Coordinated Control of HVAC Systems

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ABSTRACT

This paper describes the development of new control logic for starting and stopping energy-intensive equipment in buildings such as staged air-conditioning units. The concept is to use pulse-width modulation (PWM) instead of level-crossing logic. A finite state machine is used to handle the case where a single unit has multiple stages of operation. An optimized coordinator determines the phase of the PWM signals of each unit so that peak demand for power is minimized over each PWM period. Control logic for the PWM function was developed so that the phase could be manipulated by the coordinator. Computer simulations were used to assess the performance of the new strategy and to compare it to level-crossing logic. The following five metrics were used to assess the performance: 1) magnitude of the control error, 2) start/stop frequency, 3) average power consumption, 4) standard deviation of the power consumption, 5) peak power consumption. The computer simulations showed that the new strategy could reduce peak power consumption by 20% relative to level-crossing logic. The computer simulations also showed that the new strategy increased the magnitude of the space temperature control error by 11% and increased the number of start/stop operations by 27% relative to level-crossing logic.

1 INTRODUCTION

Equipment used to control process variables in buildings such as temperature is often designed to be operated by cycling on and off (or between stages if more than one "on" state) rather than as continuously modulating. Examples of such equipment include small to mid-sized packaged air-conditioning systems, furnaces, chillers operating at low loads, cooling tower fans, and some types of electrical heaters. In U.S. commercial buildings, approximately 2.2 quadrillion Btu of primary electrical consumption (30% of the total) can be attributed to on-off and staged electrical devices (EIA, 1998). Consumption for California is roughly one-eighth of the U.S. consumption. An equally large quantity of energy is consumed by on-off equipment in residential buildings. On-off equipment is so prevalent in buildings that even modest improvements in efficiency can have a tremendous cumulative effect on reducing energy consumption.

On-off control units normally start and stop equipment when the process variable (e.g., space temperature) crosses a level. The advantage of using level-crossings to trigger operation is that the control logic is extremely simple. However, there are a number of disadvantages. One disadvantage is that it is difficult to control the variation in the process variable with level-crossing logic, even if it is implemented digitally, because of the phase lag of the process. Even if the differential between the levels that trigger the on and off events is very small, the variation in the process variable may be large because of the lag or delay between the action of the controller and its effect on the process. This lag differs from one installation to another, which further complicates the problem.

Another problem is that level-crossing logic is also not well-suited for staged operation in which there exists more than one "on" state. One approach to operating staged units with level-crossing logic is to choose a different set of on-off levels for each stage. This is essentially proportional-only control action. Proportional-only control action leads to offset; the process variable increases or decreases with the load. For air-conditioning systems, this is undesirable for comfort and humidity control. It also is difficult to select the appropriate difference between the levels for each stage because of the inherently oscillatory nature of on-off systems even when they are tuned properly. If the sets of on-off levels for each stage are too close, then the unit will short-cycle, which wastes energy. Discriminating between short-cycling and normal cycling takes time and expertise. The fact that each installation is different complicates the problem even further.

An alterative method to using different on-off levels for each stage is operate the stages based on the output of a proportional-integral-derivative (PID) controller. If the integral gain is not zero, then this eliminates the problem of offset. However, it is more difficult to tune this kind of control system because there are now at least two parameters that must be chosen instead of just one.

The objective of this project is to develop and test the feasibility of a new control strategy for the operation of on-off and staged equipment in buildings using computer simulation methods. The performance metrics used to determine whether or not the objective has been met are energy consumption, peak demand, thermal comfort, and maintenance cost. Energy consumption and peak demand are derived from the equipment model. Thermal comfort is assessed by comparing the variability of the space temperature. Start-stop operations are used as a proxy for maintenance cost.

2 APPROACH

The project involved the development of computer-based models of control system components (state machine logic and pulse-width modulation logic), HVAC equipment, and building heat transfer, and the use of these models to study how an alternative method of starting and stopping equipment affects energy usage. This section describes the development of the models and their use in computer simulations.

2.1 Control Logic and Optimization

2.1.1 Pulse-Width Modulation Logic

Pulse-width modulation (PWM) is a technique for continuously modulating a process that has binary input states. It is commonly used for motion control problems involving DC motors and is sometimes used for controlling flow with solenoid valves. The input is switched rapidly between two states, and the fraction of the time that input is in the “on” state determines the fraction of input effort applied to the process.

For most PWM applications, the frequency of the PWM signal is high compared to the control bandwidth, so the PWM residual in the process output is very small. When using PWM signal with electro-mechanical devices such as compressors or fans, the PWM signal cannot be high because these devices cannot be cycled on and off frequently. Since the PWM frequency must be low, the PWM residual will be significant. Since the PWM frequency must be low, there is an opportunity to use the phase of the PWM signal to control the demand for power.

We developed capacity control logic that uses pulse-width modulation (PWM) and level-crossing logic. The PWM logic was designed so that the phase could be manipulated by a controller. The code was implemented in Matlab as an m-file. The syntax for Matlab m-files is similar to C.

Figures 2-1, 2-2, and 2-3 show examples of PWM signals that can be produced. The PWM signals in these figures have a PWM period of 30 minutes. The first figure shows a PWM signal with a phase of zero and a duty cycle of 25%. The second figure shows a PWM signal with a phase of 10 minutes and a duty cycle of 50%. The third figure shows a PWM signal with a phase of 20 minutes and a duty cycle of 50%. The third figure shows that when the phase gets sufficiently high, the signal “wraps around” so that the “end” of the pulse is at the beginning of the period and the beginning of the pulse is at the end of the period.

