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Optimization of HVAC Control System Strategy Using Two-Objective Genetic Algorithm

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Intelligent building technology for building operation, called the optimization process, is developed and validated in this paper. The optimization process using a multi-objective genetic algorithm will permit the optimal operation of the building's mechanical systems when installed in parallel with a building's central control system. Using this proposed optimization process, the supervisory control strategy setpoints, such as supply air temperature, supply duct static pressure, chilled water supply temperature, minimum outdoor ventilation, reheat (or zone supply air temperature), and zone air temperatures are optimized with respect to energy use and thermal comfort. HVAC system steady-state models developed and validated against the monitored data of the existing VAV system are used for energy use and thermal comfort calculations. The proposed optimization process is validated on an existing VAV system for two summer months.Many control strategies applied in a multi-zone HVAC system are also tested and evaluated for one summer day.

INTRODUCTION

The operation of heating, ventilating, and air-conditioning (HVAC) systems is a critical activity in terms of optimizing a building's energy consumption (and, hence, reducing costs), ensuring the comfort of occupants, and preserving air quality. The operation of these systems in most of the 430,000 commercial and institutional buildings in Canada is suboptimal, resulting in energy losses of 15% to 30%. One example of the deficient operation of HVAC systems is the simultaneous use of air-conditioning and heating systems. The performance of HVAC systems can be improved through better supervisory control strategy. Setpoints for HVAC systems can be adjusted by the supervisor to maximize overall operating efficiency.

Most existing HVAC system processes are optimized at the local loop level; for example, in the existing HVAC system investigated in this paper, which is installed at the Montreal campus of the École de technologie supérieure (ÉTS), one of the most advanced "intelligent buildings" in North America today, each local control of an individual subsystem is individually determined, thus leading to the poor performance. The air supply temperature setpoint for this system is determined as a function of the outdoor air temperature and fan airflow rate, without taking into consideration other subsystems, such as supply duct static pressure and chilled water supply temperature setpoints. Decreasing the supply air temperature may result in a lower supply duct static pressure and fan energy. Zone reheats for the investigated HVAC system that are used in winter days–only when the zone temperatures and airflow rates fall to their minimum limits–are not optimized, given their interaction with other setpoints. Applying some reheat in the low-load zone (ventilation critical zone) could significantly reduce the system's outdoor air ventilation. The zone air temperature setpoints of the investigated HVAC system are kept constant in the comfort zone during occupied periods. However, a strategy using the optimization of the individual zone air temperature setpoints combined with other controller setpoints during occupied

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periods could reduce further system energy use. Using a two-objective optimization problem, when energy use and thermal comfort are the objective functions, could provide an opportunity to control the thermal comfort and energy use according to the day or month, thereby furthering the energy saving.

Several studies have investigated the optimum setpoints of one or several local-loop controllers. For example, Ke and Mumma (1997) investigated the interactions between the supply air temperature and the ventilation requirement using zone reheat in order to determine the optimal supply air temperature; Englander and Norford (1992) minimized the supply duct static pressure setpoint without sacrificing occupant comfort or adequate ventilation, and Braun et al. (1989) determined the chilled water supply temperature setpoint by optimizing chilled water systems.

With a view to optimizing the overall system performance, the system approach was utilized in optimal control strategies in a few studies. System-based optimal control and operation studies are examined by House et al. (1991) and House and Smith (1995). They proposed a system approach for optimizing multizone building systems, respecting energy use and without sacrificing the thermal comfort. A system approach was proposed for the online control strategy of HVAC in which multiple setpoints are optimized simultaneously in order to improve the system responses and reduce energy use (Zaheer-Uddin and Patel 1993; Zheng and Zaheer-Uddin 1996; Wang and Jin 2000). Wang and Jin (2000) presented a control strategy using a system approach based on predicting the responses of overall system environment and energy performance to changes of control settings of HVAC system. A genetic algorithm is used by the strategy to solve the optimization problem. The optimal control strategy based on steady-state models of HVAC systems has been investigated by Zheng and Zaheer-Uddin (1996). These models are interconnected to simulate the responses of the VAV system. The studies based on system approaches show that an optimal control strategy can improve the system responses and reduce energy use compared to traditional control strategies (Zaheer-Uddin and Patel 1993; MacArthur and Woessner 1993). The existing system-based optimal control studies did not address the following points: (1) the interaction between the reheat and outdoor airflow rate, (2) the interaction between the individual zone temperature setpoints with other setpoints, and (3) controlling and varying the thermal comfort during the day as a function of daily energy use by utilizing the two-objective optimization problem, which leads to further energy savings. To date, the online optimization of the overall system performance, including individual zone temperature setpoints, has not been investigated using the two-objective optimization problem.

This paper presents a system approach that takes into account wide system interaction involving reheat and zone air temperature, including supply air temperature, supply duct static pressure, chilled water supply temperature, and minimum outdoor ventilation using two-objective optimization problems. This work uses the multi-objective genetic algorithm optimization method. The genetic algorithm is applied to a wide range of scientific, engineering, and economic search problems (Goldberg 1989). The genetic algorithm potential is studied for different applications: (1) the control of air-conditioning systems (Nordvik and Renders 1991; Huang and Lam 1997) and (2) HVAC system design using the one-objective method (Wright 1996; Asiedu et al. 2000; Wright and Farmani 2001) and the two-objective method (Wright et al. 2002).

To that end, the following methodology is employed: (1) monitoring of the investigated existing HVAC system, (2) modeling and validation of HVAC components, (3) development of optimization algorithm, (4) development of proposed optimization process, and (5) application of the developed optimization process on multi-zone HVAC systems. This paper focuses on the last two points, which show the results of applying the *developed optimization process* on existing and modified HVAC systems. The first three points are briefly presented here and detailed in the references (Nassif et al. 2004a, 2004c). One paper (Nassif et al. 2004c) also touched on optimization of some controller setpoints for HVAC systems, which served only internal zones.

Supply air temperature, duct static pressure, and zone air temperatures were only optimized for one summer week. However, in this paper, we go into the application of the proposed optimization process on multi-internal and external zone HVAC systems. In this case, many controller setpoints, including outdoor airflow rate, chilled water supply temperature, and zone reheats, are further optimized for two summer months. Many control strategies applied in a multizone HVAC system are also tested and evaluated in order to show how the proposed optimization process can provide an excellent means of reducing energy use with maintaining environmental conditions in buildings.

Two different parts can be specified in this paper; in the first, the setpoints of the supervisory control strategy of an existing HVAC system serving interior zones are optimized with respect to energy use and thermal comfort. In this part, the energy use required for actual operation, calculated using the monitored and validated models, is compared to the energy use obtained through the optimization of controller setpoints; the resulting energy saving is then presented. To investigate the optimization of an HVAC control system serving multiple zones with high-load distributions between them, the setpoints of the supervisory control strategy of the HVAC system modified from the existing one is also optimized and discussed in the second part of this paper.

SYSTEM DESCRIPTION

The aim of this study is to realize an online optimization of the supervisory control strategy of multizone HVAC systems. Two HVAC systems will thus be investigated: (1) an existing HVAC system serving interior zones without zone reheats and (2) a multizone modified HVAC system with zone reheat.

Existing HVAC System

The investigated HVAC system is installed at the École de technologie supérieure (ÉTS) campus. It consists of ten air-handling units, one of which is abbreviated as AHU-6 and is investigated in this paper. It meets the load for 70 interior zones on the second floor. However, another air-handling unit, abbreviated AHU-4, which meets the load for 68 southwest perimeter zones located on the first, second, and third floors, is also studied but is not presented in this paper. Figure 1 shows the zones at the second floor served by the AHU-6 and AHU-4 units. Figure 2 illustrates the schematic of the AHU-6 HVAC system.

Modified HVAC System

The optimal control setpoints of the multi-zone HVAC system depend primarily on outdoor conditions and zone loads. However, the load distributions between zones may also be important if they change significantly over time (ASHRAE 1999).The multizone VAV system, which yields significant zone load distributions, is created from the existing AHU-6 system by combining the 69 AHU-6 individual zones into four effective zones. The remaining zone considered to be the *critical ventilation* zone remains unchanged. Table 1 shows the zone characteristics of the modified HVAC system. Each zone characteristic is derived from the original zone, i.e., the design airflow rate of the effective zone is equal to the sum of the design zone airflow rates of the combined original individual zones. Since the characteristics of the five zones are derived from the original seventy zones, the validated system components, such as the fan, the cooling coil, and the outdoor damper, are thus used for the two systems investigated in this paper (existing and modified HVAC systems). The HVAC system consists of supervisory control and local control loops. The setpoints of these control loops, including zone air temperatures, the supply air temperature, the supply duct static pressure, the zone supply air temperature or reheat required, the minimum outdoor ventilation flow rate, and the chilled water supply temperature, are optimized using the two-objective optimization problem.

