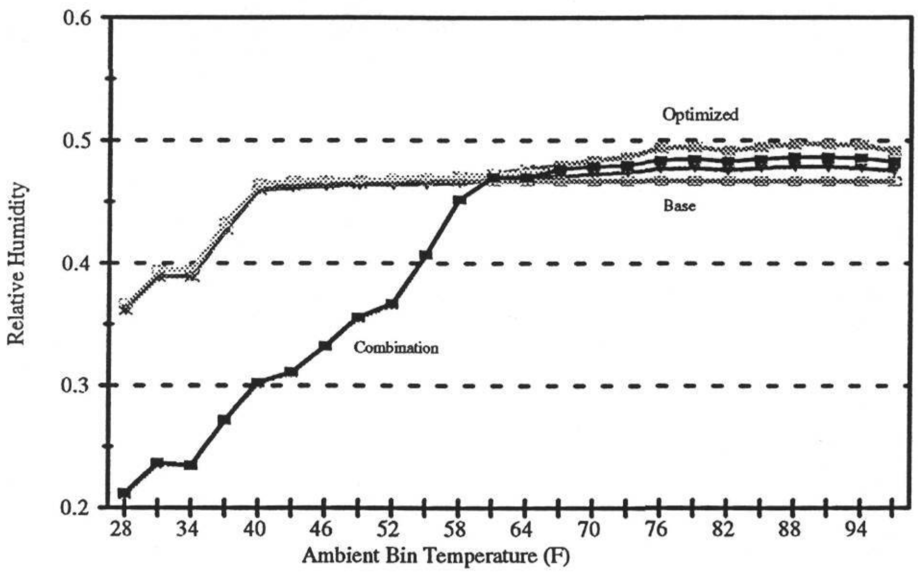

Figure 14: Comparison of the Predicted Room Relative Humidity under the Different Operation Schedules

Figure 15 compares the predicted air flow rates through cold and hot ducts under both the base and the optimized schedules. The base schedule has a cold duct flow range of 40,000 CFM to 60,000 CFM and a hot duct flow range of 15,000 to 35,000 CFM, while the optimized schedule has a cold duct flow range of 47,000 CFM to 60,000 CFM and a hot duct flow rate range of 12,000 to 25,000 CFM. Although the relationship of air flow rate with temperature changed significantly, the air flow ranges do not change under both the optimized and the base operation schedule. Consequently, the optimized schedule can be accommodated by the current duct system. Note that the economizer cycle does not affect the air flow rate through ducts.

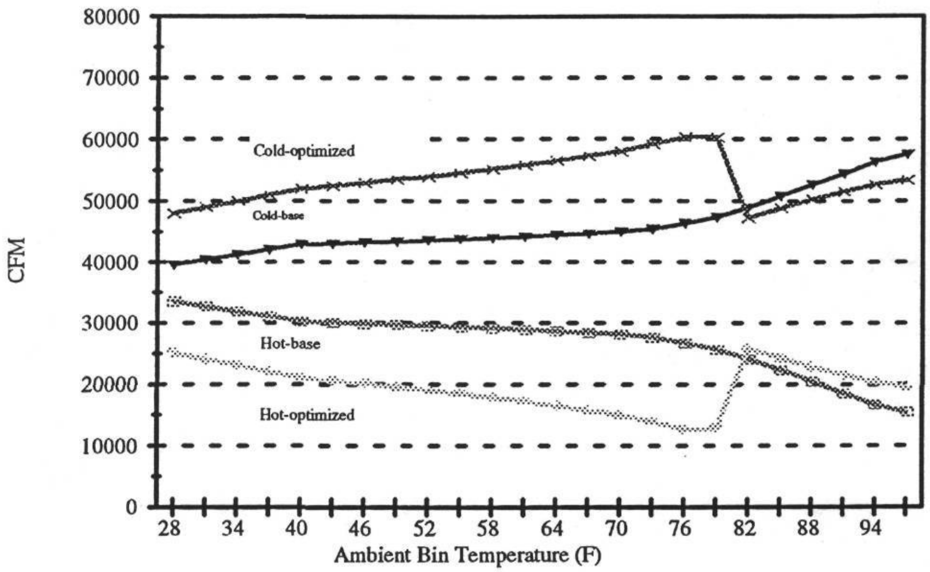

Figure 15: Comparison of Air Flow Rates Through the Cold Duct and the Hot Duct under Both the different Schedules