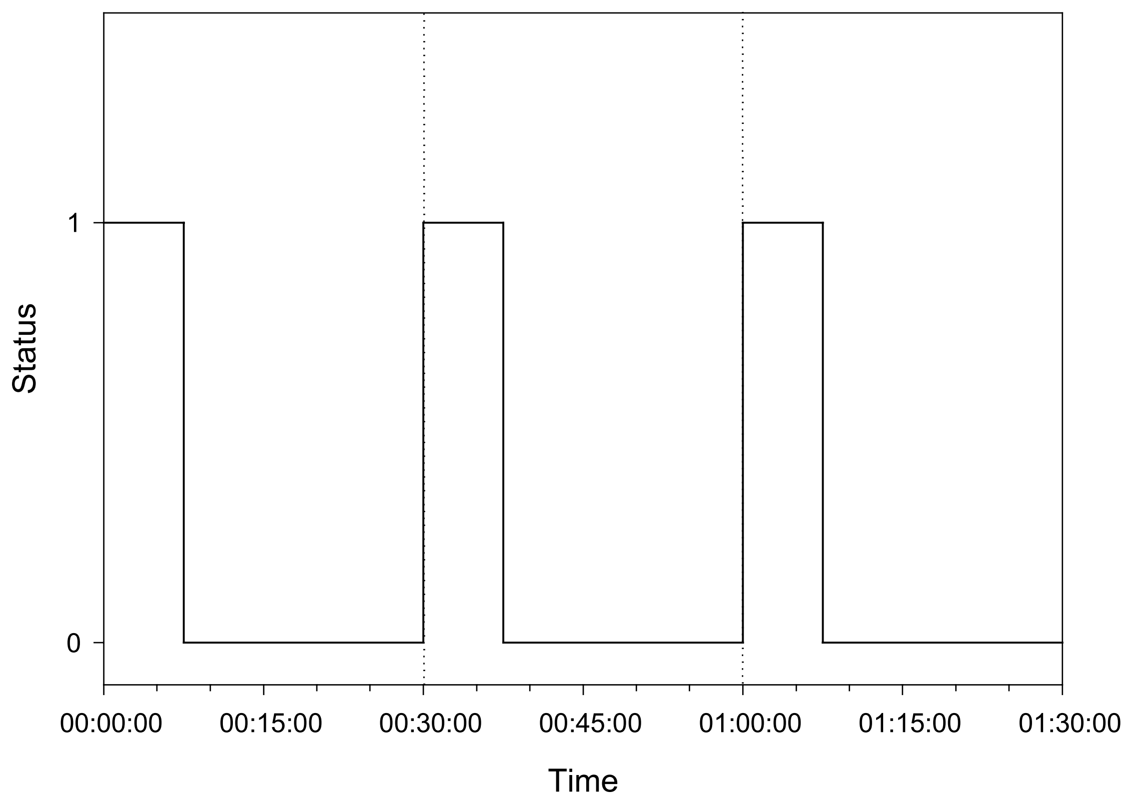

Figure 2-1: PWM signal with 30 minute period, 25% duty cycle and zero phase.

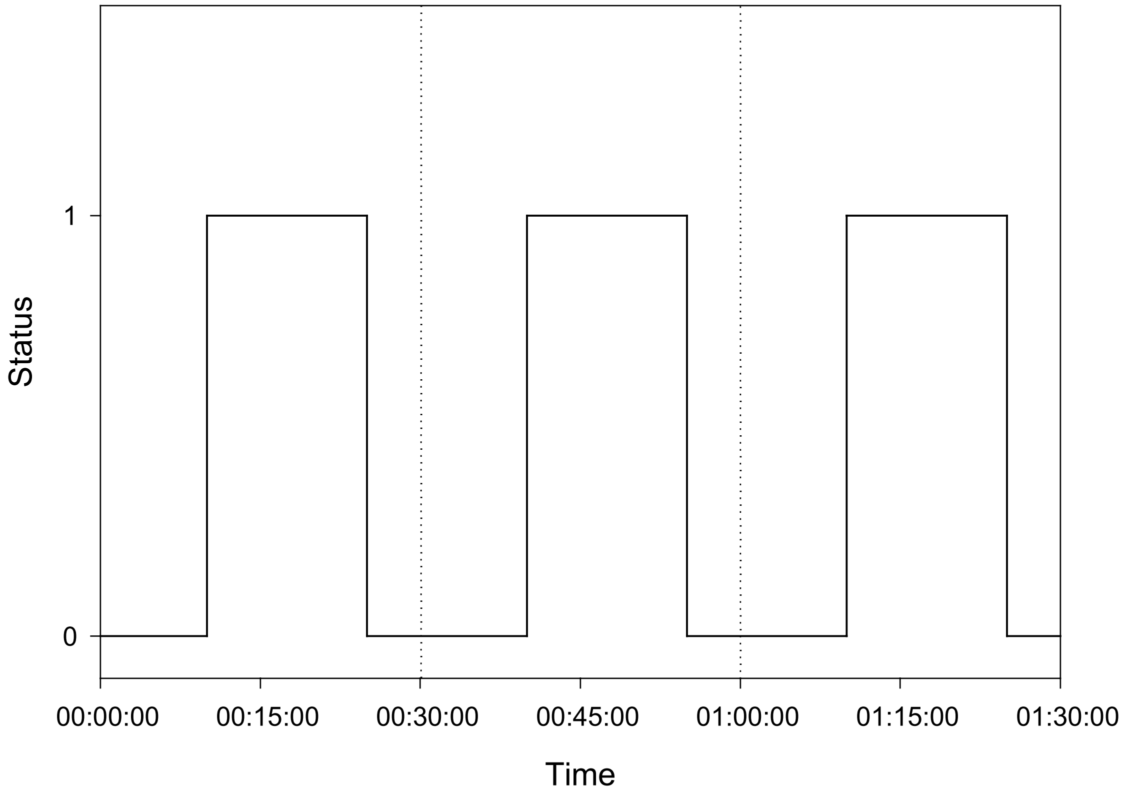

Figure 2-2: PWM signal with a period of 30 minutes, a duty cycle of 50%, and a phase of 10 minutes.