Figure 1. Schematic of zones at second floor served by AHU-6 and AHU-4 units.

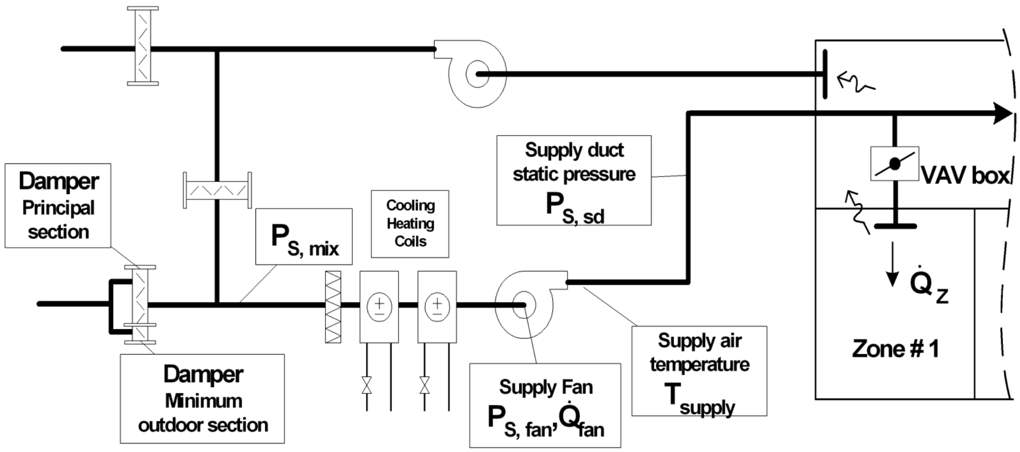

Figure 2. Schematic of AHU-6 HVAC system.

Table 1. The Zone Characteristics of a Modified HVAC System

	Z1	Z2	Z3	Z4	Z5
Design zone airflow rate*, m^3/s	0.60	7	7	7	7
Minimum zone airflow rate, m^3/s	0.18	1.8	1.8	1.8	1.8
Ventilation for zones, m^3/s	0.075	0.7	0.7	0.7	0.7
Design zone load, kW	8	88	88	88	88

* Corresponding to design duct static pressure, 250 Pa, named "maximum limit" for other duct static pressure.

SUPERVISORY CONTROL STRATEGIES AND OPTIMIZATION

HVAC systems are typically controlled using a two-level control structure. The lower-level local-loop control of single setpoints is handled using an actuator. The upper control level, the supervisory control, specifies the setpoints and the modes of operation. For the existing HVAC system at the ÉTS campus, the supervisor specifies the setpoints that are locally determined. For example, the air supply temperature setpoint for this system is determined as a function of outdoor temperature and airflow rate without taking into consideration other subsystem setpoints, such as supply duct static pressure and chilled water supply temperature setpoints. Zone reheat is used on winter days only when the winter zone temperatures and airflow rates fall to their minimum limits. The air temperature setpoint of each zone is maintained at a constant value within the comfort range during occupied periods. Although the supply duct static pressure setpoint changes gradually from zero to a maximum value (250 Pa) when the system is started up, this maximum value is always maintained at normal operation. The minimum outdoor damper position is maintained at a constant value in order to provide the required outdoor air ventilation. The chilled water supply temperature of the existing system is also constant at 7.2°C. Since these setpoints are determined at the local loop level rather than through global system optimization, the system does not perform at its full potential.

The performance of this HVAC system can be improved through the optimization of the supervisor control strategy. The *optimized supervisor* specifies the setpoints using the optimization process as shown in Figure 3, which includes (1) the VAV model, (2) the two-objective genetic algorithm optimization program, and (3) three main tools, namely, data acquisition, indoor load prediction, and selection tools. The data acquisition tool receives and processes the online measured data (i.e., unrealistic values, incomplete data, etc.). The load prediction tool predicts the sensible indoor and building latent loads for the optimization period using online measured data of the previous period. In this paper, this tool is not used for the modified HVAC system because the sensible indoor loads are merely assumed. However, for the existing HVAC system, a simple load tool is applied, assuming that the indoor sensible loads are equal to the amount of cooling that terminal boxes provide as the product of the zone airflow rate and difference in temperature between the supply and the zone air. Since a set of optimal solutions is obtained by using the two-objective optimization problem, the selection tool is used to select the appropriate solution (this tool will be discussed later).

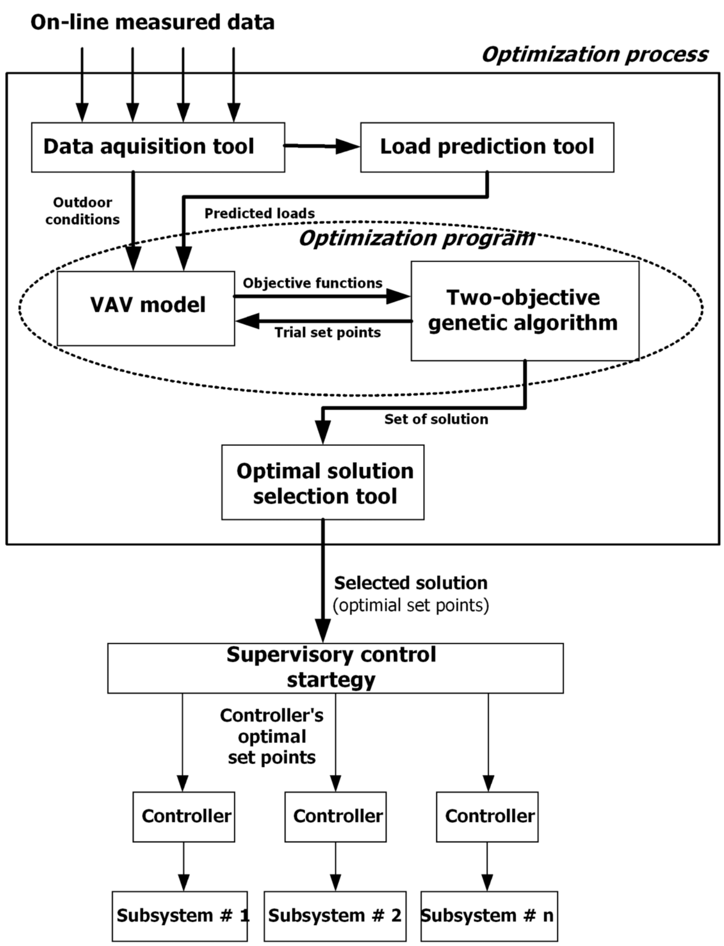

Figure 3. Optimization process of supervisory control strategy.

At each optimization period (i.e., 15 minutes), the genetic algorithm program (GAP) sends the trial controller setpoints to the VAV system model, where the energy use and thermal comfort (objective functions) are simulated and returned back to the GAP. The VAV model determines the energy use and thermal comfort resulting from the change in outdoor conditions and indoor loads (independent variables) and controller setpoints (dependent variables). For the results of this study, the computations with the optimization process are performed well within a time length of three minutes on a personal computer. This computation time allows the optimization process to be implemented online. The time could be also decreased using a more modern computer.

HVAC SYSTEM MODEL

In simulation and optimization calculations, the mathematical model of the HVAC system must include all the individual component models that influence the objective functions.

To simulate the responses of the HVAC system to the changes in outdoor conditions and indoor loads, which vary slowly compared to the optimization period (i.e., 15 minutes), the steady-state model can be used. The optimal control strategy based on steady-state models of HVAC systems has been investigated by Zheng and Zaheer-Uddin (1996). The component models are developed and validated against the recorded data of the investigated existing HVAC system (Nassif et al. 2004a). The outdoor airflow rate is determined using the damper model, and fan power and the airflow are determined using the fan model. The cooling coil demand is calculated through the detailed cooling coil model developed from the ASHRAE *HVAC 2 Toolkit* (Brandemuel et al. 1993). The chiller energy use is modeled using the performance curves of electric chillers as functions of variables, such as the cooling coil load, entering condenser water temperature, and chilled water supply temperature (NRCC 1999). The parameters of the component models are adjusted from the manufacturer's data and monitoring. To provide an adequate supply airflow rate in zones and a sufficient amount of outdoor air, the system pressure and outdoor air simulations are required, as presented in the following.

