Application of Direct Digital Temperature Control Systems for Maximum System Efficiency on VAV Systems

Frank W. Mayhew, EC/TC Consulting

The acceptance of direct digital temperature control technology by the HVAC industry has exceeded the projections of most manufacturers in the industry. Unfortunately, the application engineering of these systems has not been applied to their greatest advantage. Many of the control sequences are still applied as if pneumatic controls were being used. Open-loop and indirect control loops are still employed for many of the standard control functions that can now have closed-loop control. Integration between the major sub-systems has not been implemented to its full potential. This paper describes temperature control sequences that can be implemented with direct digital control that were not practical with discrete component pneumatic and electronic control systems. The purpose of these sequences is to operate the HVAC system with:

(1) The greatest comfort possible, within basic design.

(2) At the lowest energy cost while taking advantage of diversity of loads.

(3) Closing open control loops and indirect control loops

(4) Eliminating most of the energy waste of over-design, and correcting for some under-design of systems by taking advantage of diversity.

INTRODUCTION

Variable Air Volume (VAV) HVAC systems, by design, include taking advantage of diversity of loads in various parts of the building, while lowering the basic cost of transportation of the conditioned air and reducing peak loads. However, poorly applied temperature controls can negate much of the potential saving of VAV systems in commercial buildings. Temperature controls, for VAV systems, have traditionally been applied in sub-systems using discrete component pneumatic or electric control components. Mental inertia is one of the reasons that advanced control systems are not used and advanced control strategies are not implemented. A Direct digital control (DDC) provides the opportunity for the integration of sub-systems that work in direct response to load changes to eliminate many of the inefficiencies in HVAC system operation.

The traditional implementation of pneumatic or electronic controls would be to install controls in three basic sub-systems—zones, air handlers and the central plant—with little or no, integration between them. Feedback signals from the zones to the air handlers may be attempted, but is usually abandoned or negated when someone sets one of the thermostats up or down too far. Thermostats are installed in ‘‘representative zones’’ to control morning warm-up, cool-down and night set-back strategies, but otherwise there is no further attempt at integration of the sub-systems. Chillers, boilers, and pumps are controlled by their own individual controls and started and stopped with a time clock and outdoor air thermostat.

Typical morning warm-up for VAV systems is a crude, brute force control sequence that forces open all the VAV boxes in the hope that enough return air will be circulated from the warming exterior zones to the cooling-only interior zones to adequately warm them for occupancy.

Because of the basic limitations on discrete component controls, open-loop control (without direct feedback from the controlled variable) is often provided using outside air temperature to reset supply air temperatures, mixed air temperatures and hot water and chilled-water temperatures. At best, open-loop controls are a compromise for both energy and comfort, though they are better than nothing. Hot-water and chilled-water lockouts based on outside air temperature require one to start systems too early and keep them running too long.

Indirect control loops (control loops that control a proxy for the actual load) are found in almost every building where variable-flow water and air systems are employed, contributing to inefficiencies in building operation. Supply fan static pressure is controlled by a pressure sensor located ‘‘⅔ of the way down the duct’’ and usually set much higher than the VAV boxes require for proper operation. Differential pressure sensors for secondary chilled-water pump control are placed between the supply and return, near the end of the piping loop, or at the secondary pump manifold, requiring set-points that provide little or no turndown by the pump.