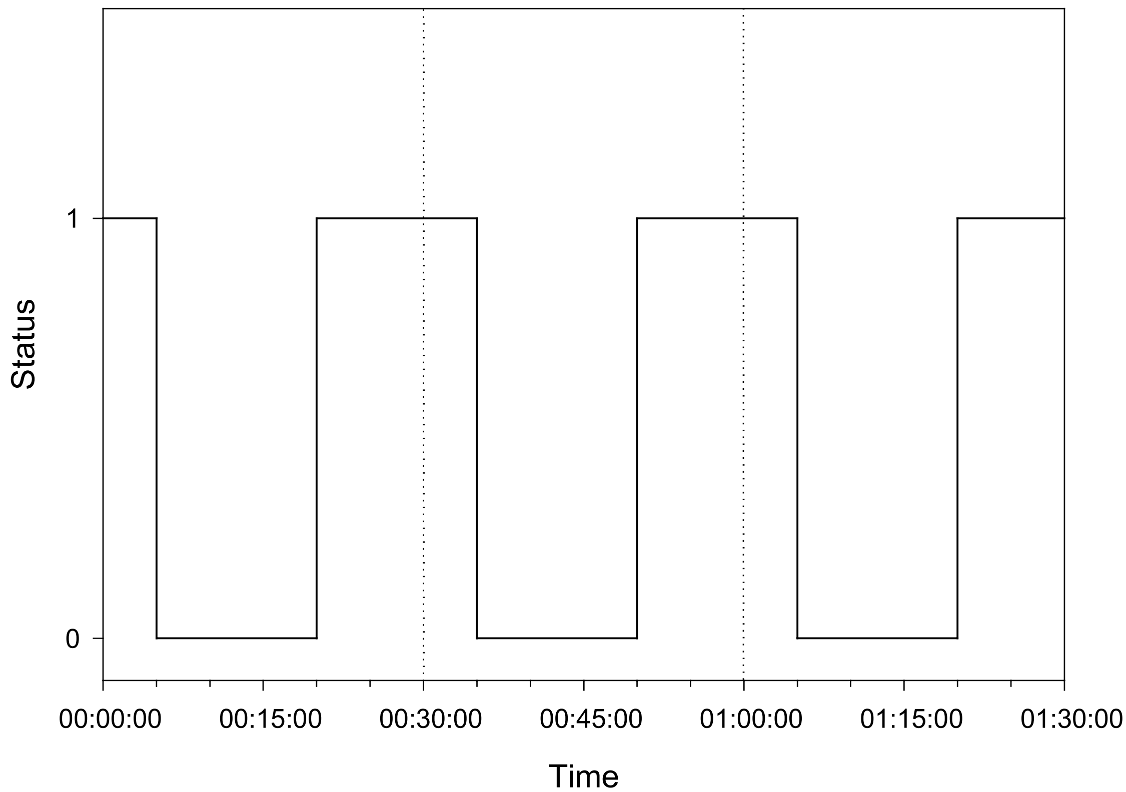

Figure 2-3: PWM signal with a period of 30 minutes, a duty cycle of 50%, and a phase of 20 minutes.

2.1.2 State Charts

We developed state charts and associated software so that we could simulate the operation of staged HVAC systems. This method of designing control logic for HVAC systems has been proposed by Seem (1999) for the operation air-handling controls. We chose to model the behavior of systems with two stages of mechanical cooling, an economizer, and four stages of heating based on personal interviews with representatives from HVAC control companies.

Figure 2-4 shows the state chart for the multi-stage HVAC system. The definitions of the states and the state transitions are shown in Table 2-1 and Table 2-2. Cool States 1 and 2 have all four heaters off and the outdoor air damper closed. The two cooling coils are turned on and off determined by the dutycycle from the PID controller and the state. In Cool State 2, the first air cooler is turned on at 100% dutycycle, while the second air cooler is turned on with a variable duty cycle from the PID controller. Cool State 1, keeps one of the air coolers completely off, while the other is turned on with a variable duty cycle.