System Pressure Simulations

System pressure simulations are required to ensure that an adequate supply airflow rate is provided to every zone at the trial supply duct static pressure setpoint. It is also required to calculate the fan static pressure used by the fan model. Typically, a duct model is required to calculate the static pressure drop in each branch and a VAV box model is required to determine the VAV damper position. These models require detailed information on system air distributions and VAV box design, which in existing systems are usually not available. A simplified simulation method is used to meet the requirements described above without using intensive calculations.

The design zone airflow rates $\dot{Q}z_{max,design}$, as shown in Table 1, are the maximum airflow rate introduced into zones at design supply duct static setpoints (250 Pa) when the VAV dampers are wide open. However, these "maximum limit" airflow rates ($\dot{Q}z_{max}$) are functions of the supply duct static setpoint ($P_{S,sd}$), as presented in the following equation:

$$\dot{Q}z_{max} = \dot{Q}z_{max,design}\sqrt{\frac{P_{S,sd} - \Delta P_{duct}}{P_{S,sd,design} - \Delta P_{duct,design}}} \tag{1}$$

The term ΔP_{duct} represents the pressure drop between the static pressure sensor location and the zone VAV box inlet. Since the airflow rate velocity is not significantly changed between sensor location and the VAV box inlet, the dynamic pressure part is not included in the equation above. To ensure that every individual zone at the trial duct static pressure setpoint ($P_{S,sd}$)

receives adequate supply air, the "zone airflow rate constraint" must be respected so that the zone airflow ($\dot{Q}_z$) obtained by optimization is equal to or lower than the maximum limit of zone airflow rate calculated by Equation 1. This equation can be simplified within the normal fan operation range as follows:

$$(\dot{Q}z_{max})_{simplified} = \dot{Q}z_{max,design} \cdot \sqrt{\frac{P_{S,sd}}{P_{S,sd,design}}} \tag{2}$$

Given that $\Delta P_{duct,design} > \Delta P_{duct}$ and consequently $(\dot{Q}z_{max})_{simplified} < \dot{Q}z_{max}$, this simplification further ensures, for a given supply duct static pressure setpoint, that no zone box is starved for supply air. Therefore, the "zone airflow rate constraint" could be expressed as

$$\dot{Q}_z \le (\dot{Q}z_{max})_{simplified} \tag{3}$$

For the optimization process, the fan airflow rate and fan static pressure are required, as the inputs of the fan model, to calculate the fan power. The fan airflow rate (Q_{fan}) is determined as the sum of zone airflow rates. However, the fan static pressure ($P_{S,fan}$) can be determined using a formula represented by the operation curve expressed in terms of known design points (static pressure and airflow rate of fan) and supply duct static pressure setpoint.

$$P_{S,fan} = \left(\frac{\dot{Q}_{fan}}{\dot{Q}_{fan,design}}\right)^2 \cdot (P_{S,fan,design} - P_{S,sd}) + P_{S,sd} \tag{4}$$

Outdoor Airflow Rate

The minimum outdoor airflow rate is determined by using the corrected fraction of outdoor ventilation air in the supply system (Y), as given in ASHRAE Standard 62-1989:

$$Y = \frac{X}{1 + X - Z} \tag{5}$$

The term X is an uncorrected fraction of the outdoor ventilation air in the supply system (the ratio of the sum of the outdoor ventilation airflow rates for all zones to the fan airflow rate). The term Z is the ratio of required outdoor air to primary air in the critical zone.

VAV System Model

The component models are interconnected to form the VAV system model. The VAV model determines the energy use and thermal comfort using the trial setpoints supplied by the genetic algorithm (dependent variables) and using the outdoor condition and indoor loads (independent variables). Figure 4 shows the flow diagram for the VAV system performance calculations required by the optimization process. There are nine steps presented as blocks labeled with the names of the calculated variables.

Step 1: The zone and supply air temperature setpoints and indoor sensible load are used to calculate the zone airflow rates.

Step 2: The fan airflow rate is calculated as the sum of the zone airflow rates.

Step 3: The fan static pressure is determined by Equation 4 using as input the supply duct static pressure and fan airflow rate.

Step 4: The fan static pressure and airflow rate are used by the validated fan model in calculating the fan energy demand.

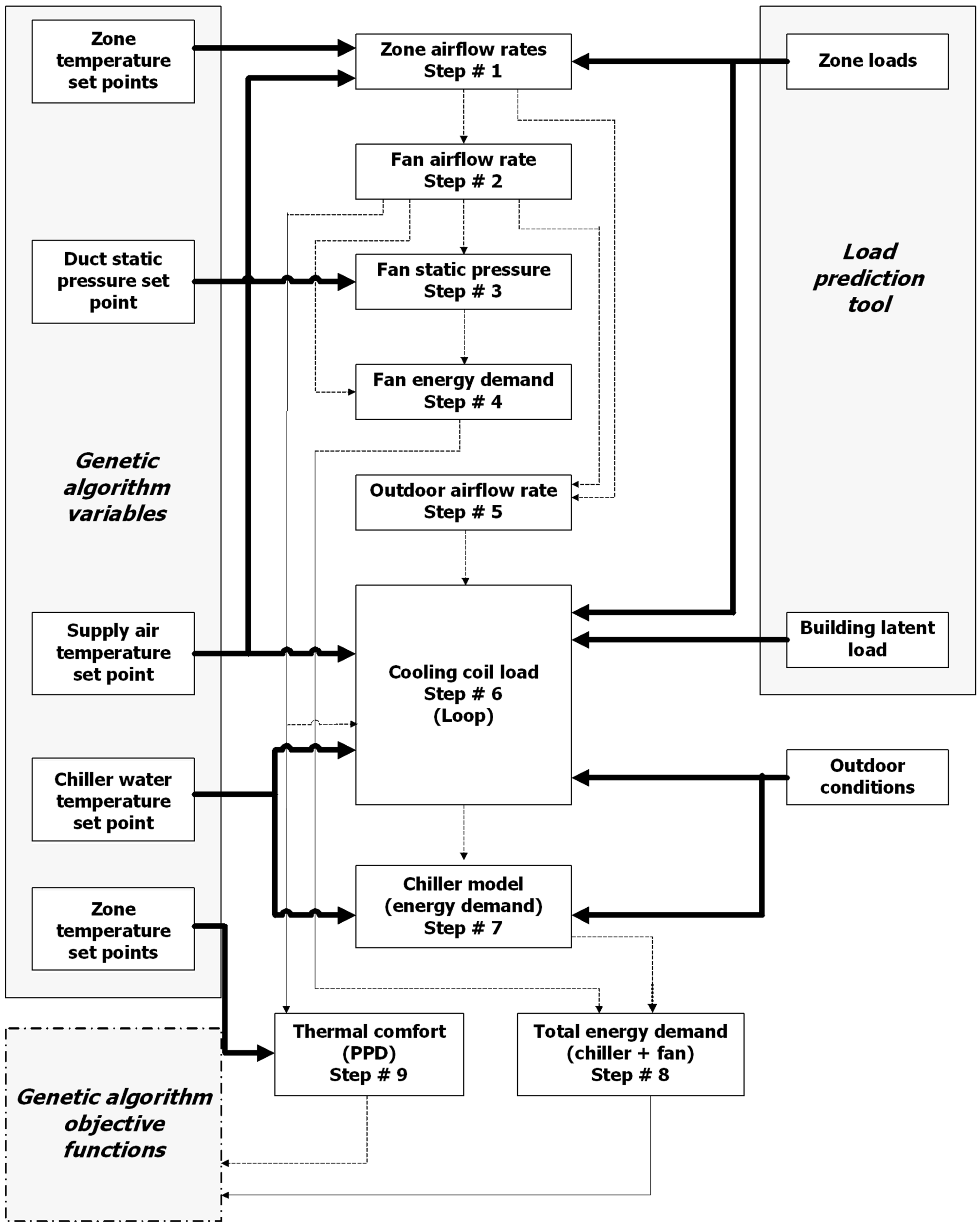

Figure 4. Flow diagram for VAV system performance calculations required by optimization process.

Step 5: The outdoor airflow fraction (Y) is determined for the modified HVAC system by Equation 5 to satisfy the zone ventilation requirement, using as input variables the zone and fan airflow rates and zone ventilation requirements. However, for the optimization of the existing system (AHU-6), the outdoor airflow fraction is assumed to be the same as the actual operation value calculated by the validated damper model using the monitoring data.