The potential annual energy savings was calculated by subtracting the optimized energy consumption from the base energy consumption. The results of the optimized schedule are shown in Figure 16. The horizontal axis is the ambient bin temperature and the vertical axis is the potential annual energy savings for each bin year. It shows that about the same amount of chilled water and steam energy can be saved. The small difference between chilled water and steam energy savings is due to the fact that the optimized schedule removes a little less moisture than that under the base operation schedules.

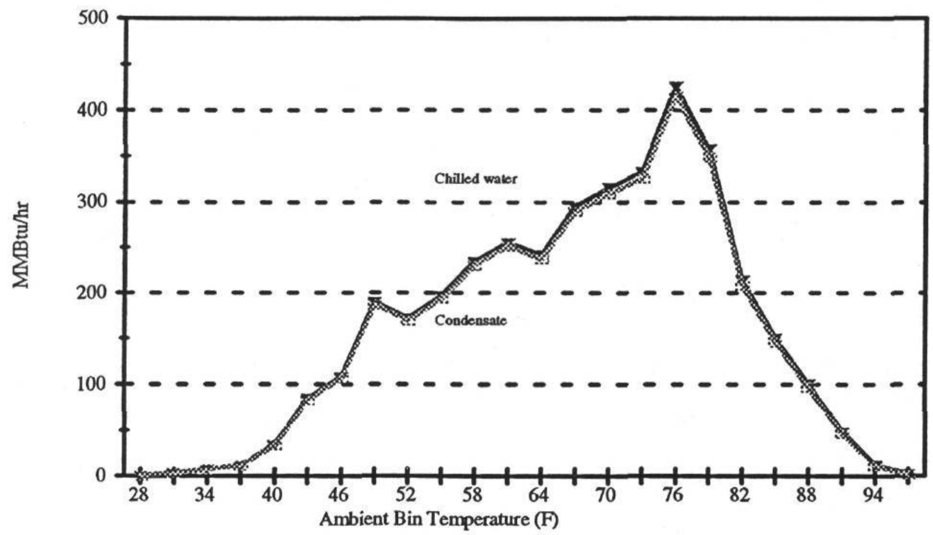

Figure 16: Predicted Potential Annual Chilled Water and the Steam Energy Savings under Optimized Operation Schedule

Figure 17 shows the potential chilled water and steam energy savings with implementation of both the optimized schedule and economizer cycles. It shows that the chilled water savings is about the two times of the steam energy savings due to optimal use of the outdoor air intake.

