Energy Conservation

System dynamics and energy use

Evaluation of energy-saving effectiveness of computer technology when linked with commercial building energy management control systems

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NUMEROUS COMPANIES are now marketing energy management control systems (EMCS) for use in commercial buildings. One of the goals of these systems is to reduce building energy consumption by using computer technology to help control the HVAC equipment and to optimize the HVAC "process" within a building. This article presents the results of a joint research effort between ASHRAE, IBM and the University of Wisconsin-Madison to evaluate such control. The project was ASHRAE RP-321, entitled "HVAC System Dynamics and Energy Use in Existing Buildings." The goals of the project were:

1. To study the process dynamics and interactions of a building HVAC system through the use of collected test data and computer model of equipment;
2. To determine, via computer simulations, the effect of the time between control decisions in the dynamic control of an HVAC system;
3. To determine, via computer simulations, dynamic HVAC operating strategies that will potentially reduce the HVAC system energy consumption.

The initial findings on equipment dynamics and model development have been presented (Hackner et al. 1984). The significant transient effects for the test building were found to be the cooling tower response to fan speed changes, the "flush time" of the chilled water through the system, and effects of the building structure due to thermal capacitance. The insignificant transient effects were the chiller response to set point (supply water) changes and the air handling unit response to set point (supply air) changes. Component models that included the main effects were developed for the HVAC system. These models were based on ASHRAE standard algorithms (Stoecker 1971) and used experimental data from the test facility to determine the model parameters.

Nomenclature

$\dot{m}$	mass flow rate
P	pressure
T	temperature

Subscripts

CHWR	chilled water return
CHWS	chilled water supply
CWR	condenser water return
CWS	condenser water supply
RA	return air
refr	refrigerant
SA	supply air
stat	static pressure
WB	wet bulb

The final results are described in this article. The optimal control strategy for the cooling season is developed. System simulations are performed and the effect of system dynamics on control is determined. The energy savings possible through dynamic control is evaluated. The subject of comfort dynamics is not specifically addressed due to the lack of computer monitoring and control of comfort variables.

System description

The HVAC and control systems were evaluated on a commercial building located in Atlanta, Georgia. A schematic of the main portion of the HVAC system is shown in *Figure 1*. It consists of five water chilling units, four constant-volume perimeter and two variable-air-volume interior air handlers, and a two celled cooling tower. Two constant speed 550-ton centrifugal chillers provide the bulk of the chilled water used during the cooling season. The cooling tower rejects the heat from the condenser. Each of the two cells of the tower has its own two-speed fan.

The cooling loads in each of the four perimeter zones (SW, SE, NW and NE) of the building are met by constant volume air handling units (AHU). The modes of operation are either cooling or recirculation in which air is only circulated through the perimeter zones.