Step 6: This step represents the iteration process during which the initial value of the cooling coil leaving air humidity ratio is assumed, and the new value is calculated and reused. This iterative process continues calculating through the loop several times until the values of cooling coil leaving air humidity ratio stabilize within a specified tolerance. The following are all the required calculations in this step:

- The initial value of the cooling coil leaving air humidity ratio is assumed to be equal to the saturated humidity ratio corresponding to the cooling coil leaving air temperature (calculated from trial supply air temperature setpoint and the fan and duct heat-up).
- The return air temperature and humidity ratio are calculated using the building latent and sensible loads, fan airflow rate, and supply air temperature and humidity ratio.
- The mixing plenum air temperature and humidity ratio are determined using the energy balance and economizer logic, knowing the outdoor and fan airflow rates and outdoor and return air conditions.
- The chilled water supply temperature and supply air temperature setpoints provided by the GAP, the outdoor airflow rate, and the mixing plenum air condition (cooling coil inlet conditions) are the inputs of the inverted, detailed cooling coil model, allowing calculation of a new value of the cooling coil leaving air humidity ratio to verify the assumed value as well as the cooling coil load and water flow rate.

Step 7: The chiller power is calculated through the chiller model, using as inputs the entering condenser water temperature, chilled water supply temperature, and cooling coil load. The entering condenser water temperature is assumed to be higher than the outdoor wet-bulb temperature by 3°C.

Step 8: The total energy use is calculated as the sum of the fan and chiller energy use. For the modified HVAC system, the zone reheats are also added.

Step 9: The zone thermal comforts presented by the PPDs index are determined using Equation 7 presented below.

PROBLEM FORMULATION

The optimization seeks to determine the setpoint values of the supervisory control strategy of the HVAC system. These setpoints should be optimized for the operating consumption energy and the building thermal comfort. The optimization problem is formed through the determination of the problem variables, the objective functions, and the constraints.

Problem Variables

The following are the problem variables for existing (AHU-6) and modified HVAC systems:

- Zone temperature setpoints (5 variables for modified HVAC system and 70 variables for existing one)
- Supply duct static pressure setpoint
- Supply air temperature setpoint
- Chilled water supply temperature setpoint
- Required reheat (five variables) and minimum outdoor ventilation airflow rate (only for modified HVAC system)

Objective Functions

The setpoints of the supervisory control strategy are optimized in order to (1) reduce energy use and (2) improve thermal comfort. The energy use, including the reheat, chiller, and fan power, is calculated using the VAV system model presented above. The thermal comfort of each zone is represented as the predicted percentage of dissatisfied (PPD) and calculated using the following equation:

$$PPD = 100 - 95 \cdot EXP[-(0.03353 \cdot PMV^4 + 0.3179 \cdot PMV^2)] \tag{6}$$

The predicted mean vote (PMV) is an index devised to predict the mean response of a large group of people according to the ASHRAE thermal sensation scale (ASHRAE 1997). In a practical situation, the tabulated PMV values can be used to predict the performance of a VAV system for a combination of clothing insulation, ambient temperature, and relative velocity variables (Fanger 1970). In this paper, the zone air velocity is only assumed to be constant at less than 0.1 m/s. However, the air velocity could be varied as a function of airflow rate in the zone.

Constraints

The constraints result from restrictions on the operation of the HVAC system. They cover the lower and upper limits of variables, such as supply air temperature, zone air temperatures, etc. The constraints also cover the design capacity of components. The fan and zone airflow rates, for instance, are restricted within the maximum and minimum limits:

- The fan airflow rate must be less than the design value (23000 L/s) and higher than 40% of the design value.
- The zone airflow rates must be higher than the minimum limits given in Table 1 for the modified HVAC system and derived from the operation manual for the existing HVAC system (30% of design value).
- The zone airflow rates must be lower than the "maximum limits" corresponding to the optimal duct static pressure setpoint (with respect to Equation 3).

The PPD of each zone is limited within the [5-12%] range, and the chilled water supply temperature within the [6-10°C] range. The water flow rate calculated through the inverted cooling coil model must be equal to or lower than the design value when the valve is wide open (33 L/s).

OPTIMIZATION ALGORITHM

In this study, a genetic algorithm search method based on the mechanics of Darwin's natural selection theory was developed to solve the optimization problem. Since energy use and thermal comfort are the objective functions, a multi-objective genetic algorithm must be investigated. The two-objective genetic algorithm optimization method investigated here is an elitist, non-dominated sorting genetic algorithm (NSGA-II) developed by Deb (2001). This algorithm uses the elite-preserving operator, which favors the elites of a population by giving them an opportunity to be directly carried over to the next generation. After two offspring are created using the crossover and mutation operators, they are compared with both of their parents to select the two best solutions among the four parent-offspring solutions.

The flow chart of the NSGA-II program is shown in Figure 5. It starts with a random initial generation. First, the parents and offspring are combined, and, secondly, the problem variables (controller setpoints) are encoded into real numbers and concatenated to form a string that represents an individual in the population. Using the VAV model, the energy use and thermal comfort (two objective functions) are determined. As a result of the constraint functions, a penalty must

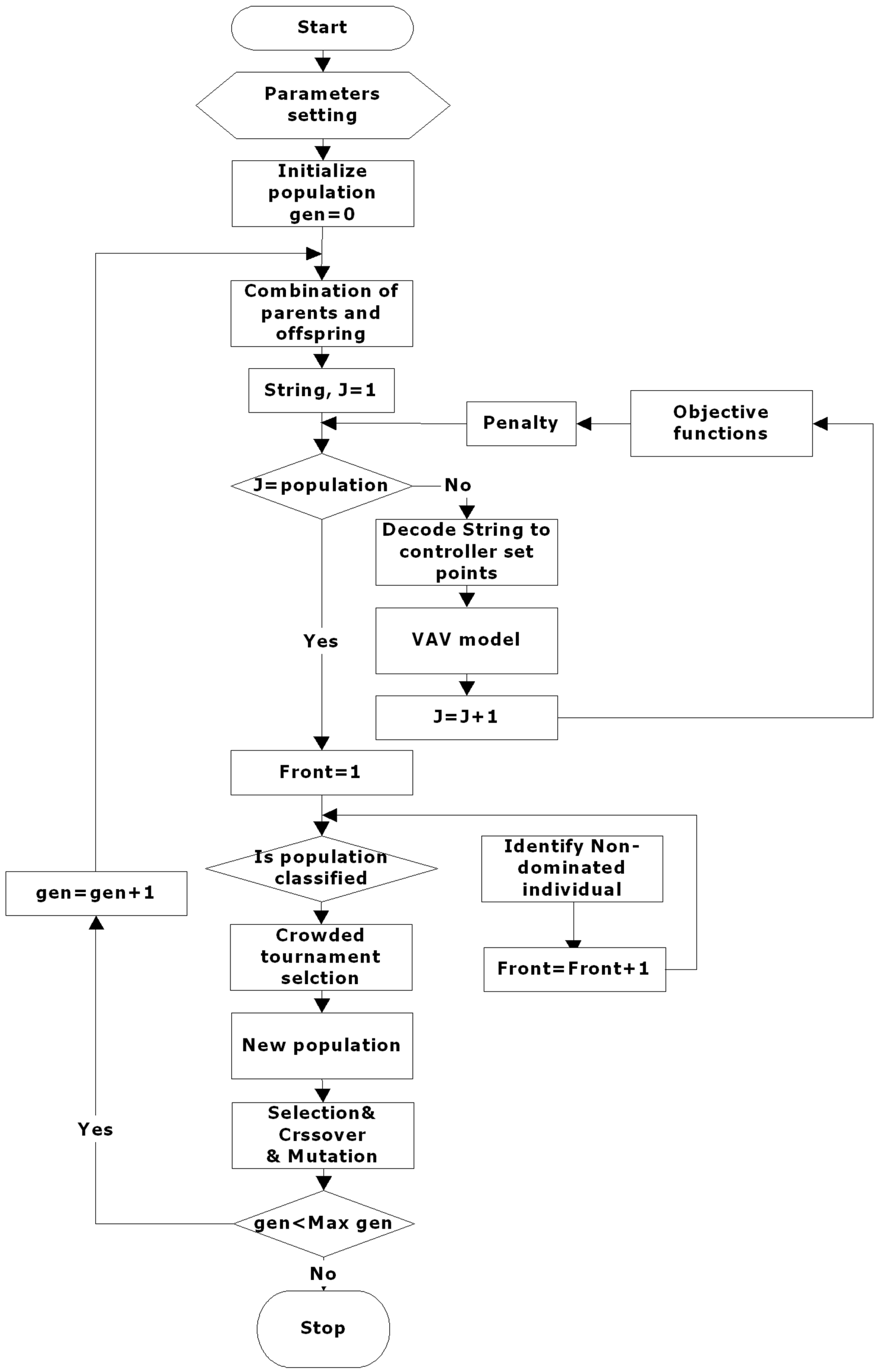

Figure 5. Flow chart of NSGA-II program.

be imposed on the objective functions. The constraint violation is calculated using the penalty function approach. The penalty parameters are set at 100 and 5 for the energy use and thermal comfort objectives, respectively. When the objective functions of all strings in a generation are calculated, the solutions are classified into various non-dominated fronts. The crowded tournament selection operator (Deb 2001) is used to compare two solutions and returns the winner of the tournament according to two attributes: (1) a non-dominated front in the population and (2) a local crowding distance. The first condition makes sure that the chosen solution lies on a better non-dominated front, and the second condition ensures a better spread among the solutions. The simulated binary crossover (SBX) is used here to create two offspring from two-parent solutions. The random simplest mutation operator is applied to randomly create a solution from the entire search space.