Background

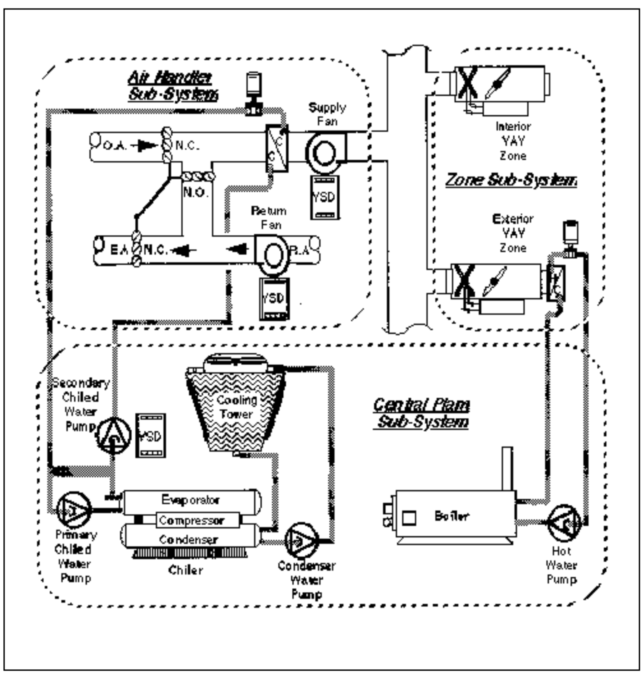
The mechanical system designer is faced with the battle between designing a comfortable, energy-efficient building and the capital costs of providing a truly energy-efficient design that still can produce comfort within the bounds of the selected mechanical systems. Rarely is there a good life-cycle cost analysis done before significant reductions are made in the budget. The ripple effect is passed on to: the tenants who pay higher operating costs; the utility which cannot build additional generation capacity except at great cost; and, eventually to the environment that must absorb the additional 1.3 to 2.5 pounds per year of CO_2 per kWh consumed, if fossil-based fuels are used (Fischer, et al. 1992).

Scope

A ''typical'' office occupancy is examined to demonstrate how an integrated control system can save energy while furnishing design comfort. The typical office used will be a multistory building with multiple tenants. As with the majority of large office buildings, the building is air conditioned using variable-air-volume, cooling only for the interior zones, with re-heat for the exterior zones, a central air handler, a centrifugal chiller and a central boiler (Figure 1). The chilled-water system employs primary/secondary pumping. The hot water system is variable flow and variable temperature. Fans and secondary chilled-water pumps are provided with variable-speed drives to control speed.

Direct digital control provides the first real opportunity to integrate all of the sub-systems and to modify control sequences to take advantage of diversity in part-load performance. The following are descriptions of how the various control loops would be controlled if discrete component controls were used and how the control loops would be modified with direct digital control.

***Figure 1.** Sub-Systems of a Typical Building Air Conditioning System.*



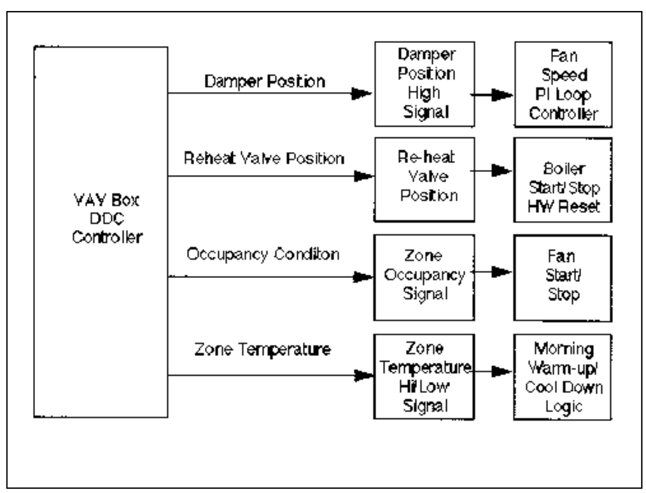
ZONE CONTROL

The zone control is really the most important element in any temperature control system. The zone, when within the range of the zone controller, provides comfort. The air-handlers and central plant only provide the source and conversion of the energy for HVAC, the zone is the user. If the air-handler and chiller are not properly controlled, the zone will either not be able to provide comfort, or will waste energy by throttling back. With this as the basic premise let's look at the various control functions required for the building and how DDC controls can improve performance. Zone controls can provide an immense amount of information that can be used as feed-back for integrated control strategies. (Figure 2.)