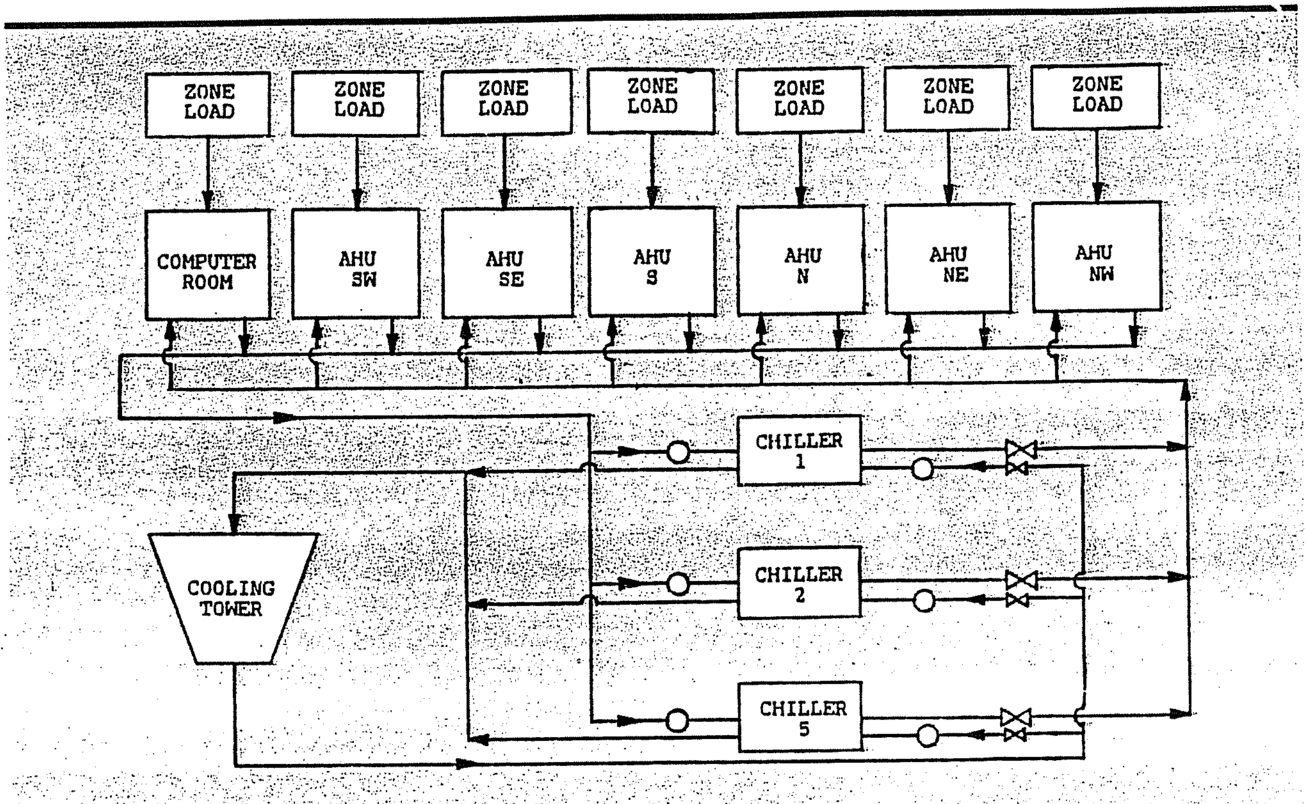


Figure 1—Schematic representation of an Atlanta HVAC system

Two variable air volume AHUs provide the air conditioning for the core of the building (S and N zones). The units run during occupied hours in either "free cooling," "pay cooling," or evaporative cooling modes. In the "free cooling" mode, a combination of outside air and return air is used to provide the supply air for the core zone. Chilled water is supplied to the AHU in the "pay cool" mode. In the evaporative cooling mode, low humidity outside air is brought into the unit and evaporatively cooled. Additional information on the building and equipment modeling can be found in Hackner et al. (1984) and Hackner (1984).

The HVAC system is made up of many different components, each with its own operating characteristics and its own internal control strategy. For the energy management control system (EMCS) to be effective, it must have information on the individual HVAC system component. The interaction between the components must be known in order for the EMCS to make effective control decisions. *Figure 2* is a simplified schematic of the HVAC and control systems for the Atlanta facility.

The sequence of events for a reduction in room thermostat setting is as follows: First, the supply air dampers in the terminal units open to allow an increased flow of supply air. The corresponding drop in duct static pressure changes the pitch of the blades in the VAV fan and increases the supply air flow rate. The increased air flow produces an increase in supply air temperature above its setting, which then signals the chilled water valve to open. The chilled water return temperature increases due to the greater cooling load. This provides a signal to the prerotation vanes on the compressor inlet, which open up and increase the refrigerant flow rate. The increase in evaporator load and compressor power increases the condenser load. The consequent increase in the temperature of the condenser water supply is sensed by the EMCS and used to set fan status.

The EMCS in the Atlanta facility monitors the flow rates, temperatures and pressures shown in *Figure 2*. It also controls the operating status of all of the major components. The three aspects of the main AHU unit control are the on-off status of the units, the set point for the supply air temperature, and the settings for the return and outside air dampers (not shown on *Figure 2*).

In the current control strategy, the main AHUs are brought on-line at 7:30 a.m. and are shut down at 6:00 p.m. on a normal working day. On weekends and holidays the main AHUs are usually in the "off" condition. The set point depends on the outside air temperature. At ambient temperatures below 10 F (−12 °C) the setpoint is 65 F (18 °C), at temperature above 55 F (12 °C) the set point is 55 F (12 °C), and the set point varies linearly between these ambient temperatures. The choice of a 55 F (12 °C) limit was based on providing thermal comfort. It was found that 55 F (12 °C) was the lowest supply temperature that could be used before people began complaining to the building engineers about cold drafts. The position of the return and outside air dampers for the main AHUs is based on the enthalpies of the return and outside air.

The EMCS also controls the chilled water supply set point temperature. At present, the chilled water return temperature is used to determine whether a change in set temperature is desired. The decision to use the chilled water return temperature was made by the building operations personnel. It was felt that the return water temperature would provide a stable base on which to make set point changes. The choice of 58 F (14 °C) as the base-line was the result of experiments done by the operators to determine the "best" temperature upon which to key set point changes.

The operating status of the cooling tower fans is controlled by the approach temperature, which is the tem-