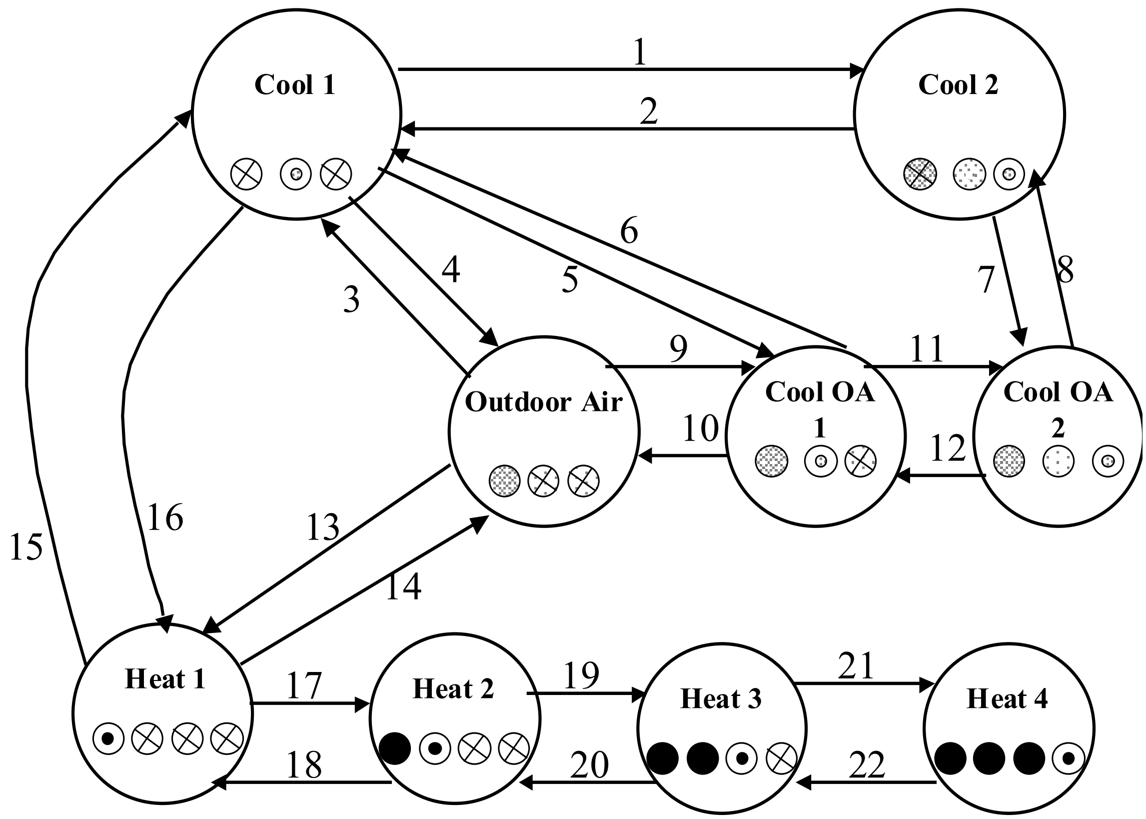

Figure 2-4: State transition diagram

Table 2-1: State definitions.

State	Equipment On
Cool 1	Compressor 1 cycling
Cool 2	Compressor 1 on, Compressor 2 cycling
Outdoor Air	Outdoor air damper open
Cool1 OA	Outdoor air damper open, Compressor 1 cycling
Cool2 OA	Outdoor air damper open, Compressor 1 on, Compressor 2 cycling
Heat 1	Heater 1 cycling
Heat 2	Heater 1 on, Heater 2 cycling
Heat 3	Heater 1 and 2 on, Heater 3 cycling
Heat 4	Heater 1, 2, and 3 on, Heater 4 cycling

Table 2-2: State transition definitions.

Number	Condition
1	Cool 1 saturated at 100%, Outdoor air above threshold
2	Cool 2 saturated at 0%, Outdoor air above threshold
3	Cool 1 saturated at 0%, Outdoor air temperature below threshold
4	Outdoor air is above threshold
5	Outdoor air is above threshold
6	Outdoor air is below threshold
7	Outdoor air is below threshold
8	Outdoor air is above threshold
9	Outdoor air damper fully open, Outdoor air is below threshold
10	Cool 1 saturated at 0%, Outdoor air is below threshold
11	Cool 1 saturated at 100%, Outdoor air is below threshold
12	Cool 2 saturated 0%, Outdoor air is below threshold
13	Outdoor air damper is fully open
14	Heat 1 saturated at 0%, Outdoor air is below threshold
15	Heat 1 saturated at 0%, Outdoor air is above threshold
16	Cool 1 saturated at 0%, Outdoor air temperature is below threshold
17	Heat 1 saturated at 100%
18	Heat 2 saturated at 0%
19	Heat 2 saturated at 100%
20	Heat 3 saturated at 0%
21	Heat 3 saturated at 100%
22	Heat 4 saturated at 0%

The outdoor air states work the same way as the cooling states, however they include the use of an outdoor air damper that allows outdoor air to enter and cool the building when the outdoor air temperature is low. Outdoor air states are enabled and disabled by a separate state machine.

Heating States 1-4 keep the air compressors off, the air damper closed, and control how many of the heaters are on or off. For each of the states, some of the heaters are either 100% on or off, while the PID controller starts and stops one of the heaters at a certain duty cycle determined every period.

The state transitions occur when the controller for the equipment in a certain state becomes saturated for three periods. A saturated controller will determine the duty cycle to be either 100% or 0%. If the controller is saturated at 100%, the transition will go to a state of higher power. If the controller is saturated at 0%, the transition will go to a state of lower power. The state, temperature, and outdoor temperature are monitored every fifteen minute period. Transitions can only occur when a controller has been saturated for two consecutive periods, thus giving some hysteresis to the system and limiting wear and tear on the equipment.

2.1.3 Demand Minimization by Coordination

A primary advantage of using PWM signals to start and stop equipment is that the phase of the PWM signal can be manipulated to improve the cumulative energy consumption characteristic of a number of systems. We developed a strategy for determining the optimal phase values every PWM period for a set of systems using PWM logic for starting and stopping equipment. That strategy is described below.

The optimal phase for reducing peak power was calculated using a grid search. The resolution of the grid corresponds to the time resolution of the computer simulation. The combination of phase values that minimizes peak demand for power is not unique, so we selected the phase values that minimize peak demand for power and that minimize the phase change from the previous PWM period.

2.1.4 Other Control System Components

In addition to the state machine logic and the pulse-width modulation logic, the system included a proportional plus integral (PI) controller and a moving average filter. The PI controller was tuned by first selecting the controller gains using the Ziegler-Nichols step response method and then fine-tuning the gains by hand.

A moving-average filter was added to the feedback loop to eliminate the residual from the PWM signal. When using PWM signals to control electro-mechanical devices such as compressors or fans the cycling frequency must be low or the equipment may be damaged. This means that there is always a significant residual in the controlled variable due to the operation of the PWM function. A moving average filter with a filter length equal to the PWM period will exactly cancel the residual. Canceling the PWM residual will ensure that the PI controller does not respond to the PWM residual. It also makes it easier to tune the PI controller.

2.2 Computer Simulations

2.2.1 HVAC System and Building Model

In order to test the control logic, we developed a mathematical model of the heat transfer dynamics of a section of a building, which includes and HVAC system and the conditioned space.

The HVAC system model consists of equations for the heat and mass transport through an air distribution system and equations describing the behavior of heating and cooling equipment. Figure 2-5 shows a schematic diagram of the HVAC system. The system consists of five ducts for outdoor air, supply air, return air, exhaust air, and recirculation. The system has a supply fan and a return fan. Heating and cooling is provided in the supply duct.

