Evaluation of Energy Conservation Measures by Model Simulation

Tim Giebler, Mingsheng Liu, and David Claridge
Energy Systems Laboratory
Texas A&M University

Abstract

Numerous energy conservation measures are being implemented into the air handler units of today's commercial buildings. The economizer cycle has proven potential, and has become increasingly more common. Work has also been done demonstrating that hot and cold deck reset schedules, optimized according to outside air temperature, can result in significant energy savings. This paper presents a case study of these energy conservation control schemes in a dual duct VAV building on the Texas A&M campus in College Station, the Harrington Education Tower. The current system was simulated and the model used to investigate the effects of economizer cycles and optimization of the hot and cold deck reset schedules.

Introduction

There is a growing necessity to design energy efficiency and conservation measures into new commercial building systems and also retro-fitting them into existing buildings. The economizer cycle has become a recognized and popular measure used in building air handler units (AHU). The temperature economizer minimizes the AHU mechanical cooling by controlling the outside air flow rate and using it to cool the mixed air to the cold deck set point, if possible. The temperature economizer operates between a protective low temperature limit and the change point temperature. The change point temperature should be at least a few degrees lower than the return air temperature. The mechanical cooling is eliminated when the outside air temperature is below the cold deck set point and within the economizer operating range. The temperature economizer can continue to reduce the amount of mechanical cooling when the outside air temperature is above the cold air discharge temperature and below the change point temperature, by using maximum outside air [1]. Above the change point, as well as below the low temperature set point, the economizer is disabled and minimum outside air is used. There is also an enthalpy economizer, which works the same way, only the outside air intake is determined by air enthalpy rather than temperature [2]. An additional sensor, measuring dew point temperature or relative humidity, is required for enthalpy based control. Economizers can reduce the cooling energy significantly. However, the heating penalty may be higher than the cooling savings [5, 7, 8]. Therefore, a careful analysis should be performed before its installation.

Optimized hot and cold deck reset schedules are another energy conservation measure. Many dual duct VAV systems in operation today only reset the hot deck according to outside air temperature, leaving the cold deck set point constant. Hot and cold deck reset schedules optimized according to outside air temperature have been studied and documented by Liu et al [3, 4]. Knowledge of outside air dew point temperature or relative humidity can further improve the operation schedule.

In this paper, computer simulation was used to evaluate the economizer cycle and optimized hot and cold deck reset schedule energy conservation measures. The case building was modeled using AirModel, a steady state method simulation program first written in 1993 at the Energy Systems Laboratory, Texas A&M University [6]. Given hourly outside air temperature, outside air relative humidity, measured heating energy consumption, and measured cooling energy consumption, AirModel simulated the building based on the building characteristics entered into the input file. Upon development of an accurate baseline model, the aforementioned energy conservation measures were each included into the model separately, with the results to be discussed.

Building Description

The building simulated in the case study is Harrington Education Tower (see Figure 1), at Texas A&M University, College Station. It is an eight story education building, and consists primarily of offices and other meeting rooms.

It has a basement as well.

Figure 1. Harrington Tower on Texas A&M

There are an estimated 400 occupants, and the occupancy schedule is essentially 8 a.m. to 5 p.m., Monday through Friday. The overall building dimensions are 124 feet by 136 feet by 110 feet high. The first and second floor are 90 feet wide by 102 feet deep, and the eighth floor is 114 feet wide by 81 feet deep. There is 19,000 square feet of glazing, with 50% of it being on the first, second, and eighth floors, as they are predominantly glass sided. Harrington Tower receives hot and chilled water from the Texas A&M Physical Plant for its HVAC systems. The building received retrofit in 1995, from DDCAV system with pre-treated outside air to DDVAV with temperature economizer. The sole DDVAV air handler unit (see Figure 2) is housed in the basement, with a 200 hp motor producing up to 138,000 cfm of air for the second through eighth floors, as well as portions of the basement and first floor On the upper floors, the hot and cold supply ducts run through a central chase to ducting and then VAV terminal boxes. The first floor has three small constant volume single duct systems to meet its primary heating and cooling requirements. The control program for the VAV system has nighttime setback of temperature and air flow.