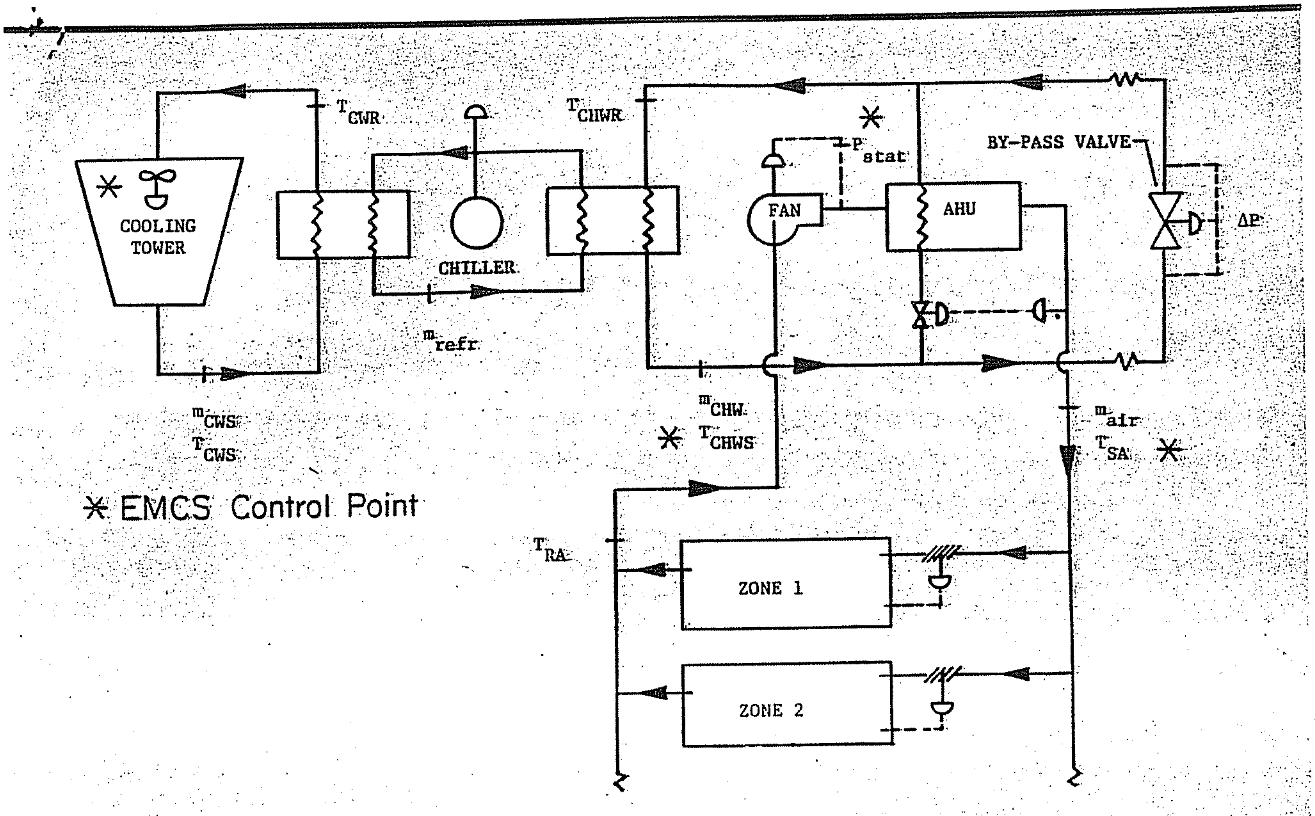


Figure 2—HVAC system schematic including the control points

perature difference between the ambient wet bulb temperature and the condenser water supply set point. The condenser water set temperature is based on the ambient wet bulb temperature. It is a constant value of 65 F (18 °C) below an ambient wet bulb of 58 F (14 °C), and increases linearly above that valve. The status of the cooling tower fans is set according to the deviation between the temperature at which condenser water returns to the building and the set point temperatures.

There are a number of local control loops that are not part of the EMCS. The room thermostat is set by the occupants, and controls the VAV terminal units. The pressure in the air distribution is sensed and used to control fan speed. The supply air temperature provides a signal to the valve in the chilled water line that supplies the cooling coil in each AHU. A bypass loop in the chilled water line maintains a constant flow of chilled water through the chiller evaporator. These control loops were included in the system models, but changes to them were not evaluated in this study.

Alternative strategies

One of the major goals of this project was to determine and evaluate HVAC system control strategies that would potentially reduce the system energy consumption while maintaining comfort. The alternative control strategies were developed using the TRNSYS simulation program (Klein et al. 1983) and based on the computer models of the HVAC equipment previously described (Hackner et al. 1984). The new strategies, which are described in this section, include limitations due to the characteristics of the individual HVAC components and occupant comfort levels.

Chiller operating status

The optimal point to switch from operation with one chiller to operation with two chillers depends on chiller load and ambient wet bulb temperature. In *Figure 3*, the coefficient of performance (COP) as a function of chilled water load is shown for operation with one chiller and with two chillers. Manufacturer data were used to generate the curves in *Figure 3*. The upper pair of curves is for a COP calculated by dividing the chilled water load by the chiller power. The lower set of curves is for a COP that also includes the power to run the chilled water pumps and the condenser water pumps. The latter value reflects actual system energy use.

Using the COP based on chiller power only, the optimal "switch point" would be point 1, or a load of approximately 460 tons. However, the inclusion of pump power indicates a switch point of approximately 700 tons (point 4). If the decision to switch the operating status of the chiller did not include the power to run the pumps, a significant energy penalty could be incurred. At 460 tons, the actual system operation is at point 2. Switching at this point would lower the COP by 20 percent to point 3, whereas the switch should not occur until point 4. The COP that takes into account the power to run the pumps represents a more correct value on which to base control decisions.

Figure 4 shows the effect of ambient wet bulb temperature on the optimal switch point load. At a wet bulb temperature of 85 F (29 °C) the optimal switch point is decreased from 690 to 610 tons. The curves are relatively flat, and the switch point is not significantly affected by wet bulb temperature.

The optimal points to switch from one chiller to two chiller operation occur above the maximum power draw limit for a single chiller. In practice, the optimal control strategy is to allow a chiller to meet up to its maximum chilled water load for the given operating conditions. If the building load increases further, an additional chiller should be brought on-line. The controller must "remember" the chilled water load that occurred just prior to the switch point. If the load decreases, then the controller can make a deci-