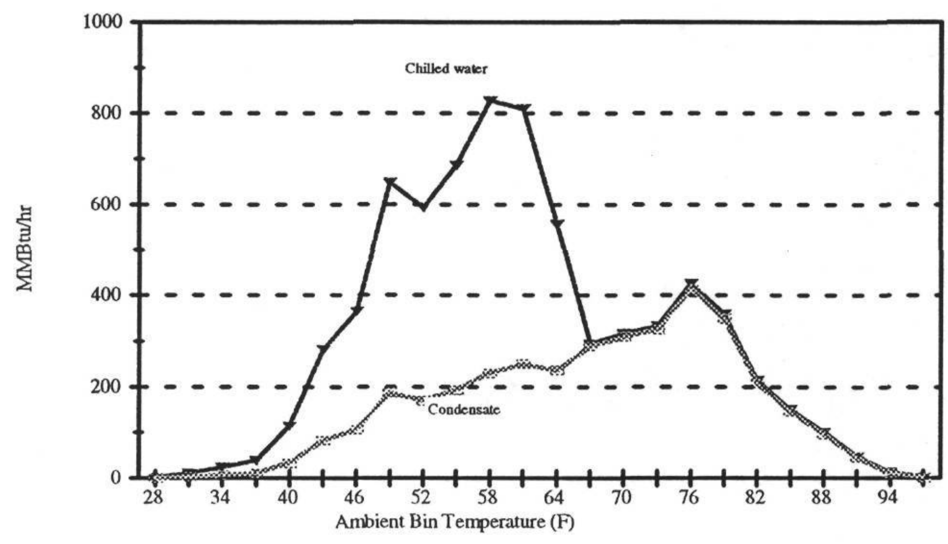

Figure 17: Predicted Potential Annual Chilled Water and Steam Energy Savings When Both the Optimized Schedule and the Economizer Cycle Are Implemented

The overall optimized energy performance and the potential savings are summarized in Table 4. It shows that the optimized schedules can reduce the annual chilled water consumption from 15,700 MMBtu to 11,900 MMBtu with a savings of 3,800 MMBtu/yr, reduce the annual steam energy consumption from 8,700 MMBtu to 5,000 MMBtu with a savings of 3,700 MMBtu/yr. These energy savings reduce the annual cost by $27,700 for chilled water and $18,800 for steam. The total potential savings is $46,500/yr, which is 24% of the annual building energy cost, or 29% of the chilled water and steam energy costs.

Table 5 shows that the economizer cycle can reduce the annual chilled water consumption from 15,700 MMBtu to 11,900 MMBtu with a savings of 3,800 MMBtu/yr, which reduces annual energy costs by $28,100 or 14% of the annual building energy cost, or 18% of the chilled water and steam energy costs. The combination of the optimized schedule and the economizer cycles can reduce the annual chilled water consumption from 15,700 MMBtu to 8,500 MMBtu with a savings of 7,200 MMBtu/yr, reduce the annual steam energy consumption from 8,700 MMBtu to 5,000 MMBtu with a savings of 3,700 MMBtu/yr. These energy savings reduce the annual cost by $52,700 for chilled water and $18,800 for steam. The total potential savings is $71,500/yr, which is 37% of the annual building energy cost, or 47% of the chilled water and steam energy costs.

Table 4: Summary of Potential O&M Savings at the Moody Library Building

No	Description	Consumption		Savings					
		MMBtu		MMBtu		Dollars		Total	
		Ch-Wate	Steam	Ch-Water	Steam	Ch-Water	Steam	Dollars	%
0	Base	15,700	8,700						
	Optimized	11,900	5,000	3,800	3,700	$27,700	$18,800	$46,500	24%
	Economizer	11,900	8,700	3,800	0	$28,100	0	$28,100	14%
1	Combination	8,500	5,000	7,200	3,700	$52,700	$18,800	$71,500	37%

Note:

The annual energy costs were $194,900, which includes $41,800 electricity costs (1992, Moody Library, LoanSTAR measured energy consumption data), $109,900 chilled water costs, and $43,200 steam costs. The chilled water and steam consumption were calculated using a simplified model which was calibrated using the measured chilled water and hot water.

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

Table 5 summarizes the energy indices of the Moody Library building based on gross floor area. The optimized schedules can reduce chilled water consumption for unit floor area from 0.233 MMBtu/ft^2-yr to 0.176 MMBtu/ft^2-yr and reduce steam energy index from 0.129 MMBtu/ft^2-yr to 0.074 MMBtu/ft^2-yr. The potential chilled water and steam combination savings are \$0.69/ft^2-yr for the optimized schedule. The economizer cycles can reduce chilled water consumption per unit floor area by 0.056 MMBtu/ft^2-yr and reduce cost index by \$0.42/ft^{2}yr. The combination of optimized schedule and economizer cycles can reduce chilled water consumption for unit floor area from 0.233 MMBtu/ft^2-yr to 0.126 MMBtu/ft^2-yr and reduce steam energy index from 0.129 MMBtu/ft^2-yr to 0.074 MMBtu/ft^2-yr. The potential chilled water and steam combination savings are \$1.06/ft^2-yr for the optimized schedule.