The control parameters of NSGA-II must be adjusted to give the best performance. A number of genetic algorithm methods with adjustments made to their control parameters were investigated for solving different mathematical problems. The results showed that the real coded NSGA-II with its setting parameter described next performed better than the others with respect to two performance metrics: (1) metrics evaluating the closeness to the Pareto-optimal solutions-*generation distance* method (Veldhuizen 1999) and (2) metrics evaluating diversity among non-dominated solutions-*spread* metric (Deb 2001). The results are presented in Nassif (2002). The parameters are: crossover probability $p_c = 0.9$ with distribution index $\eta_c = 4$, mutation probability $p_m = 0.04$ and population size $p_z = 100$. The control parameters are also tested and evaluated for the HVAC system problem (Nassif et al. 2004 b). It found that the NSGA-II with those control parameters produces better convergence and distribution of optimal solutions located along the Pareto optimal solutions. The 500 generations, when the program stops searching, are quite enough to find the true optimal solutions.

OPTIMIZATION OF EXISTING HVAC SYSTEM

The proposed optimization process presented in Figure 3 is applied on the existing AHU-6 system. Evaluations are done for July and August 2002. The data acquisition tool provides the averaged monitored data for these periods. The energy use of the optimized control strategy of the existing HVAC system (optimal energy use) was calculated using the VAV system performance calculations illustrated in Figure 4. The independent input variables, such as the zone loads, the building latent load, and outdoor conditions, are derived directly or calculated from monitoring data. Indoor sensible loads are determined by assuming that they are exactly the same as the product of monitored zone airflow rate and the difference in the monitored temperature between the supply and air zone. The building latent load is calculated using the product of the fan airflow and the difference in supply and return humidity ratios. Both humidity ratios are determined through the monitored air temperature and relative humidity. The fan airflow rate is not measured but rather is calculated by the fan model using the monitored system static pressures and fan speed.

The energy use of the actual control (actual energy use) is also simulated using the VAV model presented in Figure 4. In this case, the actual setpoints are used. It should be noted that when the airflow rates introduced in each zone are calculated in step 2 of the VAV system model section, the resulting zone airflow rates are the same as the monitored values. The outdoor damper is used to calculate the outdoor airflow rate instead of step 5 of VAV system model section.

The two-objective genetic algorithm optimization program was run for 500 generations with the parameters indicated above. The bonds on setpoints are specified from the actual operation system. The bonds for the supply air temperature, zone temperature, supply duct static pressure, and chilled water supply temperature are 13-23°C, 20-25°C, 250-150 Pa, and 6-11.5°C, respectively. The supply air temperature setpoint may become higher than the upper bound and supply duct static pressure may become less than the lower bound during the generations while the constraints are respected.

Figure 6 shows the feasible solutions obtained after 500 generations at 5:00 p.m. for July 25. The PPD presented here is the mean value of the PPD determined for each of 70 zones. An increase in thermal comfort (decrease in PPD) requires an increase in energy demand. To compare the optimal energy use with the actual one, only one solution, with the same PPD value as the actual one, is selected (the bold point in Figure 6). Figure 7 shows the actual and optimal energy demand for one week in July. The energy savings by optimization is about 19.5% for this week and about 16% for July and August. Figures 8 and 9 show the optimal supply air and chilled water temperatures and duct static pressures for July 25 to 31. The actual supply air temperature, chilled water temperature, and duct static pressure for this week of July are 14°C, 7.2°C, and 250 Pa, respectively.

Figures 8 and 9 show only the results relating to the AHU-6 system. Concerning the zone variables obtained by the monitoring and by the optimization, the following could be noted:

- The optimal zone airflow rates are limited by the zone airflow rate constraint described above (30% of design value). However, the actual zone airflow rate, shown by the monitoring, could be less than this limited value.
- Most optimal zone temperatures are situated between 23°C and 24.5°C and the optimal zone PPDs are limited within the 5-12 range.

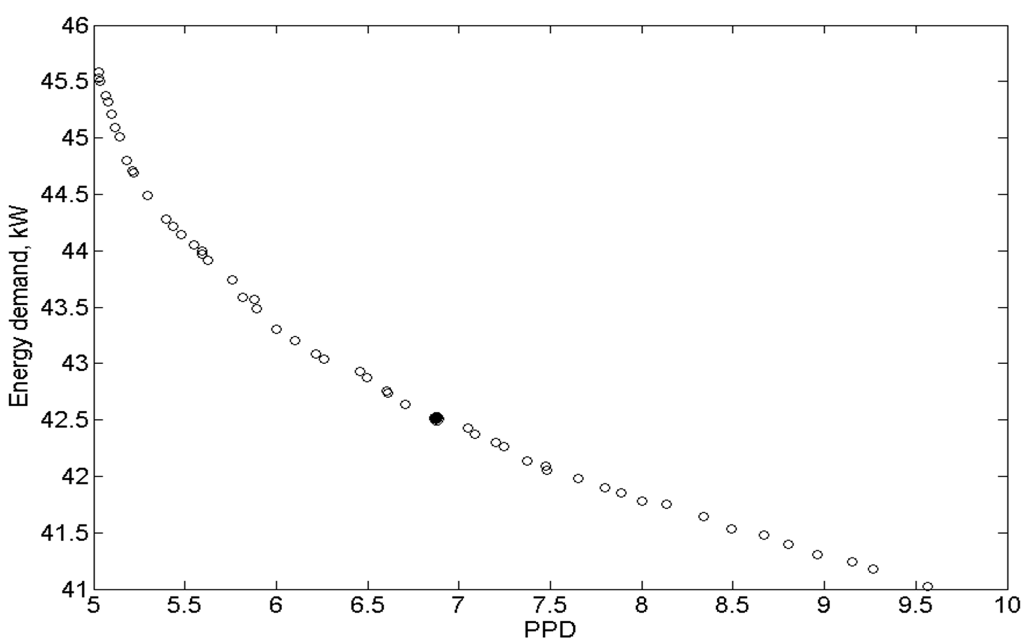

Figure 6. Feasible solutions obtained after 500 generations at 5:00 p.m. for July 25, 2002.

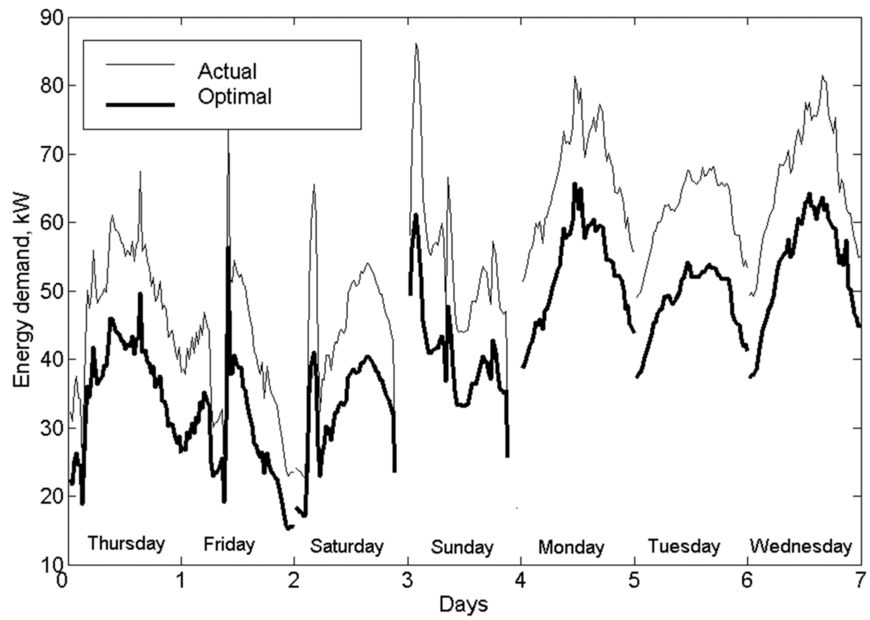

Figure 7. Actual and optimal energy demand for July 25 to 31, 2002.