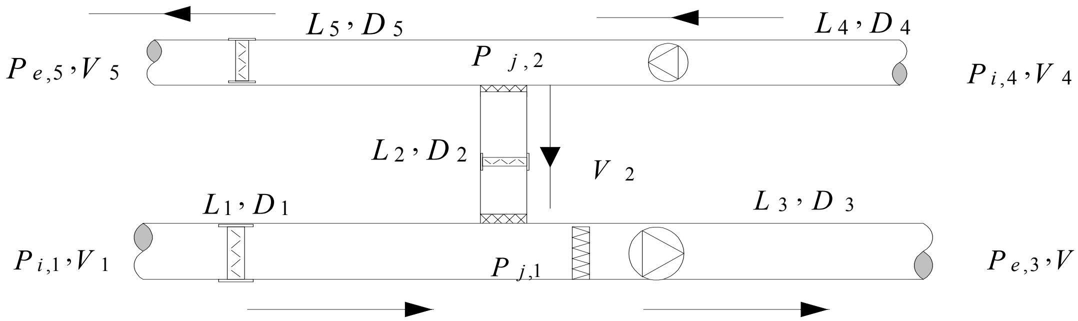

Figure 2-5: Schematic diagram of HVAC system

The transient behavior of the system includes the energy transport throughout the distribution system and the momentum transport (acceleration and deceleration of air and fan wheels). The momentum transport is important because the settling time of the air velocities after starting or stopping the fans is comparable to the minimum on-time and minimum off-time for compressors (1-5 minutes).

The momentum equations are as follows:

$$M\dot{V} = -F\langle V|V|\rangle - K\langle V|V|\rangle + A_f P_f + A_e P_e + A_j P_j \tag{1}$$

In Equation 1, M is a diagonal mass matrix, V is a vector of velocities in each branch, F is a matrix containing the friction factors in each branch of the air distribution system, K is a matrix containing the minor loss coefficients, A_f is a matrix of cross-sectional areas at each fan location, P_f is the static pressure across each fan, A_e is the cross-sectional duct areas at the points where the system connects of external points such as the outdoors or indoor spaces, P_e are the external pressures (outdoors or indoors), A_j is a matrix describing the areas at the junctions of the air distribution system, and P_j is a vector of pressures at each junction. The quantity $\langle V|V|\rangle$ is a vector whose elements are the product of the velocity times its magnitude.

At each junction there is a mass continuity constraint that can be expressed as follows:

$$-A_j^T V = 0 \tag{2}$$

The pressure at each junction can be computed explicitly by differentiating the constraint equation and substituting the state equation. The result is as follows:

$$P_j = \left(A_j^T M^{-1} A_j\right)^{-1} A_j^T M^{-1}\left[(F+K)\langle V|V|\rangle - A_f P_f - A_e P_e\right] \tag{3}$$

For the system shown in Figure 2-5, the system coefficients are as follows. The mass matrix, M, is a diagonal matrix where $M_{i,i} = \rho A_i L_i$. The duct friction matrix, F, is a diagonal matrix where $F_{i,i} = \pi D_i L_i \rho f_i / 8$. $\langle V|V|\rangle$ is a vector where the ith element is $V_i|V_i|$. The minor loss matrix is a diagonal matrix where $K_{i,i} = A_i \rho K_i / 2$. The joint pressure vector is $P_j = [P_{j,1} \quad P_{j,2}]^T$. The external pressure vector is $P_e = [P_{i,1} \quad P_{o,3} \quad P_{i,4} \quad P_{o,5}]^T$. The fan pressure vector is $P_f = [P_{f,3} \quad P_{f,4}]^T$. The joint area matrix, the external pressure area matrix, and the fan area matrix are as follows:

$$A_j = \begin{bmatrix} -A_1 & 0 \\ -A_2 & A_2 \\ A_3 & 0 \\ 0 & -A_4 \\ 0 & A_5 \end{bmatrix} \qquad A_e = \begin{bmatrix} A_1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & A_3 & 0 & 0 \\ 0 & 0 & A_4 & 0 \\ 0 & 0 & 0 & A_5 \end{bmatrix} \qquad A_f = \begin{bmatrix} 0 & 0 \\ 0 & 0 \\ A_3 & 0 \\ 0 & A_4 \\ 0 & 0 \end{bmatrix}$$

Equations 1-3 describe the momentum equations for the air in the HVAC ducts. The fan wheels also have a significant amount of momentum that affects the dynamic behavior. For each fan, the momentum equation is as follows:

$$J\dot{\omega} = \tau_m - \tau_a \tag{4}$$

where J is the moment of inertia of the fan wheel, ω is the speed of the fan wheel, τ_m is the torque exerted by the motor on the fan wheel, and τ_a is the torque exerted by the air on the fan wheel. We modeled an AC motor with the following torque-speed relationship:

$$\tau_m = \begin{cases} \tau_{\max}, & \omega \le \omega_L \\ \dfrac{\tau_{\max}}{\omega_L - \omega_r} + \tau_{\max} \dfrac{\omega_r}{\omega_r - \omega_L}, & \omega \ge \omega_L \end{cases} \tag{5}$$

The torque-speed characteristic is constant when the speed is below ω_L. Above ω_L the torque is proportional to the speed, decreasing with increasing speed. The torque-speed characteristics of real AC motors are more complex than predicted by this model, but this model captures the most relevant features for this application.

The momentum transport equations use standard, steady-state fan models to relate fan speed and air velocity to the static pressure across the fan wheel and the efficiency of the fan. We use a model that has been used for in other simulation environments such as HVACSIM+ (Park et al., 1986). The non-dimensional pressure coefficient is a polynomial function of the non-dimensional speed, and the efficiency is also a polynomial function of non-dimensional speed. Outside of the normal operating range, we use extrapolation functions proposed by Haves and Norford (1998). The non-dimensional pressure is a quadratic function of the non-dimensional speed, and the efficiency is constant.

We model the dynamic behavior of the heating and cooling components by assuming the coefficient of performance has a first-order transient response characteristic. We used time constants of 1.5 minutes for both heating and cooling. These values were derived from the experiments reported by Mulroy (1986).