Table 5: Summary of Thermal Energy Indices

Item	Ch-water (MMBtu/ft^2 yr)	Steam (MMBtu/ft^2 yr)	Savings (MMBtu/ft^2 yr)		Savings (\$/ft^2 yr)
Base	0.233	0.129	Ch-water	Steam	Total
Optimized	0.176	0.074	0.056	0.055	\$0.69
Economizer	0.176	0.129	0.056	0	\$0.42
Combination	0.126	0.074	0.107	0.055	\$1.06

The optimized schedules should only be implemented by using partially closed cold deck coils to maintain the room relative humidity levels. Figure 18 shows the predicted room relative humidity levels under the base schedules, the optimized schedules, and the cold deck resetting. Cold deck resetting can increase the room relative levels by 10% to 57% while the optimized schedule only increases the room relative humidity by 2%. Due to the strict limit on the room relative humidity levels in the building, cold deck resetting cannot be used. This difference is caused by the different working processes of the cold decks, which are explained in Figure 19 with an example.

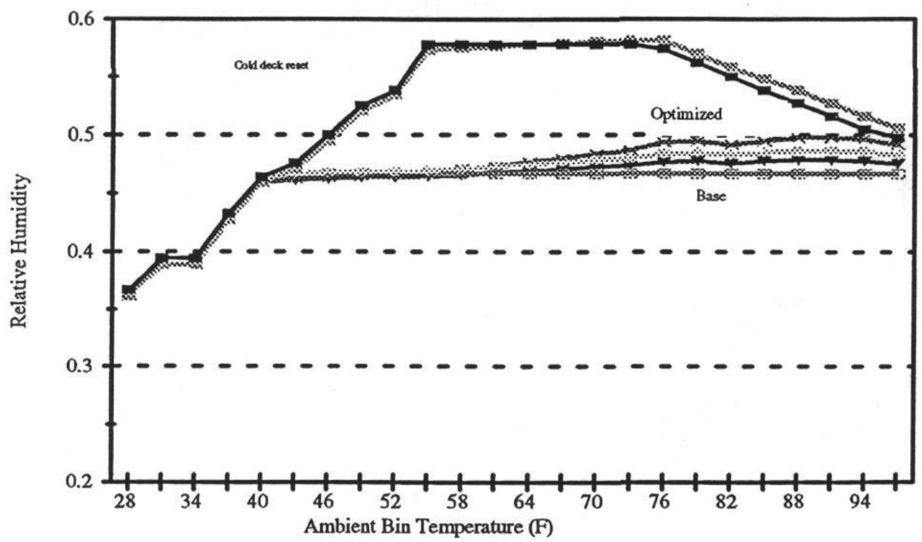

Figure 18: Comparison of Humidity Levels Under Different Schedules

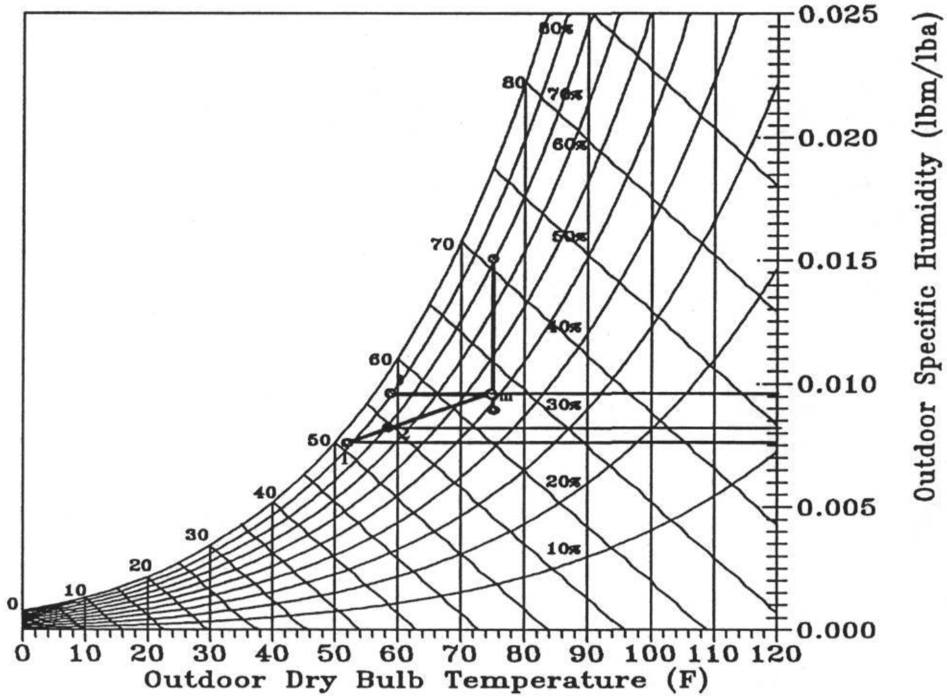

Figure 19: Working Process of the AHU Under Different Schedules

The working processes of the cold deck are plotted in a psychometric chart for the base schedules, the optimized schedules, and the cold deck resetting schedules under the following conditions: outdoor air, 75 °F and 80% (RH); return air, 74 °F and 50% (RH); and the return air fraction 0.96. The supply air temperature (after the cold deck) is 59 °F under the optimized schedule and cold deck reset while it is 52 °F under the base schedule. The mixing air condition is marked as "m", and the supply air conditions are marked as "1", "2" and "3" for the base, optimized, and the cold deck reset, respectively. The base schedule cools all the supply air to 52 °F. The optimized schedule cools 2/3 of the air to 52 °F using coils 1 and 2 and keeps 1/3 of air at the entrance condition by closing