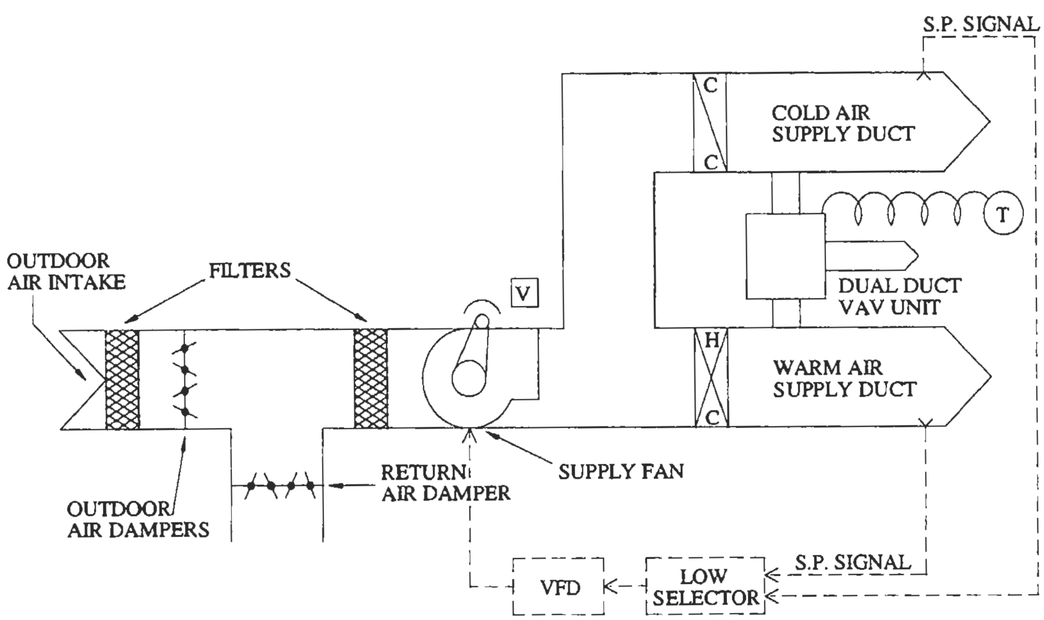

Figure 2. General HVAC layout of Harrington Tower DDVAV system

Model Calibration

The building parameters must be entered into the AirModel input file in order in begin a simulation. These parameters include the type of system, number of air handler units, floor area, wall area, window area, interior area ratio, and others. Since Harrington Tower consists of two different types of HVAC systems, it was grouped into two "buildings" for simulation purposes, where each different system is treated as a "building". Table 1 presents the information for each system. System I is the primary dual duct variable volume system, while System II consists of the three small constant volume units. The following assumptions were also made:

- Room temperature of 73°F in the summer and 70°F in the winter,
- Unoccupied setback of 75°F in the summer and none in the winter,
- An average floor area of 200 ft^2 per person and,
- A 2°F difference between room air temperature and the return air temperature.

The total and outside air flow rates for System II are also assumptions inferred from the blueprints, but not very critical as they add only a minor contribution to the overall building system.

It was necessary to calibrate the model, closely fitting the simulated hot water and chilled water consumption to the measured consumption data. This was a trial and error process, varying some key parameters and re-simulating. Table 2 shows the some of the final input values for the simulation. The most noticeable difference between Tables 1 and 2 would be the area served by System I. The initial value of 94,500 ft^2 was a calculation of interior area, minus the elevators, stairs, and central chase. When an effort was made to factor in wall thickness, utility closets, and other unusable space, the adjusted value of 73,000 ft^2, seen in Table 2, becomes very reasonable. Figures 3 and 4 display the calibrated simulation of the hot and chilled water consumption with the measured energy data.

Table 1. Key Input for the Building Simulation

System	Area	Interior Ratio	Wall Area	Window Area	Supply Air	Outside Air
I	94,500 ft^2	0.6	36,380 ft^2	14,675 ft^2	0.35 cfm/ft^2 min.	0.1 cfm/ft^2 min.
II	7,000 ft^2	0.5	512 ft^2	4,334 ft^2	1.4 cfm/ft^2	0.2 cfm/ft^2

Table 2. Key Final Input for the Building Simulation

System	Area	Interior Ratio	Wall Area	Window Area	Supply Air	Outside Air
I	73,000 ft^2	0.7	33,342 ft^2	16,500 ft^2	0.35 cfm/ft^2 min.	0.1 cfm/ft^2 min.
II	6,000 ft^2	0.5	500 ft^2	4,000 ft^2	1.35 cfm/ft^2	0.25 cfm/ft^2

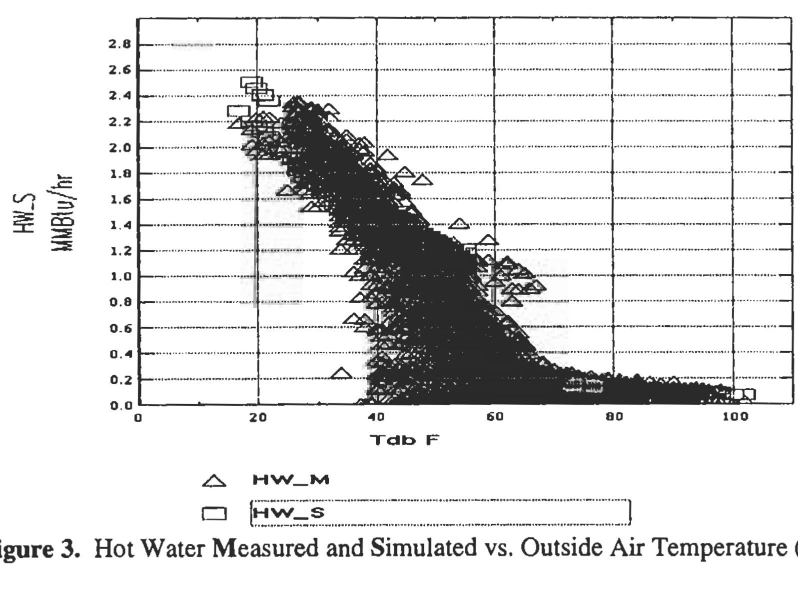

Figure 3. Hot Water Measured and Simulated vs. Outside Air Temperature (°F)