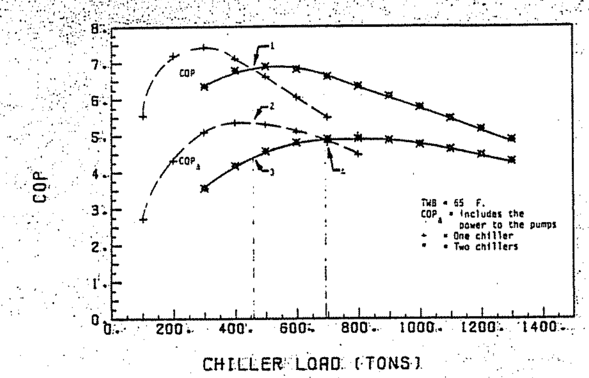


Figure 3—COP versus chilled water load, CHWS = 46 F

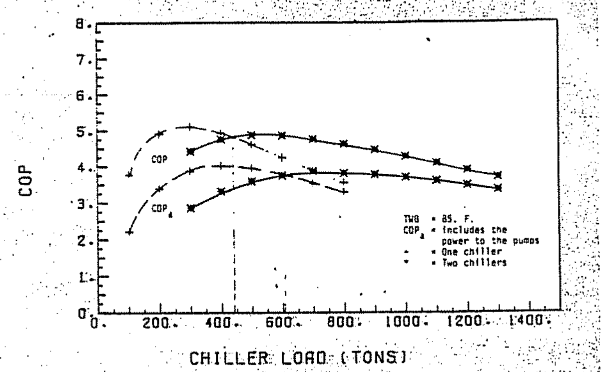


Figure 4—COP versus chilled water load, CHWS = 46 F

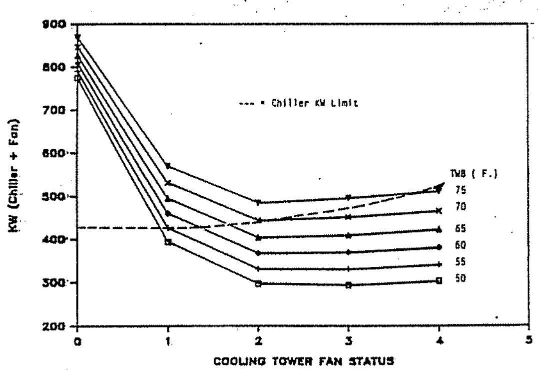


Figure 5—Cooling tower fan status optimization, chilled water load = 625 tons, one chiller operation

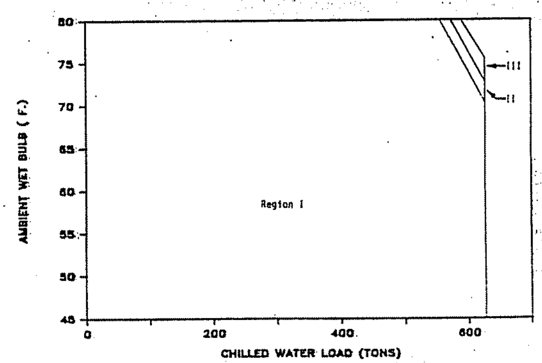


Figure 6—Optimal regions of cooling tower fan operation, Region I fan status low-low, Region II low-high, and Region III high-high, one chiller operation

sion as to whether the load is low enough to allow operation with one chiller. This strategy is currently used in the Atlanta building.

Cooling tower fan status

Approach control is currently used for the cooling tower fans. This form of control inherently neglects the interaction between the cooling tower and the chiller. A more appropriate form of control would be to optimize the operation of the chiller/cooling tower subsystem to minimize energy use. Using the chiller and cooling tower computer models, a simulation to determine the optimal control strategy was performed. Various combinations of chilled water load, ambient wet bulb temperature, cooling tower fan speeds, and number of operating chillers were used as inputs to the simulation. At each operating condition, the power consumption for all levels of fan status was computed, and the minimum value identified. The minimum power for each operating condition is shown in *Figure 5* with the various modes of operation for the two-speed cooling tower fans given in *Table 1.*

With one chiller on the optimal fan status is mode 2 under almost all conditions. This is operation with both fans on low speed. The exception to this occurs at ambient wet bulb temperatures greater than 70 F (21 °C), when the maximum chiller power draw is exceeded. Under these conditions, a decision must be made whether to switch to two chiller operation or to continue operating with one chiller and increase the fan status level. Except for very high loads, it is better to increase the fan status level, since there is a high energy "overhead" associated with turning on an additional chiller *(Figures 3 and 4).*

From the results of the chiller/cooling tower simulations, a map showing the optimal regions of operation for the fan levels was produced. *Figure 6* is a map for operation with one chiller at a chilled water set point

Table 1

Fan Status

Mode	Fan #1	Fan #2
0	off	off
1	off	low
2	low	low
3	low	high
4	high	high

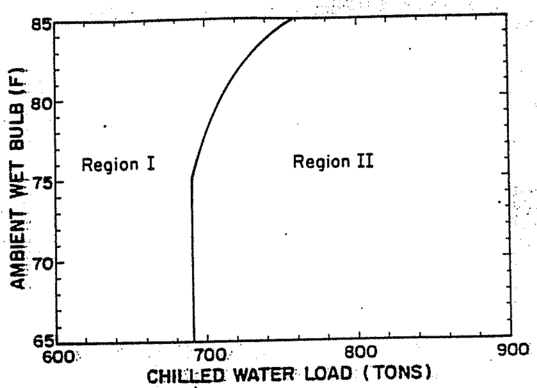


Figure 7—Optimal regions of cooling tower fan operation, two chiller operation, Region I fan status low-low and Region II low-high