coil 3. The cold deck reset cools all the supply air to 59 °F. The basic parameters of supply air and the mixing air are summarized in Table 6. It shows that the optimized schedule has 70% of the dehumidification capacity of the base schedule while it consumes 29% less chilled water than the base schedule. Although cold deck resetting consumes 49% less chilled water than the base schedule, it can remove no moisture. Consequently, the room relative humidity levels are increased.

Table 6: Summary of the Supply Air Conditions Under Different Schedules

	Base	Optimized	Resetting	Inlet
Temperature (°F)	52	59	59	74.1
Specific humidity (lb/lb)	0.0076	0.0082	0.0096	0.0096
Enthalpy (Btu/lb)	20.6	23.0	24.6	28.7
Moisture removed (lb/lb)	0.0020 (1)	0.0014 (0.70)	0.000	
Energy consumption (Btu/lb)	8.1 (1)	5.7 (0.71)	4.1 (0.51)	

It should be pointed out that the simplified model analysis did not investigate the potential savings of nighttime setback. However, it is suggested that nighttime setback 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 operators.

6. CONCLUSIONS

The annual building energy costs can be reduced by $46,500/yr (24% of the current annual energy costs) using the optimized schedules, which can be implemented by installing an automatic valve at the chilled water exit of the top coil and a supply air temperature sensor for this coil. The economizer can reduce the annual energy cost by $28,100 (14% of the total current energy cost). The combination of the optimized schedule and economizer can reduce the building energy cost by $71,500/yr (37% of the building energy cost).

Although the cold deck reset, or increasing cold deck temperature, can maintain the room temperature levels with less than 2/3 of the current chilled water and steam consumption, it cannot maintain the room relative humidity levels. The cold deck coils have to be partially closed to reduce the energy cost and maintain the room relative humidity levels without major retrofits.