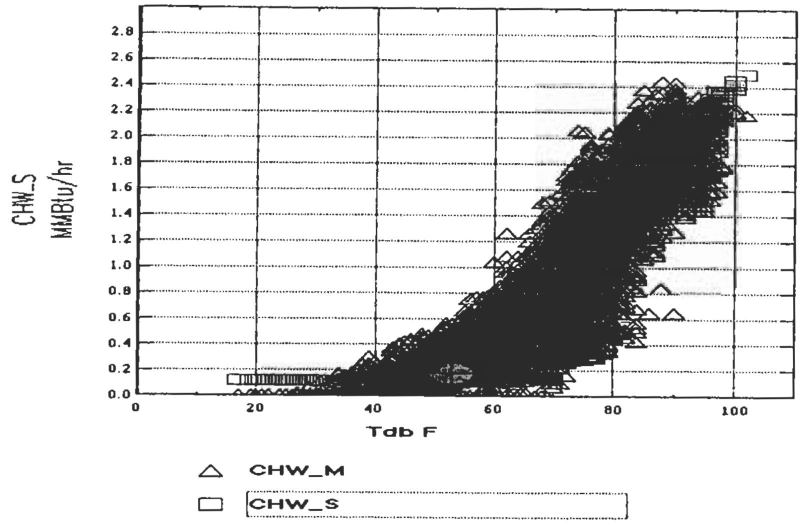

Figure 4. Chilled Water Measured and Simulated vs. Outside Air Temperature (°F)

The energy measurement and weather data used for calibration was from November 1, 1996 to November 1, 1997. There was 8,444 hours where outside air temperature, outside air relative humidity, measured heating energy consumption, and measured cooling consumption corresponded and were usable by the AirModel program. As can be seen, the simulated heating does not match the measured hot water usage well in the low temperature region. A large amount of time was spent trying to get a better fit, but all improvements adversely affected the simulated chilled water energy profile. There is good reason to believe the temperature economizer is not operating as ideal. This can be seen in Figure 4, as the cooling consumption dropped at 55°F outside air temperature and remained constant for all lower outdoor temperatures. In order to match the measured cooling consumption, the economizer was allowed to continually operate below the 37°F temperature limit specified in the HVAC control program. This will have a negative consequence on the heating requirements at these low temperatures, and the operational inconsistency may be difficult to simulate accurately.

Simulation

The building performance can now be accurately determined using the calibrated model. The potential benefits of economizers and optimization can be investigated using this model. To examine the impact of the temperature and enthalpy economizers, the baseline hot and cold deck reset schedule was replaced with a more standard one. This change, as well as the operating range of the economizers, can be seen below in Table 3. Three simulations were performed using the same time period as the calibrated model. For comparison, the temperature and enthalpy simulations results were each plotted along with the simulation run with no economizer versus outside air temperature. Throughout all simulation runs, the fuel costs of $0.03483 per kWh, $3.50 per MMBtu/h of heating energy, and $3.00 per MMBtu/h of cooling energy were used since the energy conservation effort was not expected to reduce the fixed costs, such as maintenance and materials, in this building. The simulation results can be seen on the following page.

Table 3. HVAC System Economizer Operating Ranges and Hot and Cold Deck Set Points

Dual Duct VAV System Simulated	Economizer Operating Range		Hot Deck Set Point		Cold Deck Set Point	
	Lower Limit	Upper Limit	40°F ambient	70°F ambient	40°F ambient	70°F ambient
Calibrated Model	7°F	55°F	98°F	70°F	60°F	54°F
Base, No Economizer	N/A	N/A	110°F	80°F	55°F	55°F
Temperature Economizer	40°F	55°F	110°F	80°F	55°F	55°F
Enthalpy Economizer	40°F	70°F	110°F	80°F	55°F	55°F