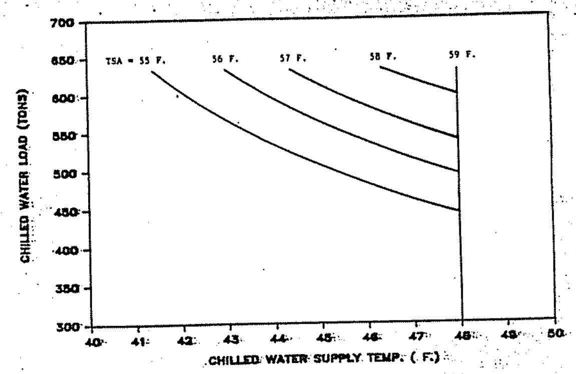


Figure 9—Chilled water supply temperature limits for various chilled water loads and AHU supply air temperatures

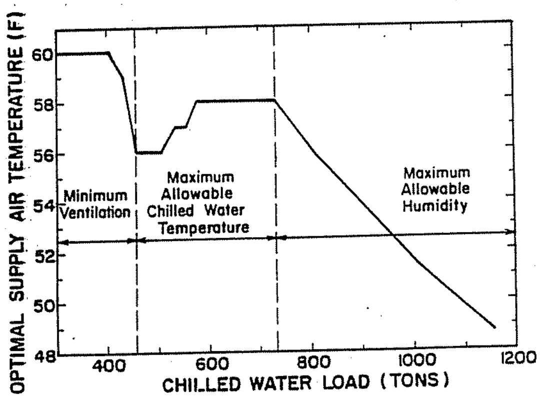


Figure 10—Optimization results for the AHU supply air temperature versus chilled water load; the determinant factor for each division is indicated for each region

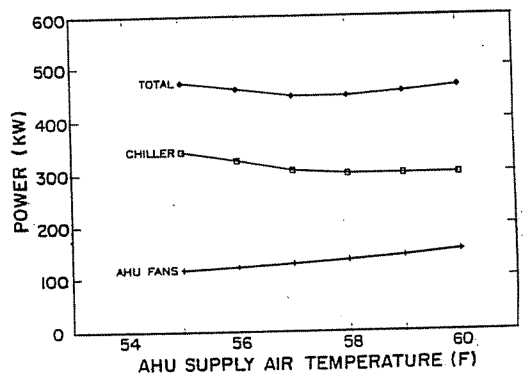


Figure 11—Power use and distribution versus AHU supply air temperature, load = 550 tons and T_{we} = 65 F

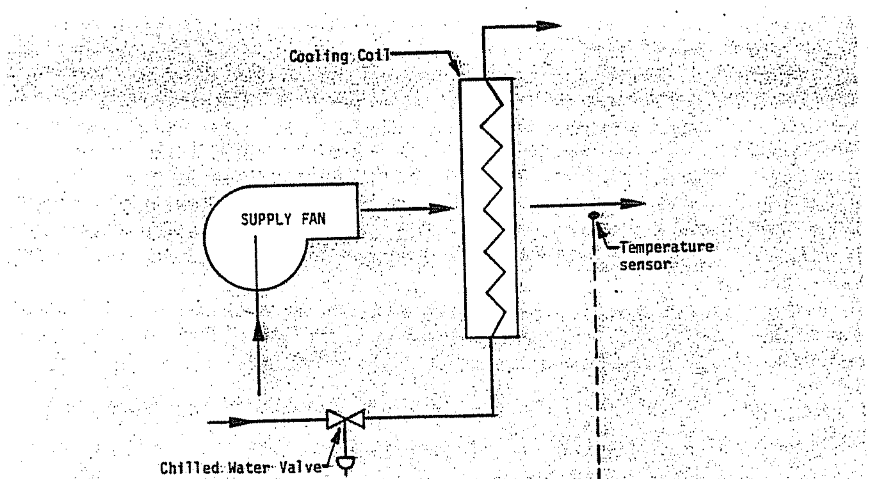
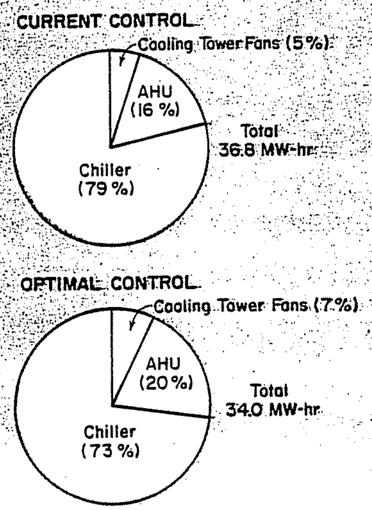


Figure 8—Schematic of the cooling coil and fan arrangement for an air handling unit

temperature of 48 F (9 °C). Region I is the mode 2 level, while regions II and III are modes 3 and 4, respectively. Regions II and III result from the chiller power limit. For lower chilled water temperatures, Regions II and III would expand to the left on the figure. At lower chilled water supply temperatures, the power limit is reached at lower chiller loads and ambient wet-bulb conditions. A more rapid switch to a new fan operating level would occur for chilled water supply temperatures less than 48 F (9 °C).

Figure 7 shows the optimal cooling tower fan levels for two chiller operation for a chilled water set point of 48 F (9 °C). Region I is, again, the mode 2 level and Region II is the mode 3 level. Operation with both fans on high speed is never best.