Energy transport in ducts was modeled by assuming perfect mixing in each branch of the distribution system. The energy transport equations have velocity variables that have only a positive sign that is dependent on the sign of the velocities in the branches. This technique makes it easy to model the energy transport even if the flow in one or more branches reverses direction.

The building model includes heat accumulation by walls, indoor air, and furniture, and the dynamic behavior of the space temperature sensor. Time-dependent internal loads and external temperatures were used to simulate real disturbances. The building section was modeled as a single room with one exterior wall.

The wall was modeled as a three-layer slab. The heat accumulation for a layer is as follows:

$$\rho_l \, V_l C_l \dot{T}_l = h_l \, A_l \left(-T_l + T_{a1} + T_{a2} \right) \tag{6}$$

where ρ_l is the density of the layer, V_l is the volume of the layer, C_l is the specific heat of the layer, T_l is the temperature of the layer , h_l is the heat transfer coefficient, A_l is the surface area of the layer, and T_a is the temperatures with which the layer exchanges heat. The values of T_a may come from adjacent layers or the indoor or outdoor temperatures. The heat transfer coefficients may differ from one layer to another and from one surface to another.

Furniture and other indoor furnishings accumulate a significant amount of heat. The model we used assumes that all of the heat transfer with the furniture is by convection with the indoor air. The equation for the heat accumulation of the furniture is as follows:

$$\rho_f \, V_f C_f \dot{T}_f = h_f \, A_f \left(-T_f + T_a \right) \tag{7}$$

where T_a is the indoor air temperature.

The heat accumulation of the air in the space is as follows:

$$\rho_a \, V_a C_{p,a} \dot{T}_a = f C_{p,a} \left(T_s - T_a \right) - h_f \, A_f \left(-T_f + T_a \right) - h_w \, A_w \left(-T_w + T_a \right) \tag{8}$$

Only sensible heat transfer was modeled. The first term on the right-hand side of Equation 8 is advection due to air supplied by and exhausted to the HVAC system. The second term is heat transfer between the furniture and the air. The third term is heat transfer between the inner wall layer and the air.

The reading from the space temperature sensor normally lags the actual temperature by several minutes. We modeled this lag using as a first-order, linear, time-invariant transfer function with a time constant of 7.5 minutes.

2.2.2 Numerical Integration Details

We used Matlab to simulate the dynamic behavior of the system. Matlab has several built-in numerical integration methods. We used ode45(), which is a fourth-order Runge-Kutta algorithm with adaptive step size control. Runge-Kutta is an explicit numerical integration method, so it is slow when used with stiff systems. Our model contains a combination of relatively fast dynamics associated with momentum transport and equipment behavior, and relatively slow dynamics associated with building heat transfer. The result was that the simulations ran slowly, about 10 times as fast as real time.

The ode45() function has a feature that enables the solver to integrate from event-to-event. We used this feature to force the solver to integrate up to each starting or stopping event. We also used it to produce periodic outputs of important variables.

The control logic code and system model code was designed as modules representing distinct components of the system. This modularity makes it easy to test and modify component models. Each component was an m-file. For the HVAC system and building dynamics components, the m-file returned the differential values of the state variables associated with that component. A separate file was used to assemble the entire state equation and pass it to the ODE solver.

2.2.3 Benchmarks and Performance Metrics

As a performance benchmark, we used PWM signals with constant-delayed phase. The constant-delayed phase signals are delayed by a uniform fraction of the PWM period. For example, if there were three systems operating with a period of 30 minutes, then the first system of a constant-delayed set would have a phase of zero minutes, the second system would have a phase of 10 minutes, and the third system would be operated with a phase of 20 minutes. This strategy is extremely simple, and minimizes peak demand under some, but not all, conditions.

Simple thermostats use level-crossing logic for starting and stopping equipment, so we included level-crossing logic as a benchmark. To avoid rapid cycling caused by noise or high-frequency disturbances, the level-crossing logic contained a differential. We defined the setpoint as the average of the upper and lower level, and the differential as the magnitude of the difference between the setpoint and one of the levels.

We used five metrics to compare the performance of the three start-stop methods. The first is the mean absolute deviation of the space temperature from setpoint. This metric is a standard control performance metric, and is a proxy for thermal comfort, since large temperature swings from a desired temperature should be associated with discomfort. We used the number of start and stop events per hour as a proxy for maintenance cost. If the controls force the units to start and stop more frequently they will fail sooner. We used average power, the standard deviation of the power, and the peak demand for power as energy performance metrics.

2.2.4 Simulation Conditions

We simulated one week of wintertime operation of three HVAC systems conditioning three independent spaces. Each space had 1500 square feet of floor area. Typical meteorological year (TMY) data from Sacramento, California were used to simulate outdoor conditions. Figure 2-6 shows the outdoor temperatures used for the simulation. Internal loads were simulated using the profiles shown in Figure 2-7. Each system operated at a different setpoint. The values we used were 21.9 °C, 22.4 °C, and 23.1 °C for systems 1-3, respectively. For the simulations using PWM logic, the PWM period was 30 minutes. The level-crossing logic had a differential of 1 °C.