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2. 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.

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APPENDIX A: SIMPLIFIED MODEL

The schematic of air handling units (AHU) and the building is shown in Figure A1, where the two AHUs are treated as one and the building is idealized as two zones: an interior zone and an exterior zone.

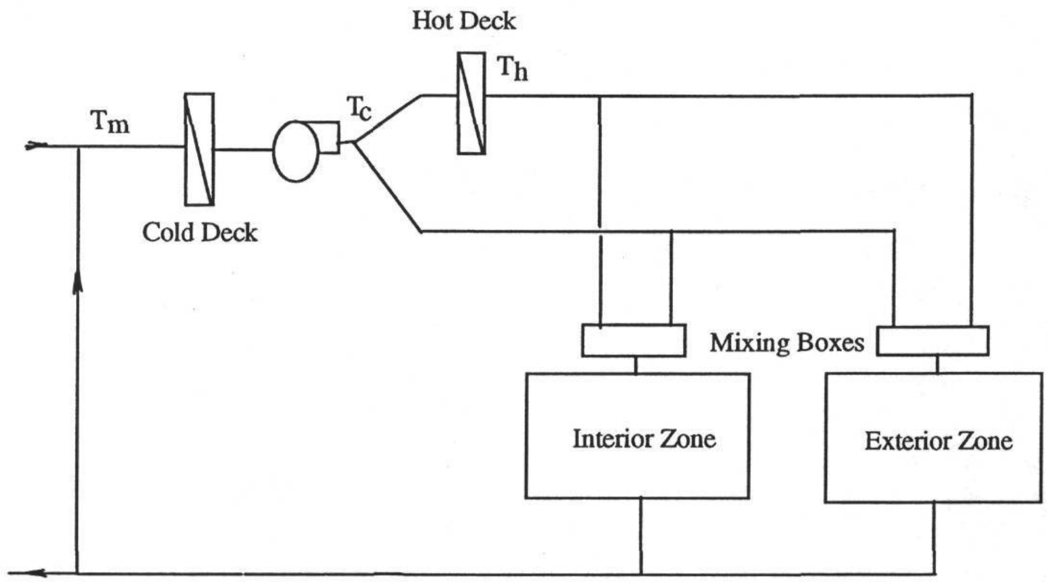

Figure A1: Schematic Chart of Air Handling Units for Moody Library Building

The chilled water consumption is calculated by the formula:

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

where E_c is the chilled water energy consumption, $\dot{m}$ is the total supply air mass flow rate, h_m is the air's specific enthalpy at the entrance of the cold deck, h_c is the cold deck supply air's specific enthalpy, and E_{fan} is the fan's power consumption.

The steam energy consumption is calculated by the formula:

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

where E_h is the steam energy consumption, $\dot{m}_h$ is the hot deck's air flow rate, T_c is the cold deck's supply air temperature, T_h is the hot deck's supply air temperature, and C_p is the air specific heat.

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

$$h_m = f_o \times h_o + (1 - f_o) \times h_r$$

$$T_m = f_o \times T_o + (1 - f_o) \times T_r$$

where h_r and T_r are the return air's specific enthalpy and temperature, respectively, E_{fan} is the supply air fan's energy consumption, h_o and T_o are the outdoor air's enthalpy and temperature, respectively, and f_o is the outdoor air intake fraction.

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 if these boxes are not changed. 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 $\dot{m}_{ext}$ and $\dot{m}_{int}$ are the air flow rates to the exterior and interior zones respectively, A_{ext} and A_{int} are the conditioned floor areas in the exterior and interior zones respectively, and A is the total conditioned area.

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

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

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

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

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

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

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

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

The room air's 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's 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.

APPENDIX B: RETURN AIR FRACTION

We tried to measure the return air fraction during our visit on July 15. However, it was found that accurate measurement was not possible. Our impression was that the return air flow was very low, substantially lower than 15% of the total supply air, which was the design value.