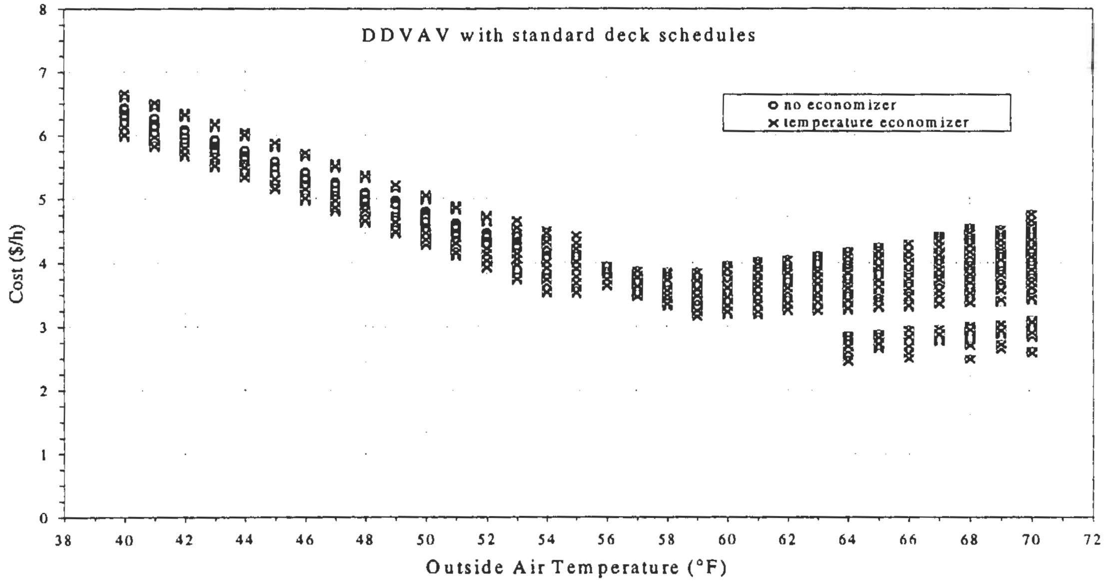

Figure 5. HVAC Operation Cost vs. Outside Air Temperature for no Economizer and Temperature Economizer Case

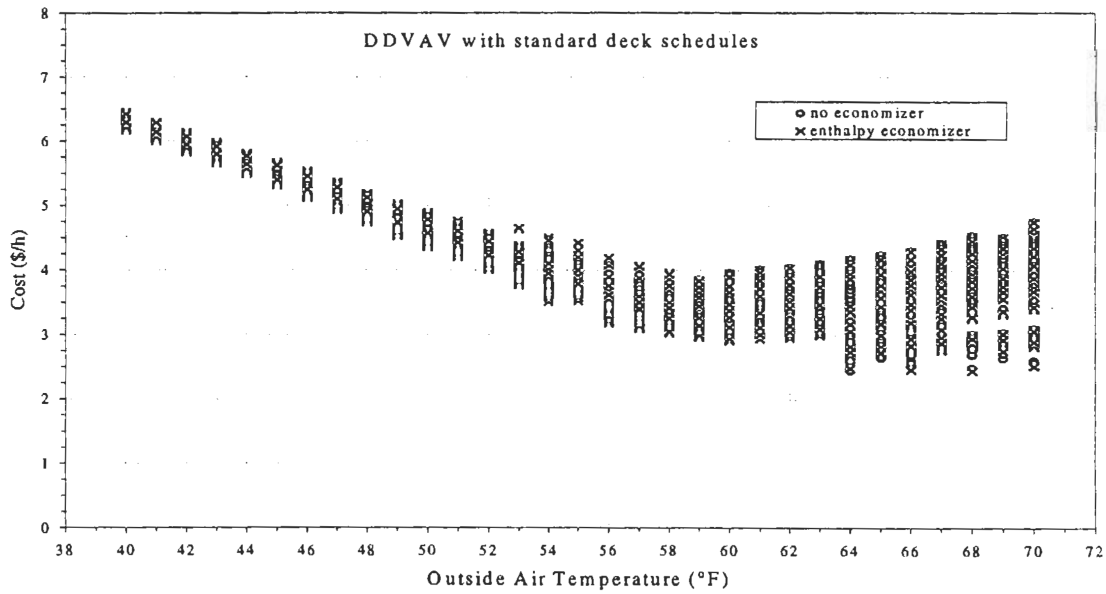

Figure 6. HVAC Operation Cost vs. Outside Air Temperature for no Economizer and Enthalpy Economizer Case

Only weather data in the 40°F to 70°F range was considered for simulation. This is because all other data would be outside of the operating ranges of the economizers, producing the same results as a normal DDVAV system. There were 3,708 hours of 40°F to 70°F temperature and corresponding relative humidity data within the November 1, 1996 to November 1, 1997 time period that were able to be used by the AirModel program. Optimization of the hot and cold deck schedules was also investigated using the same 3,708 hours of weather data. From Figures 5 and 6, there seems to be very little savings potential in this application. The optimization results for a DDVAV system operation cost plotted versus outside air temperature is displayed in the figure on the following page.