Morning Warm-up

Under this category falls optimum start control and morning warm-up and/or morning cool-down control. Cool-down is similar to warm-up, but with cooling strategies instead of heating. Some areas of the country need only warm-up, while warmer climates may need both. The intent is to have

***Figure 2.** Basic Control Block Diagram for Zone Feedback to Fans and Central Plant.*



the building at a comfortable temperature in the morning at occupancy. Since the sample building is of multiple tenancy, the fans are started early enough for the first tenant and the entire building is operated, whether occupied or not. A microprocessor-based calendar-clock is typically used for this purpose. In a pneumatically controlled building the warm-up routine is controlled by a thermostat located in a ‘‘representative’’ exterior zone that, when the temperature is below its setpoint, starts the fan system with the outside air dampers closed, the chiller off, the boiler and hot water pump on and all the interior VAV zone boxes forced wide open to allow the return air to warm up the interior zones. The building is considered to be warmed up when the return air temperature is more than 68°F to 70°F. The ‘‘representative’’ zone is often chosen during construction and may not be ‘‘representative’’ at all, complicating the process.

This method of morning warm-up is inadequate in several ways. The intent of such a system is to move the warm air from the exterior zones that have re-heat to the interior zones via the re-circulation of return air. It takes time for the exterior zones to warm up the return air enough to provide any significant heating to the interior zones and, if the interior zones are already at a satisfactory temperature, this method may even sub-cool the interior zones resulting in poor comfort at occupancy time.

During morning warm-up, the supply fan cannot adequately serve all the zones if the boxes are all wide open, because a fan for a VAV system is designed for diversity of zones and is not designed to serve all zones with full flow simultaneously. With reduced airflow, diffuser velocities are reduced, causing the air to short-circuit to the ceiling return grilles, not providing any significant heating to the space. The fan may not even be able to get air to the zones farthest from the fan because the zones closest to the fan use all the air, and starve the rest of the system.

A return air thermostat may not be representative of whether the building has warmed up sufficiently due to the short-circuit of airflow in the exterior zones and the averaging of the air from many zones together. Energy is wasted by operating the fan at excess volume and, because the morning warm-up is flawed, the operators will invariably extend the time of the morning warm-up, starting earlier and earlier until the complaints stop, thus wasting additional energy without doing any real work.

A direct digital control system offers the opportunity to warm up the building using temperature sensors in all the zones of the building. A calculation program is still used to determine the optimum time to start warm-up, but there are significant differences from the discrete component control method. (1) If a zone does not require morning warm-up, that zone stays out of the warm-up mode. (2) Interior zones are not placed in the warm-up mode until the supply air is warm enough to heat them. (3) Interior zones, instead of being forced open, are instructed by the system to change control action from direct acting to reverse acting, or vice versa so that these zones act like a heating terminal. The minimum CFM is set to zero, and will shut down the air flow when the zone is warm enough, and, (4) the set points for the zone thermostats are all shifted down to 68°F to prevent overheating of the zones. This strategy reduces the fan load and allows more fan capacity to warm up the coldest zones, while not using fan energy to try to warm up zones that are already warm enough for occupancy. The length of time for morning warm-up can be reduced significantly.

Morning warm up works better if there is a small heating coil in the main supply duct to warm up the air for the interior zones. Attempting to warm up the interior zones with fan energy added the return air from the exterior zones wastes energy since the fan energy by itself does not significantly raise the supply air temperature and warm-up will take longer.

Once the building has been warmed up, those interior zones that have a later occupancy can be turned off to save fan energy, or placed in a standby mode. Warm-up for exterior zones of the building can be delayed until the system calculates the need for morning warm-up based on their actual occupancy, not the occupancy of the earliest tenant.

Boiler Start/Stop

In the conventional control system the boiler is stopped based on time-of-day and/or if the outside air temperature is over some fixed temperature, usually 70°F, or higher. Once a building is warmed up, and the outside temperature is above the thermal balance point of the building, the boiler can be shut off based on all the zone re-heat valves being closed. Typical building thermal balance points are between 50°F and 60°F. With a large number of hours between 50°F and 70°F, running the heating system longer amounts to a considerable amount of energy. The feedback from the zone re-heat valves will also allow automatic starting of the boiler and pump if the need for heat should arise. Note that if the minimum CFM’s on exterior zone boxes are set too high, the boiler and hot water pump run-time will be extended up to 85°F in some cases observed by the author.