The return air fraction is then determined using the measured chilled water consumption data. The chilled water consumption can be expressed as:

$$E_c = \dot{m}\{\beta h_r + (1-\beta)h_o - h_c\}$$

where E_c is the measured chilled water consumption, β is the return air fraction, M is the total supply air mass flow rate, h_o is the outdoor air specific enthalpy, and the h_c is the supply air specific enthalpy. The return air fraction can be determined if two sets of data are available.

Figure B1 shows the measured chilled water energy consumption versus the ambient temperature. Two typical points were chosen from this chart. The parameters are summarized in the following table.

Summary: Typical Energy Performance Data of Cold Deck

	T_o °F	Relative humidity	Enthalpy (Btu/lb)	E_c (MMBtu/hr)
1	80	85%	40	1.9
2	60	90%	25.4	1.75

The cold deck has a temperature of 52 °F at the exit of the coil. If 90% relative humidity is assumed at the exit of the coils, the supply air enthalpy is 20.5 Btu/lb. Finally, the return air fraction is determined as 0.96 using these data.

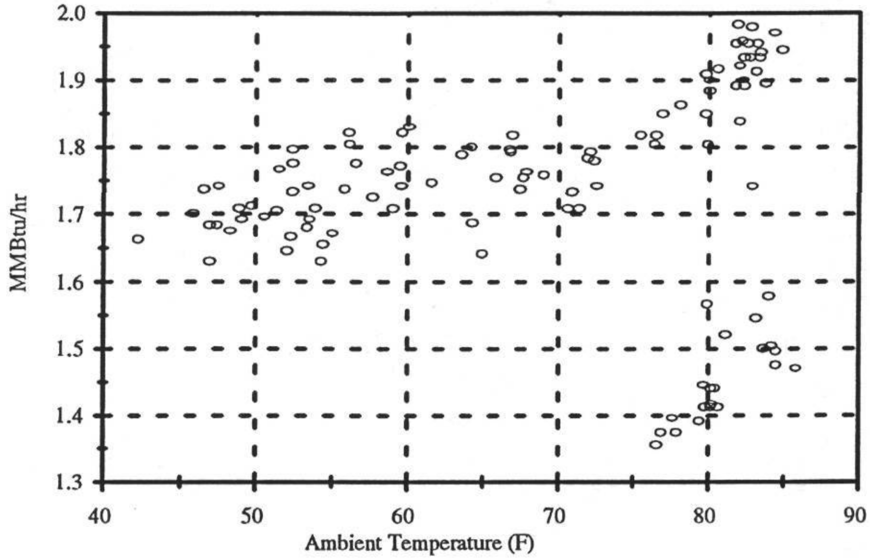

Figure B1: Measured Chilled Water Consumption at Moody Library Building

APPENDIX C: DATA QUALITY CHECK

The steam consumption is deduced by measuring the number of pumps on. This number is then converted to GPH by using a multiplying factor of 45.79, which was determined by comparing pump run time and the GPH measured by the EMCS during a short term test.

The GPH value is converted to MMBtu/hr by a multiplying factor of 0.0078653 (8.667*0.9075/1000). Note the water density is taken as 8.667 lb/gal according to the EMCS program at UTMB, and the latent heat is taken as 0.9075 kBtu/lb according to LoanSTAR. UTMB uses 1.064 kBtu/lb as the latent heat in its EMCS program.

Figure C1 compares LoanSTAR measured daily average steam energy consumption with EMCS measured data from June 16 to July 14, 1993. Figure C1 shows that EMCS measured consumption is about 3% to 14% higher than LoanSTAR measured data due to the different specific latent heat values used by EMCS and LoanSTAR. After removing this influence, EMCS measured data agrees with LoanSTAR measured data within 5%.

Figure C2 compares LoanSTAR measured monthly steam energy consumption with EMCS measured data. It shows again the difference identified in the daily energy consumption chart.