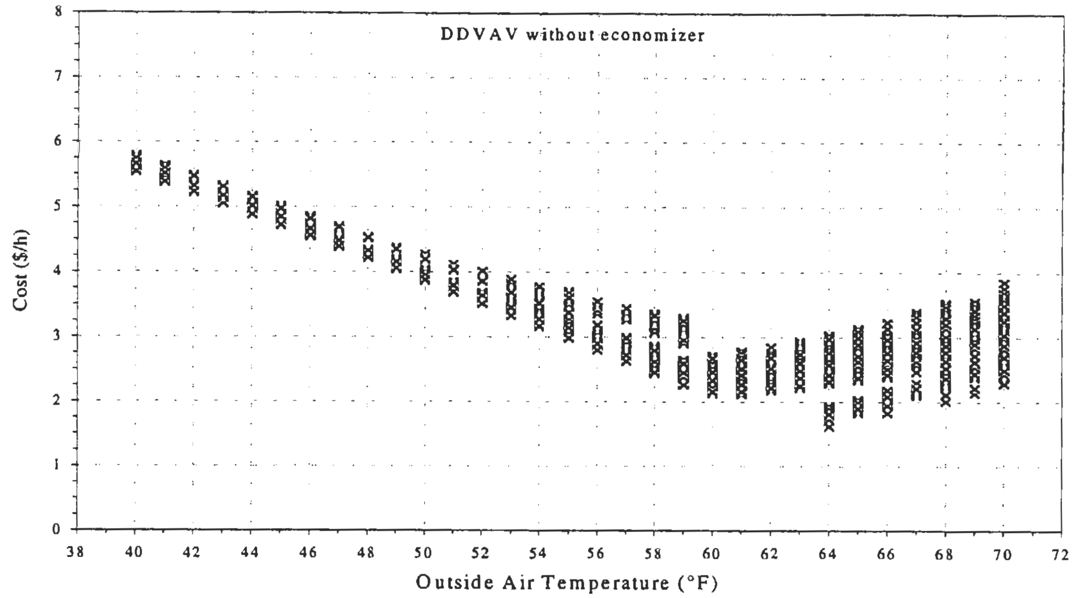

Figure 7. HVAC Operation Cost vs. Outside Air Temperature for Optimized Case

Results and Discussion

As was stated earlier, there does not appear to be any significant energy savings from the temperature or enthalpy economizers in this application. From the simulation, the temperature economizer cost 0.76% less than without the economizer, and the enthalpy economizer cost 1.32% less. From these results, it can be clearly seen that standard temperature and enthalpy economizer cycles do not have any kind of payback potential in this particular DDVAV case study. However, optimization of the hot and cold deck schedules deserves further study. Figures 8 and 9 illustrate the hot and cold deck temperatures for the standard and the optimized schedule and can be seen as follows:

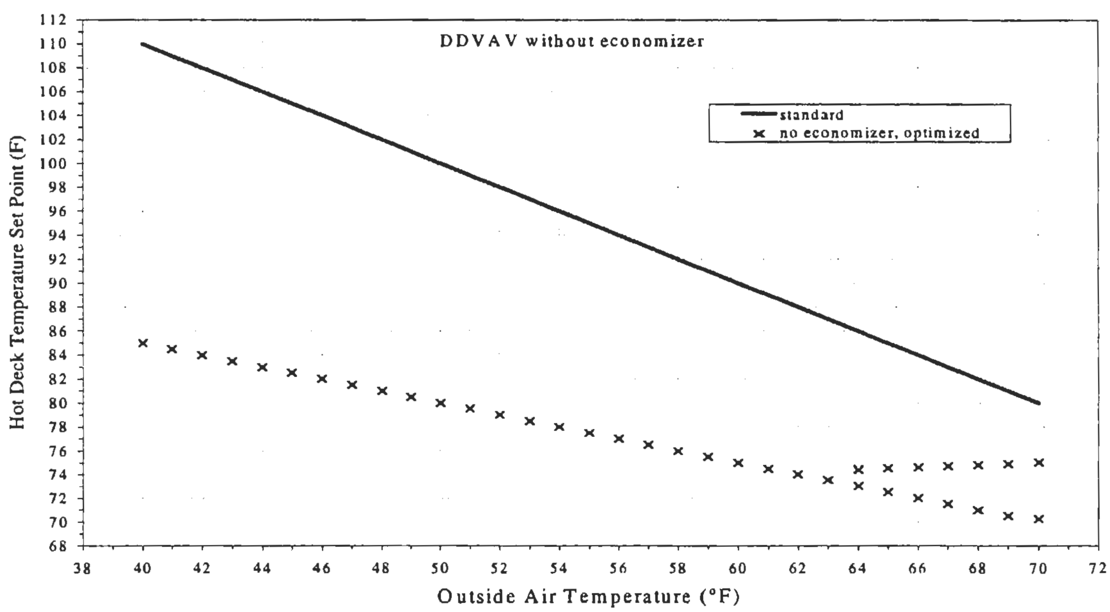

Figure 8. Hot Deck Temperature Set Point vs. Outside Air Temperature for Optimized and Standard Case