Heating Hot Water Control

Most discrete component control systems employ some form of hot water reset based on outside air temperature in order to approximate the building load and to improve the control response of the re-heat hot water valves and reduce heat loss through the insulation of the circulating system. With an integrated control system, the hot water reset can be locked out during morning warm-up to speed up the warm-up process. During normal occupancy hours the hot water reset can be reinstated using the re-heat valve position instead of outside air temperature as the reset input. This will provide the lowest temperature hot water needed for the re-heat valve calling for the most heating to satisfy the space conditions.

Because two-way valves are used on the heating coils and the hot water temperature is varied during occupied hours, it is probably not cost effective to use variable-speed pumping because as the temperature of the hot water is lowered, the volume goes up to provide the same quantity of heating energy. With feedback from the zones, the operating time on the hot water pump will be minimal except in colder climates where a variable speed pumping system may be of value.

Occupancy Control

One of the greatest opportunities for energy saving is missed with discrete component control systems by not instituting some form of occupancy control. The lighting control side of many buildings now includes occupancy sensors to turn off the lighting in unoccupied spaces as an energy conserving measure. The potential savings in a VAV system, with occupancy control, include fan, chiller and boiler energy. With intelligent control at the zone level, there are several opportunities to implement occupancy-based control systems.

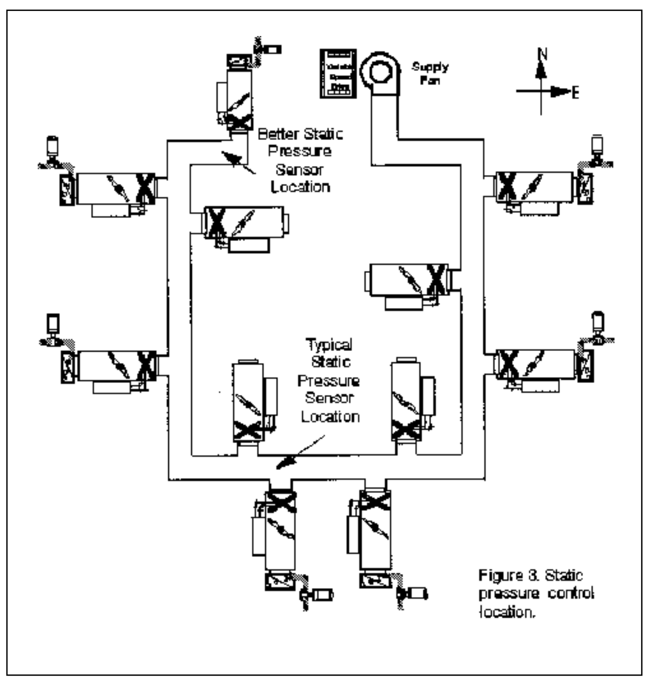
The first case is the casually unoccupied zone. A simple interface between the lighting occupancy sensors (if used) and the zone controls can make a significant contribution to energy conservation. If you take the average breaks that an office worker uses, lunch, coffee breaks and trips to the rest rooms there is at least 1.5 hours, out of a nine hour occupancy period, that the office is not occupied. Just the lunch hour alone is 11% of the nine hour day. Add to this meetings and absences for sick days and it is easy to reach 20% vacancy factor in most occupancies during the workday. Auxiliary contacts are available on some low-voltage lighting relays that can be connected to the VAV box controller to force the box closed when unoccupied. Where several lighting zones are combined with one HVAC zone, the contacts can be wired in series.