EMCS does not have complete measured data due to computer down time. The down time is corrected using the following formula:

$MMBtu_{EMS1} = MMBtu_{EMS} \times$(Days of the month)/(Days of month - Downtime of month)

Not that EMCS chilled water data is not available.

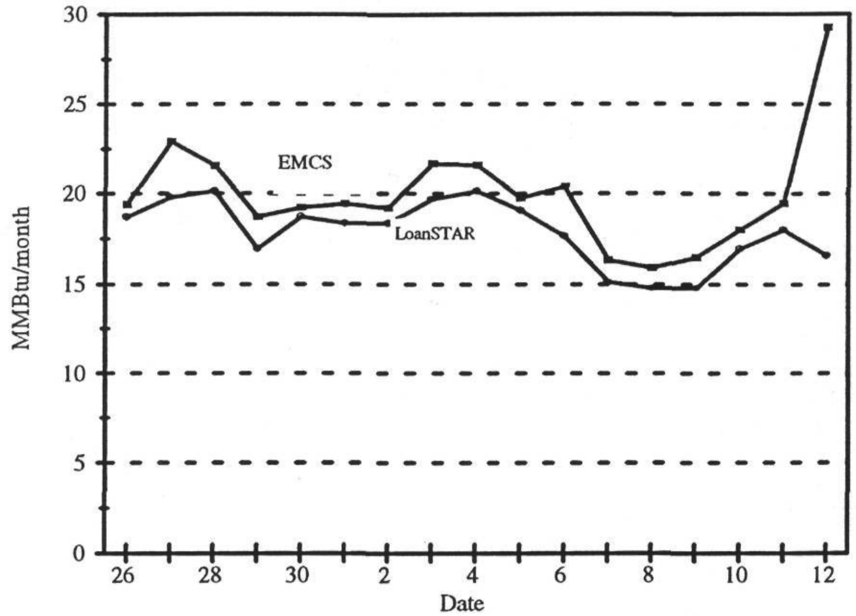

Figure C1: Comparison of LoanSTAR and EMCS measured Daily Average Steam Consumption from June 16 to July 14, 1993

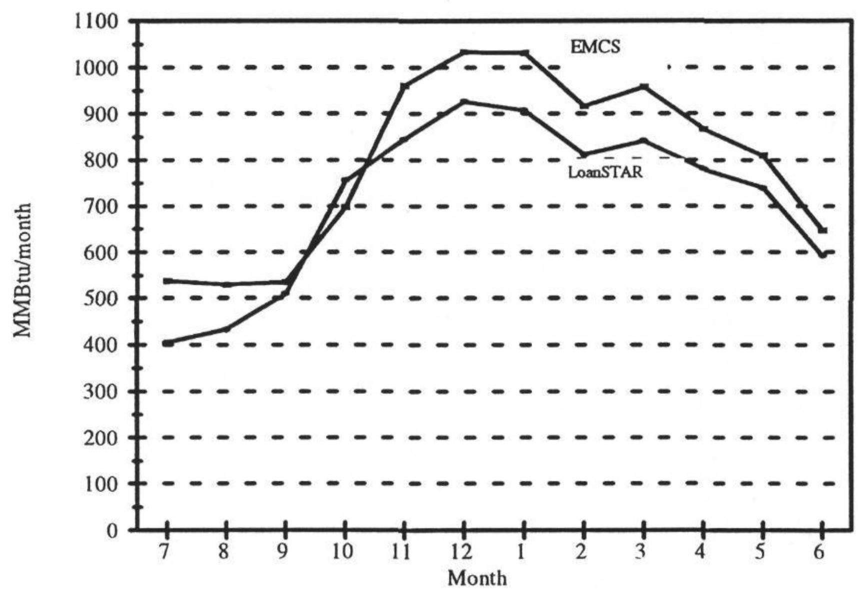

Figure C2: Comparison of LoanSTAR and EMCS Measured Monthly Steam Consumption from September 1992 to August 1993