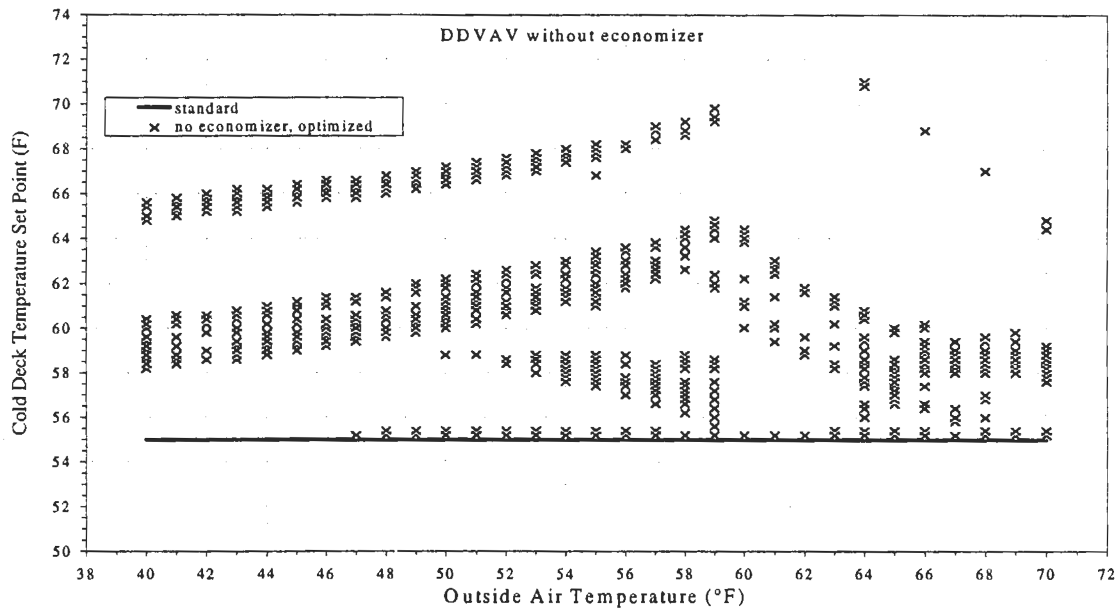

Figure 9. Cold Deck Temperature Set Point vs. Outside Air Temperature for Optimized and Standard Case

It is clear from Figures 8 and 9 that the standard AHU deck schedule results in unnecessary heating and cooling as compared to the optimized case. As can be seen from Figure 8, the optimized hot deck schedule has the same form as the standard schedule, at least until reaching the return air temperature, but is significantly lower most of the time. And whereas Figure 9 shows the standard case has a constant cold deck temperature, the optimized cold deck set point varies; in some cases approaching the hot deck set point when outside air temperature and relative humidity permit. Here is where the savings are to be had, and this is shown below in Figure 10. Here the simulation with optimized deck schedules indicated a 19.8% reduction in heating and cooling energy consumption over the standard schedule with no economizer. The simulated cost for optimized hot and cold deck reset schedules totaled $12,164 while operating in the studied 40°F to 70°F temperature range.

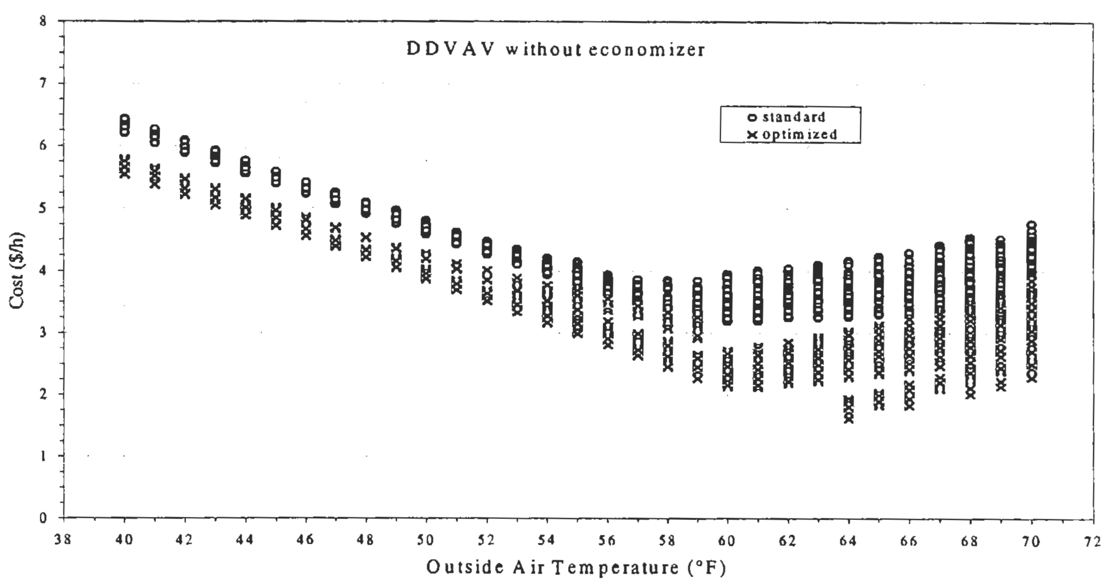

Figure 10. HVAC Operation Cost vs. Outside Air Temperature for Standard and Optimized Case