The second case is when a building is occupied by multiple tenants with varying time schedules. This can be a single company with various departments, or multiple companies. It is rare that the design includes control sequences with the ability to handle after-hour occupancy. When a building operator is faced with one tenant or department that works later than the rest, the systems are generally operated to condition the whole area that is served by the system rather than install and control isolation dampers or other automatic means to serve only the occupied area. With the control of all of the zone boxes, occupancy control can be achieved easily by manual entry into the central terminal, by push-button overrides that give a predetermined number of hours, or minutes of override, or by a telephone interface system that allows occupants to access overrides by a telephone override system. Telephone based systems allow the most flexibility, as they allow the tenant to schedule overrides in advance, and the building owner can receive printouts of override activity to use in a revenue-generating program where billing for after-hours HVAC use is included in the lease.

SUPPLY FAN STATIC PRESSURE CONTROL

Static pressure control of supply fans for variable-air-volume systems is probably one of the more misunderstood control loops in an HVAC system. The common practice of placing the static pressure sensor *''2⁄3 of the way down the duct ''* is not an energy-efficient method of control since the control set points tend to be set at 1.0 to 2.0 inches w.c. (Figure 3.). The higher the static pressure control point, the less total fan static is available for turndown. In high rise buildings with central fan systems, the problem is exacerbated by the practice of placing the static pressure sensor in the main duct serving the floors. When you have a fan design with 4.0 in. w.c. and a control point of 2.0 in. w.c., the fan does not have the full opportunity to take advantage of fan affinity

Figure 3. *Static Pressure Control Location.*



laws. The lower the control pressure, the closer the supply fan energy will follow the fan cube law. The return fan, for instance, already closely approaches the fan cube law because the relative control point is only 0.05 in. w.c., the building pressurization. The 1995 ASHRAE Applications Handbook states that the static pressure sensor should be located ‘‘at 75% to 100% of the distance between the first and last terminal’’ and that ‘‘the pressure selected provides the minimum static pressure to all air terminal units during all supply fan design conditions.’’

The proof is in the formulas that determines the static pressure at the fan and the brake fan horsepower:

$$S_{Fan} = CFM^2 \cdot (S_{Des} - S_{Conti}) + S_{Contr}$$

Where

S_{Fan} = Static of the fan

S_{Des} = Design static of fan

S_{Contr} = Static Pressure Control Point

$$BHP_{fan} = \left(\frac{CFM \cdot \text{Static Pressure}_{Fan}}{Eff_{Fan} \cdot 6359}\right)$$

Note that the static pressure control point directly affects the brake horsepower of the fan.

Zone Control of Static Pressure

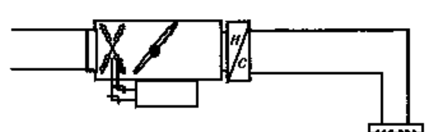
The variability and diversity of the CFM demand of a variable-air-volume system is determined by solar exposure, outside air temperature and interior load. All these factors in the load vary throughout the operating day. As the sun moves around the building, the primary load moves from the east to the south and then to the west. Cool mornings become warm afternoons and people come and go in the interior spaces, turning lights and equipment on and off, and opening and closing window coverings. The result is that it is impractical to find any single, or even multiple locations for static pressure sensing in the ducts that will be right for all loads. In addition the single setpoint control loop is really an indirect control loop, just like outdoor reset of hot water temperature. The actual load is the CFM that the VAV box must provide to the zone. There must be enough pressure in the duct to see that every VAV box in the system can provide the required CFM to the zone for the load at that time.

Integrated direct digital control provides a better method of controlling fan capacity (Englander and Norford, 1992, Hartman, ASHRAE, 1992). The problem is best shown with the duct arrangement in Figure 3. Where is the best place for the static pressure sensor? At the end of the duct run allows for the lowest set point for the static pressure control, but the most efficient method of static pressure control is using the VAV box damper position to feedback to the variable speed drive control.