- The actual zone temperatures are within 19°C and 25°C and most zone PPDs are within the 5-15 range. Some zone PPD values are as high as 30.

These results indicate that the optimization of the control strategy with required constraints could improve the operating performance of the existing HVAC system.

OPTIMIZATION OF MODIFIED HVAC SYSTEM

Definition of Investigated Control Strategies

To evaluate the performance of the optimal control strategy of a typical HVAC system using the two-objective genetic algorithm, three different control strategies (A, B, and C) applied to the modified HVAC system are tested:

- *Strategy A*: the setpoints of zone and supply air temperatures, minimum outdoor airflow rate, chilled water supply temperature, and supply duct static pressure are optimized using zone reheats.
- *Strategy B*: all controller setpoints described above are optimized without reheat.
- *Strategy C*: it is the same as that applied to the existing HVAC system at the ÉTS, such that all controller setpoints, except the supply air temperature, are constant. The chilled water supply temperature and supply duct static pressure are 7°C and 250 Pa, respectively. The supply air temperature setpoint changes linearly within the 13°C to 18°C range when the outdoor temperature varies within the –20°C to 20°C range. The supply air temperature calculated above is corrected by adding a value that varies linearly from –2°C to +2°C, corresponding, respectively, to the variation of the fan airflow rate ratio from 50% to 90%. The supply air temperature setpoint is always limited within 13°C, 18°C.

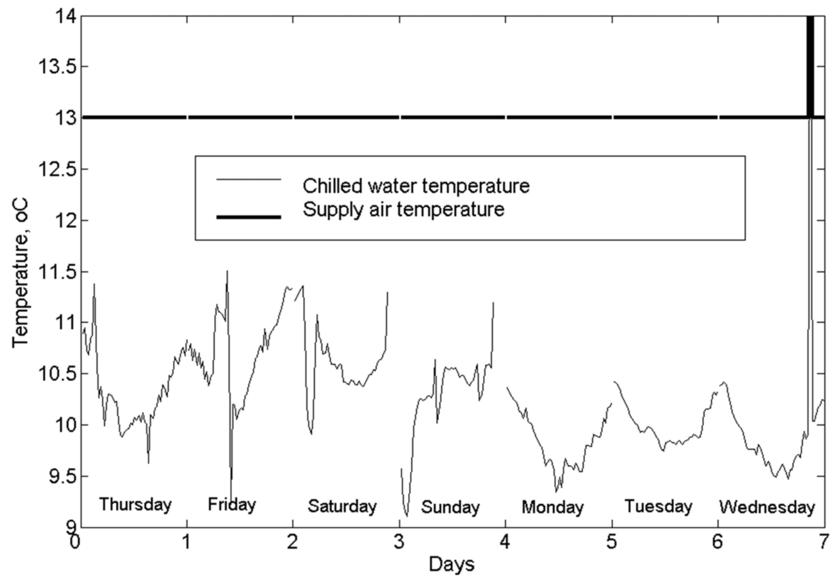

Figure 8. Optimal air supply and chilled water temperature setpoints for July 25 to 31, 2002.

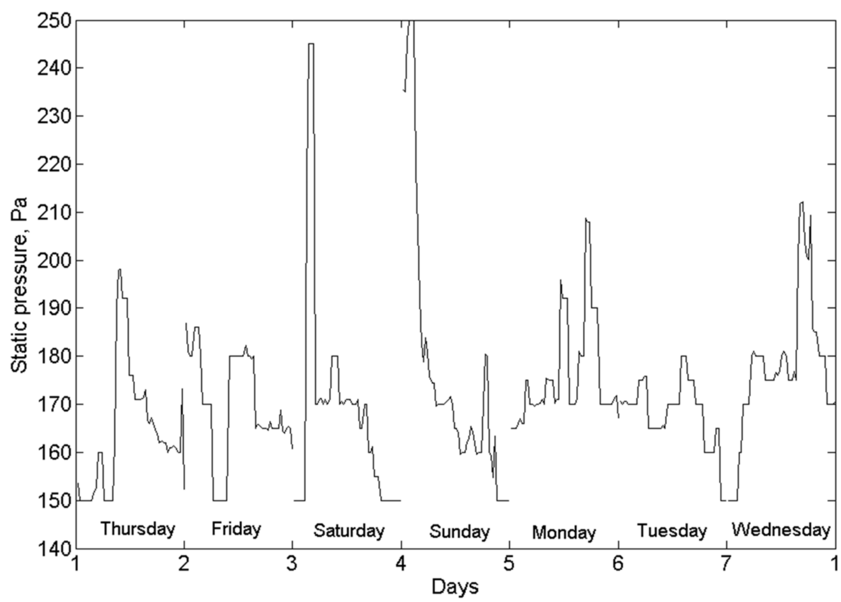

Figure 9. Optimal duct static pressure setpoint for July 25 to 31, 2002.

The zone reheat of strategy C is used differently than in strategy *A*. The zone reheat of Strategy C is used at a low load when both the zone airflow rate and zone air temperature fall to their minimum limits (i.e., 30% of design zone airflow for zone Z1 and 20°C, respectively). However, the zone reheat of strategy A is optimized, allowing the reheat to be possibly turned on in order to reduce energy use. Since the zone air temperature setpoints in strategy C are constant at 23.7°C, as in the existing ones, the optimization problem is then solved using the one-objective optimization problem, respecting only energy use and defining the thermal comfort criterion as the constraint.

Results of Investigated Control Strategies

The optimization of controller setpoints is done for one summer day under indoor sensible loads and outdoor conditions, as presented in Figure 10 and Figure 11, respectively. The outdoor relative humidity varies from 60% to 70%. The two-objective genetic algorithm optimization program was run for 500 generations at each 30 minutes, using the genetic algorithm control parameters described above. To compare the strategies using strategies A and B with those using strategy C, only one solution among strategies A and B, with a building thermal comfort PPD similar to that of strategy C, was selected.

The results of the control strategies tested are illustrated in Figures 12-16. Figure 12 shows the optimal energy demands simulated for three control strategies (A, B, and C) so that all strategies have the same building PPD. Strategy A, in which the zone reheats are optimized, needs the least energy as compared to the two other strategies. The energy saving obtained by strategy A compared to strategy C is about 12%. Since the load acting on the zone (Z1) is relatively low, this zone is considered to be the critical ventilation zone. Figure 13 shows the ratio of the optimal to design zone airflow rate for the critical ventilation zone (Z1). The 30% value indicates that the airflow rate of zone Z1 falls to its minimum. Figure 14 shows the required reheat in the critical ventilation zone (Z1). After 6:00 p.m., in both strategies B and C, the airflow rate supplied to zone Z1 is decreased to its minimum limit, whereas in strategy A it stays above its minimum limit due to the reheat applied in this zone in order to reduce the outdoor airflow rate. The optimal solution obtained indicates that no reheats are required in other zones. Figure 15 shows the optimal supply air temperature setpoint. The optimal supply duct static pressure setpoints of strategies A and B obtained for this profile zone loads are 150 Pa. The optimal chilled water supply temperature setpoints are shown in Figure 16.

DISCUSSION OF RESULTS

Reheat Vs. Minimum Outdoor Airflow Rate

To optimize the minimum outdoor airflow rate setpoint, the reheat applied in the zones must be considered. This concept was investigated by Mumma and Bolin (1994) using the corrected fraction of outdoor ventilation air in the supply system (Y), as presented in Equation 5. The reheat applied to the critical zone could reduce the fraction of outdoor ventilation air in the supply zone (Z) and, consequently, reduce the fraction of outdoor ventilation air in the supply system (Y). Depending on the outdoor conditions, this could represent a very significant reduction in the chiller load. At 6:30 p.m., since the load in zone (Z1) is relatively small, the airflow rate introduced into this zone must be decreased to its minimum limit (180 L/s) when no reheat is applied (strategy B); in this case, the corrected outdoor ventilation fraction in the supply system (Y) is 27.454%, and the outdoor airflow rate is then 3521.6 L/s. Applying a 0.7 kW reheat in this zone (strategy A) brings this fraction (Y) to 24.8% and the outdoor airflow to 3165.1 L/s due to the increasing critical zone airflow rate. Figure 17 shows the outdoor airflow rate for three strategies. Since the reheat is applied in the critical zone (Z1) for strategy A after 6:00 p.m., the outdoor airflow rate is significantly decreased.