The current system, having hot and cold deck reset schedules, as well as a temperature economizer, versus the optimized hot and cold deck reset schedules is another interesting comparison. The figures below show the hot and cold deck schedules. Just as in the standard case, the optimized deck resets are an improvement over the existing system. The comparison of HVAC system operating cost is shown on the following page. The existing operation is the result of optimization based on outside air temperature. If the cold deck can be reset based on both outside air temperature and relative humidity, the system cost can be further reduced by 7.7%.

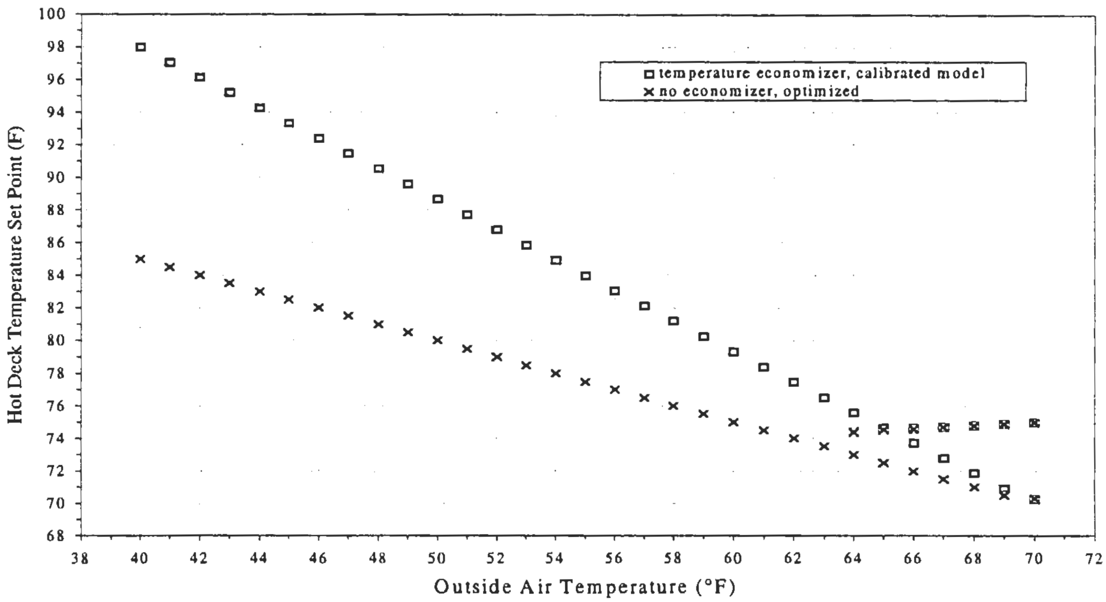

Figure 11. Hot Deck Temperature Set Point vs. Outside Air Temperature for Optimized and Calibrated Model Case

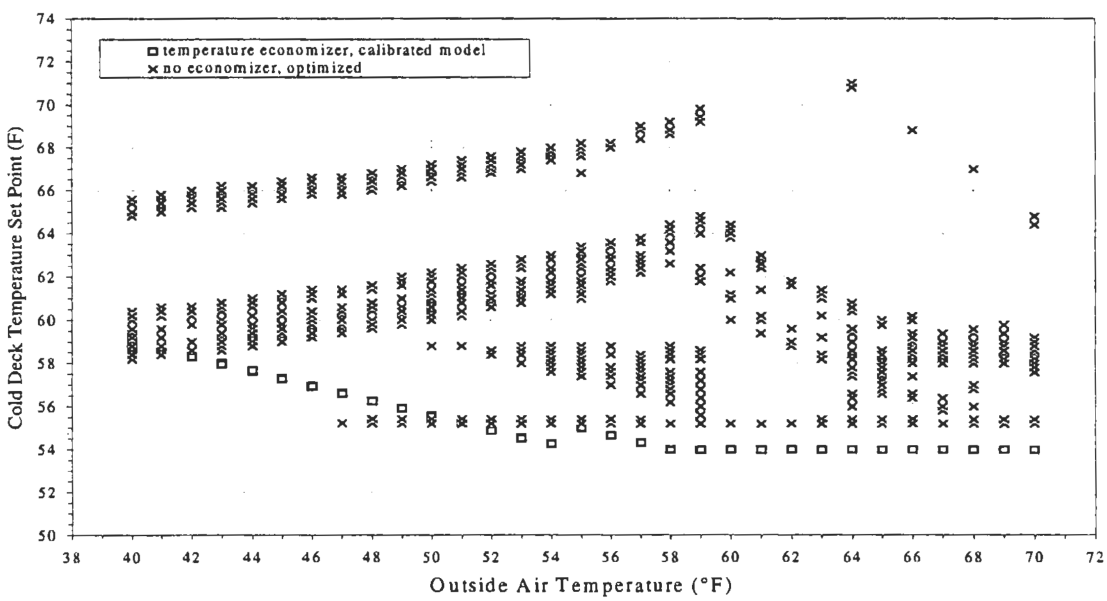

Figure 12. Cold Deck Temperature Set Point vs. Outside Air Temperature for Optimized and Calibrated Model Case