But what do pressure independent VAV boxes really require (Figure 4.)? And when? The manufacturers of VAV boxes usually rate the boxes at maximum CFM including an allowance for 0.1 in. w.c. of downstream resistance. Note that this requirement is at maximum rated CFM. Most VAV boxes are selected at some rating less than maximum rated CFM, about 80% of maximum rating to reduce noise generation and to allow some margin for error. At 80% CFM, with the rated static pressure the box would be approximately 80% open, introducing a resistance into the system to throttle the flow. If the upstream static pressure was reduced to 0.32 in. w.c. (64% of required minimum), the pressure-independent VAV box would sense the reduced flow and would open further to maintain the flow the thermostat is calling for. As the load decreased further to 50%, the box would begin to close off but if, at the same time, if the duct static pressure is lowered, the box could be fully open at 50% load, with less than 0.1 in. w.c. static pressure at the inlet.

The solution is to abandon the duct static pressure control of supply fans and to replace it with direct feedback from the VAV box damper positions. With most manufacturers of DDC VAV controllers, one of the items of information available is the box damper actuator position. With DDC systems, it is relatively easy to collect the box damper positions and select the box with the damper that is open the most. The setpoint for the variable-speed drive becomes VAV box damper position, i.e., 95% of damper opening. It is important in developing the control loop to feed the box position signal into an averaging function to slow down the control action. Averaging the readings over five minutes should be sufficient, in most cases.

Figure 4. *The Pressure Required of the Supply Fan is the Pressure Required to Push Design CFM Through any Box Requiring Design CFM.*



	At Mfg Maximum	At Actual Maximum
VAV Box	0.257"	0.155"
RH Coil	0.304"	0.184"
Duct	0.054"	0.036"
Diffuser	0.07"	0.07"
Total	0.685	0.445

Size 8 Box(with Attenuator)
Mfg Max = 900 CFM
Actual Max = 700 CFM

Using a direct-acting control loop with the box calling for the most cooling at 95% open, if the load increases in that zone, the box will open more. As the box opens beyond 95%, the supply fan VFD will be commanded to speed up until the box is back to 95%. As the load is reduced or boxes are closed off at the end of occupancy, the fan slows down until at least one box is 95% open.

Potential savings are on the order of 20% to 40% of the fan energy if the VAV boxes toward the end of the duct are the ones that are usually calling for the most cooling. In the case of the loop duct, as the sun travels around the building the load will follow and the boxes closer to the fan will require the most pressure, reducing the overall fan energy even more. In addition to energy saving, the radiated and distributed noise generated from VAV boxes when the volume damper modulates toward the closed position, will be reduced through most of the operating range.

Chiller Start/Stop

Starting and stopping the chiller has usually been a time and temperature based control with discrete component control systems. By analyzing the system we can find an alternate strategy that is even more efficient and will prevent the chiller and its associated equipment from operating on mild days. When is the chiller needed? When a zone(s) runs out of cooling capacity. When is a zone out of cooling capacity? When the zone damper is fully open. But, a zone damper can be open for two reasons, the supply duct pressure can be too low, or the air temperature in the supply duct is not cold enough. Since our system is already using zone damper position to control the supply fan capacity, then we know that if a damper is fully open, and the variable speed drive controlling the supply fan is at, or near, full speed, then the temperature of the air is too warm for cooling and the chiller needs to be turned on. This gives the information necessary to provide an optimum start time for the chiller only when there is need for chilled water.

This strategy trades fan energy for chiller energy by not starting the chiller and associated pumps at 55°F outside air temperature, by keeping the chiller off until the outside air temperature is near 60°F, and should save 500 to 1,000 hours per year of chiller operation over a chiller that is started only on the basis of outside air temperature and time-of-day.