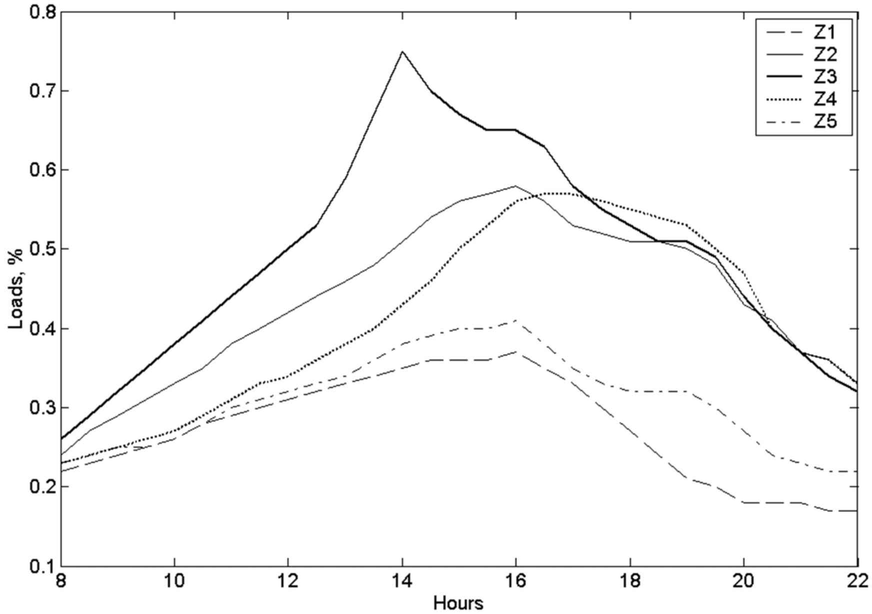

Figure 10. Ratio of actual to design sensible loads for zones.

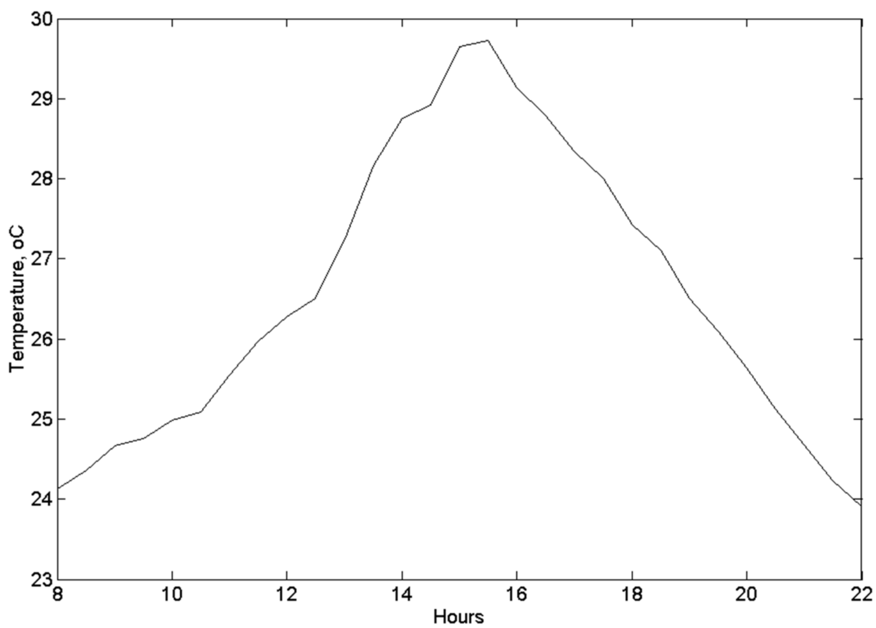

Figure 11. Outdoor air temperature profile.

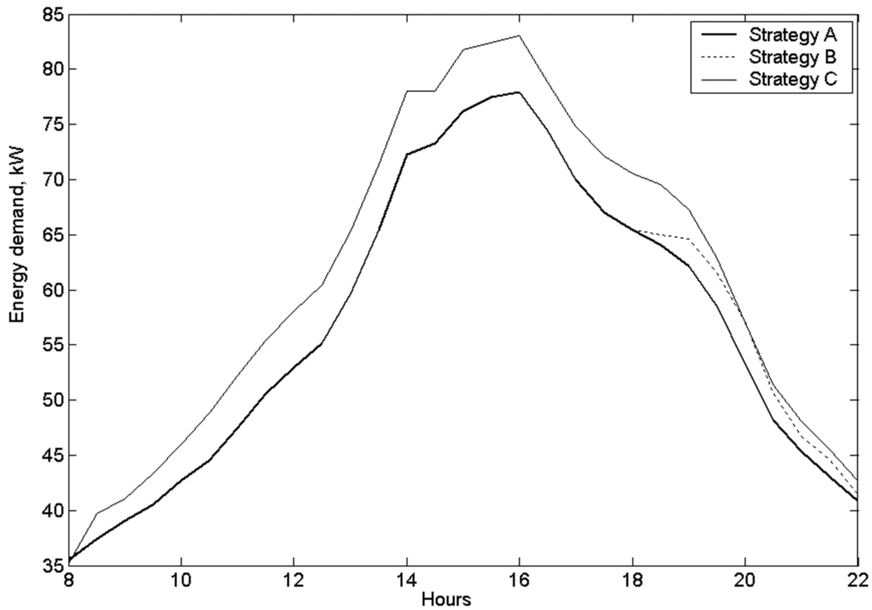

Figure 12. Optimal energy demand for control strategies A, B, and C.

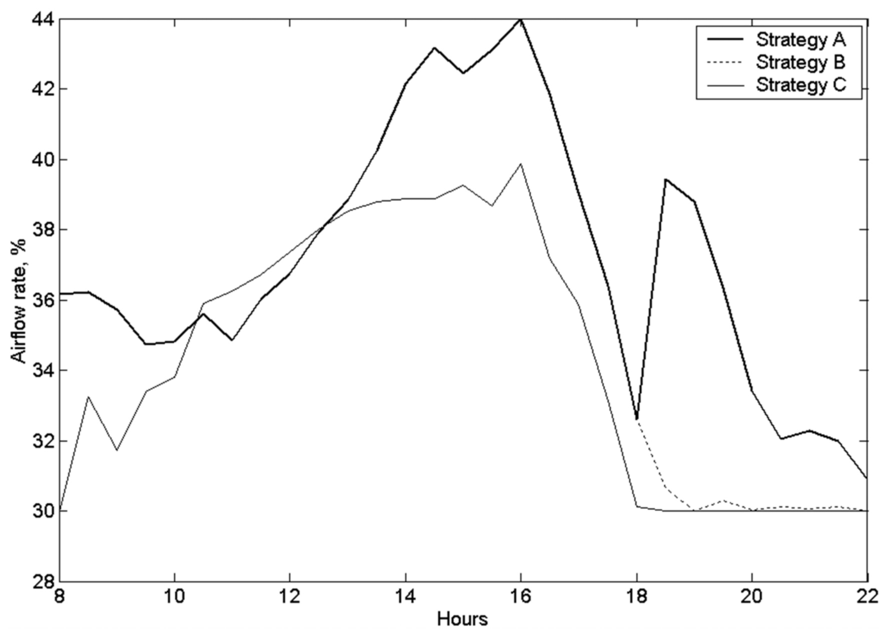

Figure 13. Ratio of optimal to design airflow rate of critical ventilation zone (Z1) for control strategies A, B, and C.

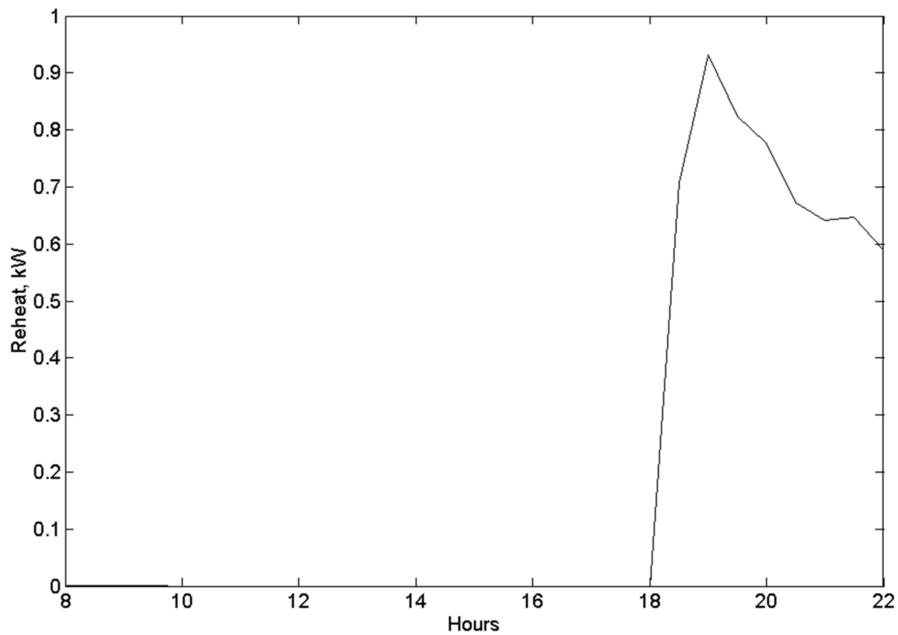

Figure 14. Reheat applied to critical zone (Z1) for control strategy A.