These maps were used in simulations of the optimal control strategy. For example, for an ambient wet bulb of 65 F (18 °C) and a load of 200 tons, operation would be with one chiller and a fan status of mode 2 (*Figures 3*



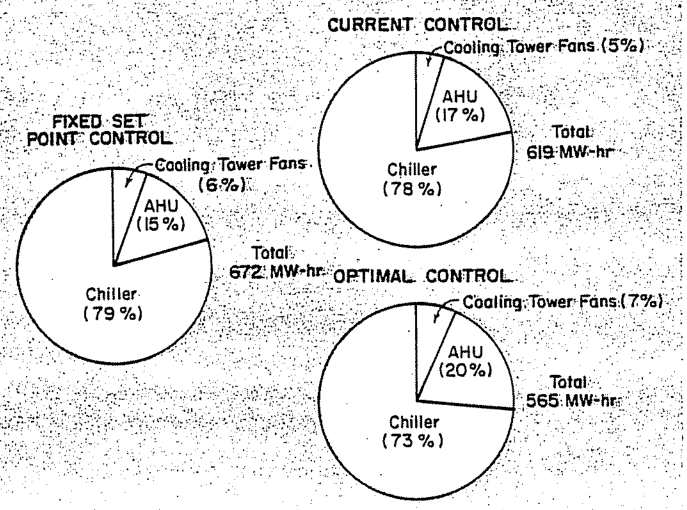


Figure 12—Energy use distribution for current control versus optimal control via simulation, one week in May

Figure 13—Energy use distribution for current control versus optimal control via simulation, mid-May to mid-September

and 6). The status would remain the same until a maximum allowable load of 620 tons was reached, at which point a second chiller would be turned on. Fan status would remain in mode 2 until a load of 690 tons was reached, and then one fan would be turned to high speed. This example illustrates the decisions made in the optimal control strategy.

Air temperature

In the current strategy the chilled water supply set point is adjusted based on the return water temperature deviation from a desired set value. The AHU supply air set point is determined based on the outdoor dry bulb temperature. The facility engineers indicate that this control scheme is stable and meets comfort criteria. However, this control scheme neglects any interaction between the AHU supply air temperature and the chilled water supply temperature. The determination of the optimal temperature of these two set points must include this interaction.

A simplified schematic showing the cooling coil and fan for the air handling unit is shown in *Figure 8*. The internal control strategy modulates the flow of chilled water through the coil over a finite range in order to achieve a desired supply air temperature. This control scheme introduces the first limit on the operation of the AHU. For a given airflow rate and chilled water supply temperature, there is a lower limit on the temperature of the air that exists from the cooling coil.

The range of chilled water flow rate is affected by the characteristics of the chilled water valve and the chilled water distribution system. If the maximum chilled water flow rate is reached or if the supply air set point temperature is too low, then the actual supply air temperature will be out of control. The desired supply air temperature can not be reached.

There are other limits on the optimal chilled water supply and AHU supply air temperature. The chilled water supply temperature is maintained above 42 F (5 °C) to lessen the chance for a localized freezing to occur in the chiller evaporator coils. The chilled water supply temperature remains below 48 F (9 °C), which was found to be the upper limit on the supply water temperature that could be tolerated by the computer room chilled water system. The lower limit on the AHU supply air temperature is 55 F (12 °C), based on occupant comfort. There is an upper limit on the supply air temperature that can maintain a desired humidity level inside the building. The supply air temperature must be low enough to handle the latent load from the space and possibly from the ambient ventilation. A minimum flow rate of ventilating air must be maintained to meet the fresh air requirements of the space.

A step-by-step procedure was used to determine the optimal chilled water supply and AHU supply air temperatures subject to these constraints. A TRNSYS simulation was performed for building operation under a wide

Table 2

Energy Use With Different Control Strategies

Control Strategy	Total Energy MWh	Chiller (%)	AHU Fan (%)	Tower Fan (%)
Fixed set points	672	79	15	6
Current	619	78	17	5
Optimal	565	73	20	7

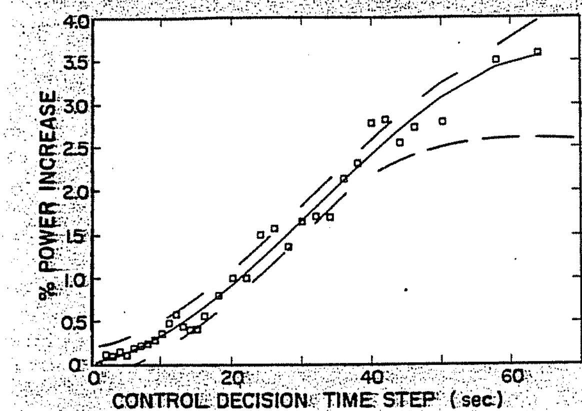


Figure 14—Percent power increase versus the control decision time step size; results via computer simulation

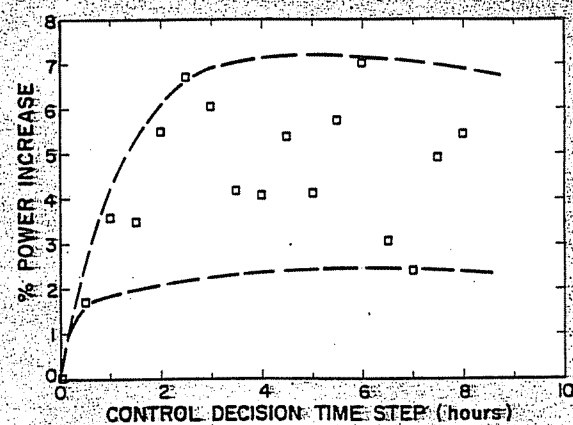


Figure 15—Energy use distribution for current control versus optimal control via simulation

range of load and ambient conditions. The building chilled water loads varied between 300 and 1200 tons and the ambient wet bulb temperature from 45 F (7 °C) to 75 F (24 °C). The chilled water supply temperature was varied between 42 F (5 °C) and 48 F (9 °C) and the supply air temperature between 55 F (12 °C) and 60 F (15 °C). A total of over 40,000 combinations of operating conditions were simulated. The minimum energy consumption for each ambient and load condition was then identified. Thus, the optimal control set points were determined by searching the results of the analysis matrix for the minimum energy use positions. This procedure is probably impractical to perform on-line during the control of a building. However, the results of the optimization can be used to develop control algorithms that an EMCS can then apply.