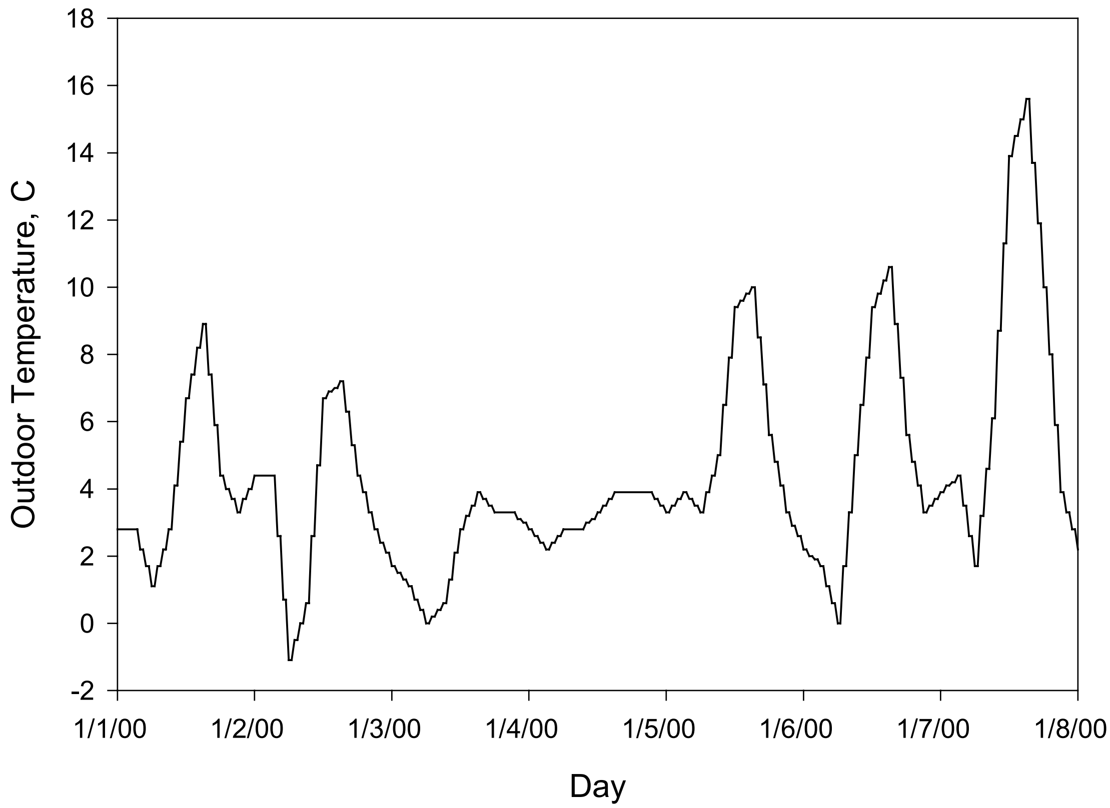

Figure 2-6: Weather data.

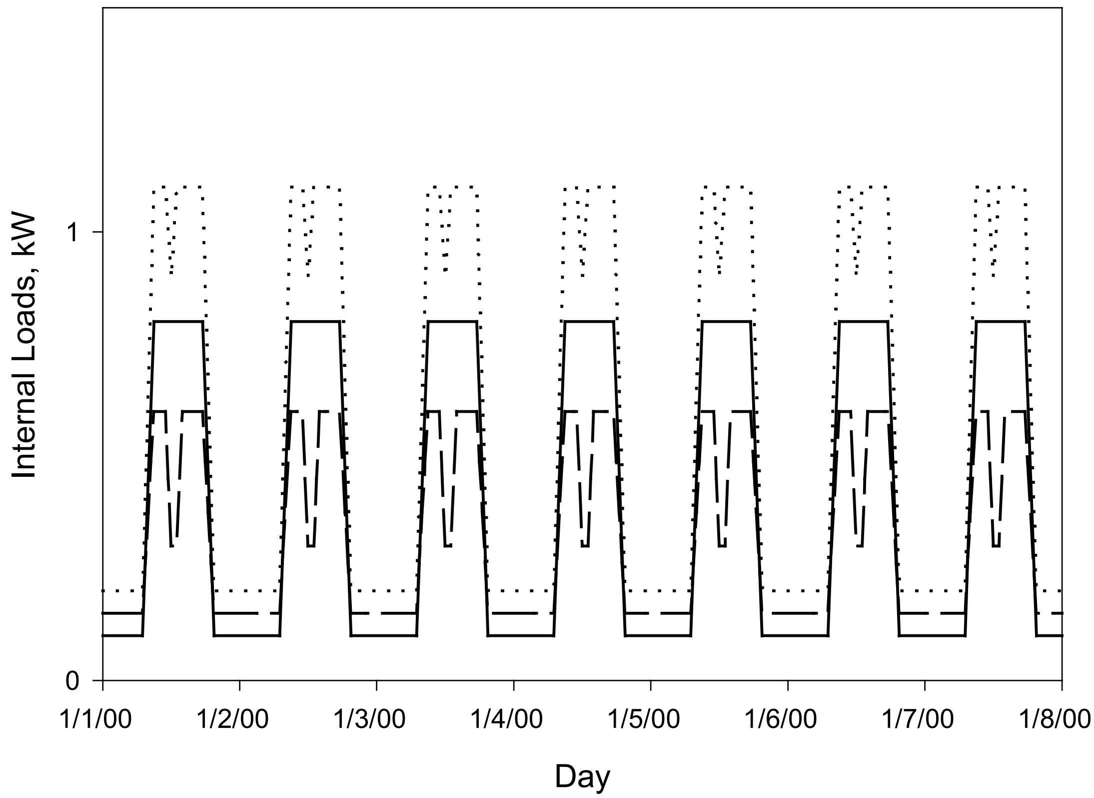

Figure 2-7: Internal loads.

3 RESULTS

We used the component models described above to compare the energy performance of systems that use pulse-width modulation as a means to start and stop equipment with the energy performance of systems that use level-crossing logic (e.g., a common thermostat). The basis of comparison was energy consumption, peak demand for power, deviation of space temperature from setpoint, and the number of start-stop operations.

Table 3-1 shows how the three start-stop strategies compare on the basis of the five performance metrics. The first metric is the mean absolute deviation of the temperatures of the three systems from the setpoint. The second metric is the total number of starts and stops of the three systems during the simulation. The third metric is the total energy consumption of the set of three systems. The fourth metric is the standard deviation of the power. It is computed by averaging the variances of the three systems for each start-stop method. The fifth metric is the peak demand of the set of three systems.

Table 3-1: Performance of the three start-stop methods.