Secondary Chilled-Water Pump Control

Using a primary/secondary chilled-water loop is another means to reduce the energy use of the building. In multiple-chiller operation, well-controlled secondary loops provide a means for staging on and off chiller capacity. Too often much of the energy savings from a secondary chilled-water pumping system is wasted by either using a bypass valve to control differential pressure or by using differential control of the variable-speed drive but selecting the wrong location for the control sensor. The differential pressure control sensors are often found a few feet away from the secondary chilled-water pumps set at 30 psid. The more efficient installations put the differential pressure sensor between the supply and return, toward the end of the chilled-water loop set at about 10 psid. Just as with the supply fan, the control setpoint is critical to the amount of saving available from a variable-flow pumping system. The setpoint is selected to ensure that there is enough differential pressure at full flow to provide design flow through the control valve, the coil, and associated piping.

An analysis of the system will show that the control valve is very similar to the VAV box and is selected based on pressure drop at full flow. Two-way chilled-water valves are typically selected with 1 to 2 psi drop at full flow, though occasionally they are selected at up to 5 psi. The chilled-water coils are also selected with approximately 5 psi drop at full flow. While it is readily apparent that the coil pressure drop varies with the square of the flow, the control valve also has the same characteristic when it is open fully.

To take full advantage of the turndown capability of a variable flow secondary pumping system the control system must try to maintain at least one chilled-water control valve 95% open. The control logic is the same as for the supply fan static pressure control, using the information available in the direct digital control system to select the valve that is open the most and to reduce the pressure until that valve is 95% open. Because the supply air temperature loop is not controlling flow like the pressure-independent VAV box, the time lag for control action may have to be increased to approximately 10 minutes to allow time for the supply air temperature loop to detect the change in flow through the coil. It should be noted that this control sequence will only work if the chilled-water control valve is controlled by a supply air sensor. The time lag in room sensors controlling chilled-water valves is too long be responsive to this type of control system.

This control method will solve some of the problems of pressure distribution and reduce the need for balancing in variable flow water systems (Avery, 1992).

Other Strategies

With global control of the zones, other strategies can be employed to reduce energy use. Zone temperature set-points can be shifted up on warm days and down on cold days. At the end of the day, set-points can be shifted up to reduce chiller and/or fan energy. This brings up another strategy, avoid using proportional-integral control loops for zones.

Using a proportional-only loop will allow the zone temperature to droop when the load is light and rise when the load is heavy. Since much of the load variation on the exterior zones is weather related, the zone temperature will automatically shift toward the outside air temperature lowering the losses, or gains through the outside surfaces.

If the telephone interface is used for after-hour occupancy control, occupants can use the system to have control over the zone set points, but the range of setpoint adjustment can be limited to reduce the impact on feedback systems.

OTHER ISSUES

Further work is needed to implement technologies like ''fuzzy logic'' to the control systems that can make economic decisions between various control strategies. For instance, is it more efficient to start the chillers or to use fan energy for cooling. There are times when resetting chilled-water temperatures up may be more energy efficient than the increased pumping energy to handle the higher flows.

CONCLUSION

The installation of direct digital control to all controlled components in HVAC systems provides the opportunity to achieve a higher degree of energy efficiency than with discrete control components. The resulting integration of subsystems reduces operating times and increases efficiency of operating equipment. For the highest degree of energy efficiency all systems must be integrated and operate in concert, the zones being the most important because they are the load being served. Do not assume, however, that just because you have installed a DDC system that the control strategies that are implemented include the ones discussed in this paper. Many building owners, mechanical designers, and ESCO's think that the mere installation of a full DDC system automaticaly gives them all the energy efficiency strategies possible. This is not true and won't be true until all the participants in the project learn the capabilities of the control systems and learn to ask for all the features that they want or need. A good study of all the HVAC system components is required to be able to determine what control strategies will work and will save energy, while providing comfort.

Control and mechanical designers need to get away from ''the way we've always done it'' and truly analyze the way the systems operate and use energy. Computer based controls provide the means to bring about true energy efficient control of HVAC. However careful programming and complete commissioning are still necessary for an energy efficient and comfortable air conditioned building. One building, being used in an ASHRAE research project, saved over 40% of the energy they were using for HVAC just by tuning a ''state of the art'' system. This points up the need for understanding the systems that controls are being applied to and not taking for granted that once a system is installed that it will operate as planned.

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