The first step used to process the simulation results was to determine the chilled water supply temperature limits for various supply air temperatures. Since there is a limit to the rate of the chilled water flow through the cooling coil, there is a maximum chilled water supply temperature that can be used to meet a certain supply air temperature. The limits on the chilled water supply temperature are shown in *Figure 8.* For example, at a chilled water load of 550 tons and a desired supply air temperature of 56 F (13 °C), the optimum chilled water temperature is 45 F (7 °C). If the load is reduced to 500 tons, the optimum chilled water temperature becomes 47 F (8 °C).

The optimal supply air temperature was determined for various building loads, as shown in *Figure 10.* At low cooling loads (less than 450 tons), the minimum flow rate of ventilating air is the dominant factor in determining the optimal supply air temperature. A higher supply air temperature can be used at these low load conditions, because a minimum amount of supply air must be delivered to the building zones. For cooling loads between 450 and 600 tons, the optimal temperature is determined by the chilled water supply temperature limits shown in *Figure 9.* For loads between 450 and 500 tons, a supply water at 48 F (9 °C) is able to yield a supply air temperature of 56 F (13 °C). As the building load increases beyond 500 tons, it is better to maintain the upper limit on the water temperature and to meet the load with a higher mass flow rate of warmer supply air. At any cooling load, a trade-off between the chiller power and the AHU fan power occurs. *Figure 11* illustrates this trade-off between the fan power and chiller power as supply air temperature increases. For a chilled water load of 550 tons the optimal supply air temperature is 57 F (14 °C). The total power is not too sensitive to supply air temperature. The results of the optimization show that the additional fan power necessary to meet the load with a higher supply air temperature is much less than the additional chiller power necessary to produce a lower chilled water temperature and a lower supply air temperature.

At loads above 600 tons, another important limiting factor comes into effect. The limit on the humidity level for the building now has an effect on determining the supply air temperature that can be used. A number of calculations were performed to find the limit on the supply air temperature for various cooling loads under the following assumptions. The indoor conditions of the zone were to be maintained at 76 F (24 °C) and a relative humidity of 60 percent. The latent load for the zone was taken as a constant value of 40 tons based on estimates of the building occupancy level and other sources of latent loads. The outdoor conditions were such that a minimum amount of outside air was being used. To account for a possible increase in the coil load, it was assumed that an additional 1 Btu/1bm dry air was added to the enthalpy of the return air in order to calculate a "mixed" air enthalpy. Ten percent of the mixed air was assumed to bypass the cooling coil.

Based on the above assumptions together with the maximum airflow rate for the AHU supply fans, the limiting supply air temperature was determined. At a building load of 725 tons the load is met by supplying the maximum flow rate of air at a temperature of 58 F (14 °C). For loads above 725 tons, the supply air temperature is decreased in order to meet the load. The chilled water temperature will then decrease to maintain the lower air temperature.

The optimal strategies to control the HVAC system were thus determined. The results of this section were used to control the long term building simulations presented in the next section.

Results for optimal control

A simulation of the entire HVAC system was performed to compare the system energy use under the various control strategies. These are optimal control, which controls the system to operate always at the point of minimum energy use, fixed set point control, and the current building control in which set points are varied. Actual weather records were not available for the site, and TMY (Typical Meteorological Year) data were used to provide the necessary input data. These data are for a single year and contain hourly solar radiation and meteorological data for an average year. The inputs used were the ambient wet bulb temperature, the ambient dry bulb temperature, and the horizontal solar radiation measurement.

The results for a typical simulation run under optimal and the current method of control are shown in *Figure 12*. The data used to drive this simulation were for one week in the month of May, which represents an average air conditioning situation. The time increment used for making control decisions was one-eighth hour. Optimal control could reduce energy consumption by about eight percent. For both methods of control, the chiller uses the most energy, and is the component on which to concentrate in any energy saving control alternatives.

There is also a difference in the distribution of energy use between the two control strategies. For the optimal control case, the power consumption has been increased both relatively and absolutely in the cooling tower and AHU supply fan areas. However, the additional energy use in these areas is more than offset by the reduction in chiller energy use.

Figure 13 shows the results for a seasonal simulation. The current control and optimal control simulations were run from mid-May to mid-September, the major portion of the cooling season. A decrease in energy use of about nine percent could be obtained by using the optimal control strategies. A dollar value can be placed on this decrease by assuming the cost of electricity is $.06/kWh. The saving that results from optimal control is then $3240 for this particular period of operation.

Results for fixed set points

In order to put the comparison between current and optimal control in the proper perspective, a baseline was established using simulations with fixed set points. The values of the fixed set points were chosen to represent current HVAC practice. The results of this simulation allows assessment of the benefits of improved control strategies. The chilled water temperature for the delivery air was a constant value of 52 F (11 °C), which is in the range of current practice. The fan speed control on the cooling towers was based on the approach temperature, and is the same as that is used in the current Atlanta strategy. It was felt that a fixed fan speed would produce either unreasonably high fan power or unreasonably high condenser water return temperatures under some operating conditions. The remaining control parameters were the same as in the other simulations.