	Optimized phase	Constant-Delayed Phase	Level-Crossing
MAD of error, C	1.44	1.39	1.30
# starts and stops per hour	2.15	2.38	1.69
Average Power, kW	36.1	38.8	37.2
Std of Power, kW	2.3	10.5	12.4
Peak Power, kW	40.0	49.8	44.5

Although Table 3-1 indicates that the optimal-phase strategy uses less energy than the other strategies, the differences are small. The table shows that the optimal-phase strategy used 7% less energy than the constant-delay phase strategy and 3% less energy than the level-crossing strategy. Considering the large variation in the power consumption of the constant-delay phase and level-crossing strategies, these differences are not considered significant.

Table 3-1 shows that the optimized phase strategy reduces the peak demand for power by 10% in comparison to the level-crossing logic and by 20% in comparison to the constant-delayed phase strategy. The table also shows that the reduction in peak demand is achieved by reducing the variance of the total power consumption. The standard deviation of the power consumption of the optimal phase strategy is just 19% of the standard deviation of the power consumption of the level-crossing strategy and just 22% of the power consumption of the constant-delayed phase strategy. Figure 3-1 shows the power consumption of the three strategies during the simulation period. The reduced variation and reduced peak demand of the optimal-phase strategy is evident from the graph.

The coordinator increases the mean absolute deviation of the temperature and the number of start stop operations slightly in order to reduce peak demand and reduce the variance of the power consumption. Relative to the level-crossing logic, the coordinator increases the mean absolute deviation of the control error by 11% and increases the number of starts and stops per hour by 27%.

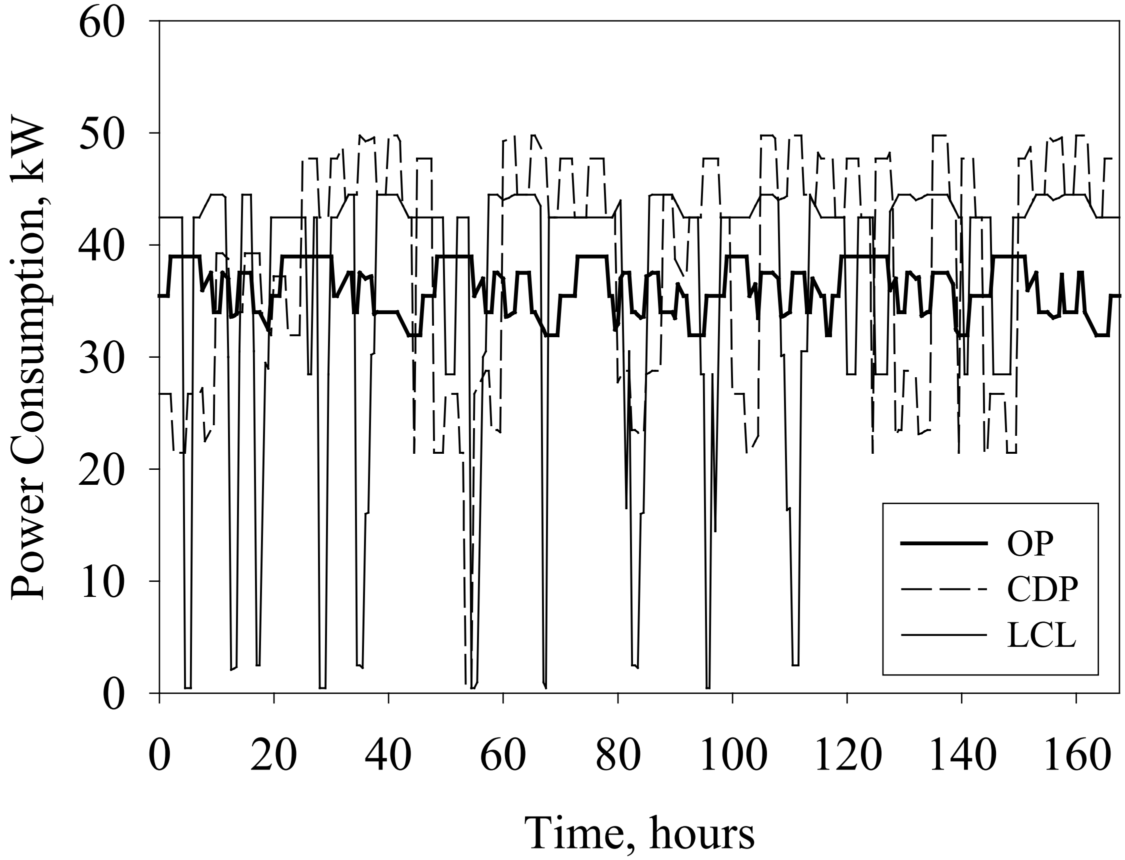

Figure 3-1: Power consumption comparison.

4 CONCLUSIONS

The results of Section 3 indicate that the new control strategy can outperform level-crossing logic on the basis of energy performance. When the PWM logic and state machine were combined with an optimized coordinator, the new strategy reduced peak demand by 20% relative to level-crossing logic.

The results also demonstrated that there is a comfort and maintenance penalty for the improved energy performance. The new strategy increased the mean absolute deviation of the space temperature from the setpoint by 11% relative to the level-crossing logic. This was only 0.14 °C, so that penalty may not cancel the energy benefit. However, the new strategy also increased the number of start-stop operations by 27% relative to the level-crossing logic. If the number of start-stop operations is proportional to the mean time to failure, then this penalty may be significant.

The new control strategy has the potential to reduce energy costs for end users. Commercialization would involve primarily control software development. The existing code would have to be adapted to a particular platform, but no hardware would be required as long as there was an existing control communication system in place so that a coordinator running on a networked computer could supervise a number of HVAC control units.

Additional work is needed to investigate whether or not it is possible to maintain the energy benefits of the new strategy while reducing the maintenance penalty. It is possible that by operating the PWM signals asynchronously and at different frequencies the maintenance penalty could be reduced. Asynchronous operation may also yield additional energy benefits. Asynchronous operation would increase the complexity of the design of the control logic, though not necessarily the complexity of implementing it.

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