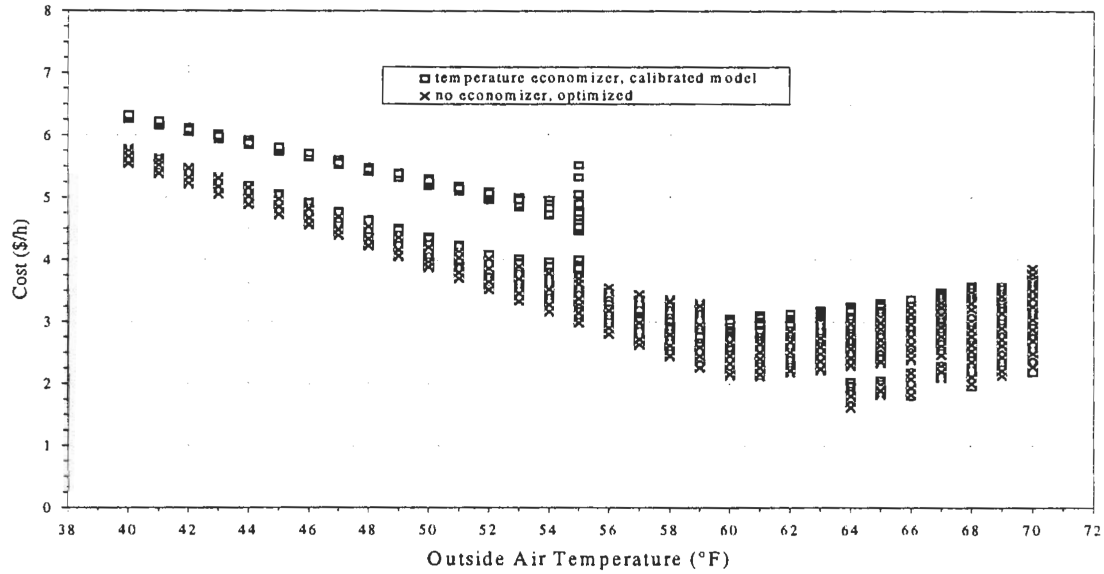

Figure 13. HVAC Operation Cost vs. Outside Air Temperature for Optimized and Calibrated Model Case

Conclusion

Model simulation has proved very useful in this case study. There was no real benefit from an economizer cycle for this building. Had simulation or some other studies been conducted, the additional cost of a temperature economizer might have been avoided. Reset of the hot and cold supply air temperature based on outside air enthalpy was also investigated. Optimizing hot and cold deck schedules can reduce the energy cost for this building by approximately 20%, when the outside air temperature is within the studied range of 40°F to 70°F.

Reference

1. Kao, J. Y., W. H. Parken, and E. T. Pierce. 1982. "Strategies for Energy Conservation for a Large Retail Store.", National Bureau of Standards, NBSIR 82-2580

2. Levenhagen, J. I. and D. H. Spethmann. 1993. HVAC Controls and Systems, McGraw-Hill, Inc.

3. Liu, M., A. Athar, Y. Zhu, and D. E. Claridge. 1995. "Reduce Building Energy Consumption by Improving the Supply Air Temperature Schedule and Recommissioning the Terminal Boxes." Energy Systems Laboratory Report: ESL-TR-95/01-04. Texas A&M University. January 1995.

4. Liu, M., and A. Athar. 1994. "Reducing Building Energy Costs Using Optimized Operation Strategies for Constant Volume Air Handling Systems." Proceedings of Ninth Symposium on Improving Building Systems in Hot and Humid Climates. May, 1994. Arlington, Texas. ASHRAE Transaction SD-95-3-4

5. Liu, M., D. E. Claridge, and B. Y. Park. 1997. "An Advanced Economizer Controller for Dual Duct Air Handling Systems-with a Case Application." ASHRAE Transactions V. 103.

6. Liu, M., 1997. "User's Manual for Air Side Simulation Programs (AirModel)." ESL. Texas A&M University. January 1997

7. Park, C. K., G. E. Kelly and J. Y. Kelly. 1984. "Economizer Algorithms for Energy Management and Control Systems." National Bureau of Standards, NBSIR 84-2832

8. Parken, W. H., C. K., J. Y. Kao, and G. E. Kelly. 1982. "Strategies for Energy Conservation in Small Office Buildings." National Bureau of Standards, NBSIR 82-2489