The simulation was performed for the period of mid-May to mid-September. The total energy requirements for the tower, air handler fans, and chiller are given below in comparison with the results of the other simulations. The energy consumption for the fixed point operation is 8.5 percent higher than that using the current control, and 19 percent higher than that using optimal control. The energy use for all three modes of control is surprisingly close.

It was expected that the fixed set point operation would use significantly more energy than either of the other control strategies. The reason that the energy requirements are so close results from the many energy tradeoffs between the components that make up the total use. For example, in the fixed set point operation, the chiller set temperature is lower than that for the dynamic control strategies and chiller power is increased accordingly. However, AHU power is decreased since a lower air flow is required to meet the same load at the lower chilled water set temperature. In control with variable set points the system operates more favorably, but compensations occur continuously.

The distribution of the energy use shows that a higher fraction of energy is used for the chiller in the fixed point operation than in the other control strategies. This is because the chilled water temperature is colder on the average than in the other strategies. The AHU fan energy consumption is less than in the other strategies since the chiller water temperature is lower and less air flow is needed to meet the load. Overall, the distribution is not significantly different from that of the other control strategies.

Results for dynamic control

A major goal of this project was to determine the impact of dynamic control on the energy consumption of HVAC systems. For this project, dynamic control means the resetting of control points in response to changes in chilled water load and ambient conditions. To study this effect, the entire HVAC system was simulated using 1/64th hour timesteps to calculate loads. The time increment used to make control decisions in the current IBM control strategy was then varied from 1/64th of an hour up to eight hours. The large amount of computing time that was necessary to generate each data value required that the simulations be run for one week intervals.

The simulation results for control decision timesteps up to one hour are shown in *Figure 14*. The percent power increase is defined in terms of the energy consumption of a simulation using the 1/64th hour time increment as the base case. The dashed lines indicate approximate error limits for the simulation of ±0.2 percent.

The results for longer control decisions timesteps are shown in *Figure 15*. The data points no longer follow

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any particular pattern. The randomness of the points can be attributed to two major factors. The first is that as the decision timestep increases, the control of the system becomes less and less dynamic. A control decision is made based on the current status of the system, even though the operating conditions of the system change before the next decision is made. The second factor involves the major switch points in the daily operation of the building relative to the time at which decisions are made. The AHUs are turned on at 7:30 a.m. and shut off at 6:00 p.m. Depending on when the decisions are made, an inappropriate decision persists for a long period. For these reasons an uncertainty envelope is shown around the data points in *Figure 15.*

The possible power increase for "non-dynamic" control could range between two and seven percent. For a conventional one-hour time between set point adjustments, the power increase would be three to four percent. A time increment between changes of 20 minutes reduces the increase to less than one percent.

The overall objective of the project was to determine operating strategies for HVAC systems which incorporate system dynamics and interactions and potentially reduce energy use. The following conclusions can be drawn:

1. Dynamic HVAC operating strategies that will potentially reduce the energy consumption can be identified through simulation techniques. This provides a procedure for exploring control strategy options.

2. The effect of the time between control decisions was evaluated. The results showed a three to four percent energy savings for a system operating under dynamic control, as opposed to a system in which control decisions were made on an hourly basis. A maximum energy increase of seven percent is possible.

3. Using an optimal control strategy, chosen to minimize energy consumption at each operating condition, an overall energy use reduction of 8.7 percent to the current strategy is possible. Compared to operation with fixed set points, a reduction of 19 percent is possible.

4. Potential energy saving HVAC operating strategies were identified and reliable equipment models were developed. These are useful to building operators, HVAC controls persons and future researchers.

References

Carrier, W.H. 1911. "Rational psychrometric formulate," *ASME Transactions*, V. 33, p. 1005.

Hackner, R.J. 1984. "HVAC system dynamics and energy use in existing buildings." MS thesis, University of Wisconsin-Madison.

Hackner, R.J., Mitchell, J.W., and Beckman, W.A. 1984. "HVAC system dynamics and energy use in existing buildings – Part I." *ASHRAE Transactions*, Vol 90, 2B, pp. 523-530.

Klein, S.A. et al. 1981. "TRNSYS A transient simulation program." University of Wisconsin-Madison, Engineering Experiment Station Report 38-11, Version 11.1, April.

Klein, S.A. et al. 1983. "TRNSYS A transient simulation program." University of Wisconsin-Madison, Engineering Experiment Station Report 38-12, Version 12.1, December.

Marley Cooling Tower Company, 1982. "Cooling tower energy and its management." Marley Technical Report h-001A, October.

Stoecker, W.F., ed. 1971. *Proposed procedures for simulating the performance of components and systems for energy calculations.* 2nd ed. New York: ASHRAE.

Stoecker, W.F., and Jones, J.W. 1966. *Refrigeration and air conditioning.* New York, McGraw-Hill Book Company.

Tuve, G.L., Domholdt, J.C. 1966. *Engineering experimentation.* New York, McGraw-Hill Book Company.

Whiller, A. 1967. "A fresh look at the calculation of performance of cooling towers." *ASHRAE Transactions*, Vol. 82, Pt. 1, p. 269.