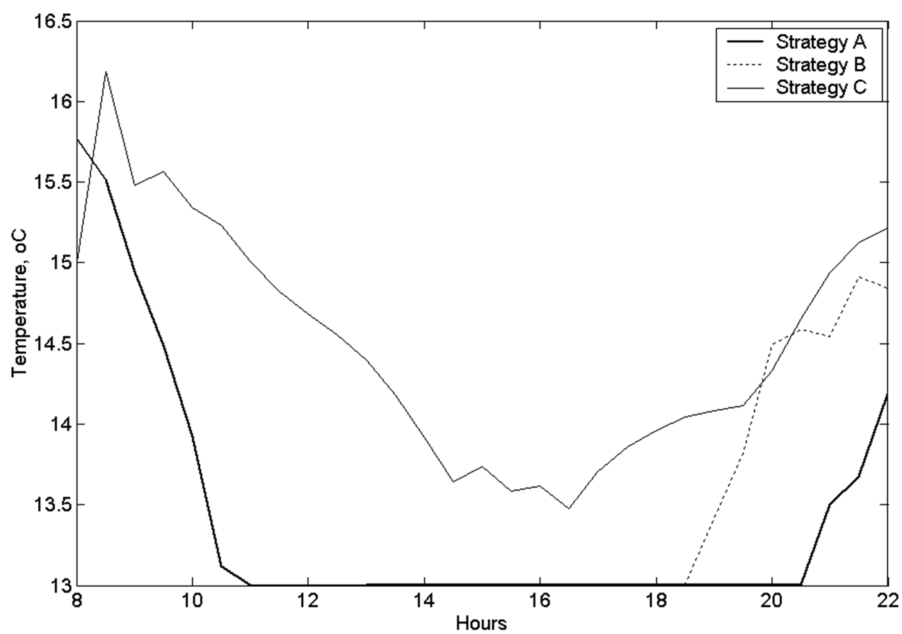

Figure 15. Optimal supply air temperature setpoint for control strategies A, B, and C.

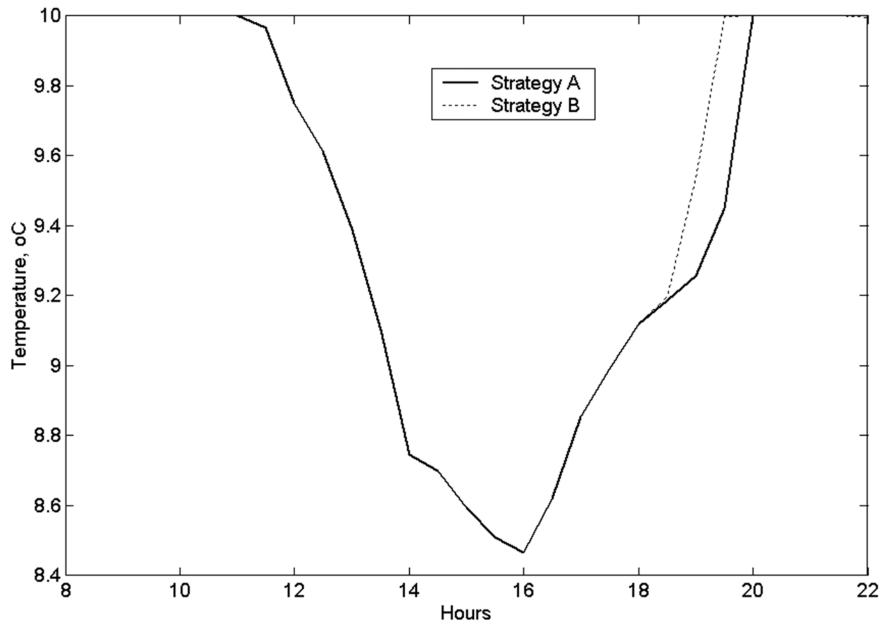

Figure 16. Optimal chilled water supply temperature setpoint for control strategies A and B.

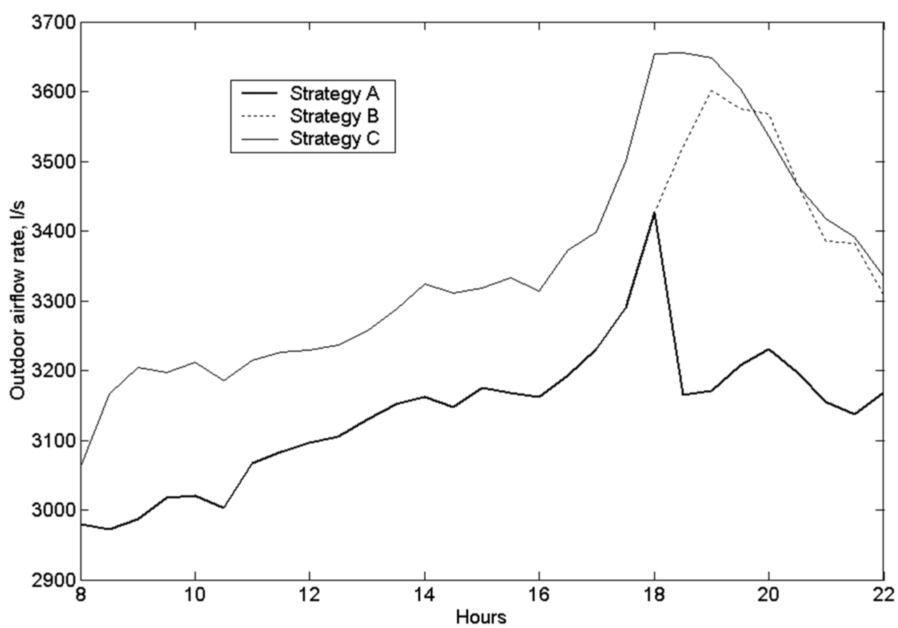

Figure 17. Outdoor airflow rate for control strategies A, B, and C.

System Supply Air Temperature

The optimal supply air temperature setpoint is determined to give a minimum sum of reheat, fan, and chiller powers while respecting thermal comfort. Before 6:00 p.m., since there are no zone reheats, the optimal supply air temperature is determined taking into account the fan and chiller power. As the fan power saving is greater than the chiller power penalty due to decreasing chilled water supply temperature (from 10°C to about 8°C) as shown in Figure 16, the optimal supply air temperature is at its minimum level (13°C) while the fan and zone airflow rates within their respective limits can meet zone loads. After 6:00 p.m., since the load acting on zone Z1 is relatively low, the zone airflow rate at its minimum limit is high enough to satisfy this low load with a relatively low air supply temperature setpoint (13°C), thus leading to the zone temperature falling below the minimum limit (i.e., 20°C). To increase this zone temperature, there are two possibilities: (1) increasing the supply air temperature, leading to a higher fan power and higher chiller coefficient of performance (COP) and (2) using some reheat in this zone. If the fan energy saving is greater than the reheat penalty, the second method will be applied, as in our case after 6:30 p.m. (Strategy A). Figure 18 shows the reheat, fan, and chiller energy demand for a period after 6:00 p.m. (when the reheat is used). The chiller energy demand for strategy A (using reheat) is smaller than that for strategy B due to the decreasing outdoor airflow rate, although the COP is greater for strategy B. The fan energy demand for strategy B is greater than that for strategy A after 6:30 p.m. due to the increasing supply air temperature setpoint for strategy B–from 13°C to 14.5°C.

It was concluded that, as in our case, using reheats in relatively low zone loads could save fan energy use as a result of working at low supply temperature setpoints; as well, there could be savings in chiller energy use due to a decrease in the outdoor airflow rate. In both cases, the savings must be lower than the energy penalty resulting from reheat and the decrease in COP at low air supply temperatures.

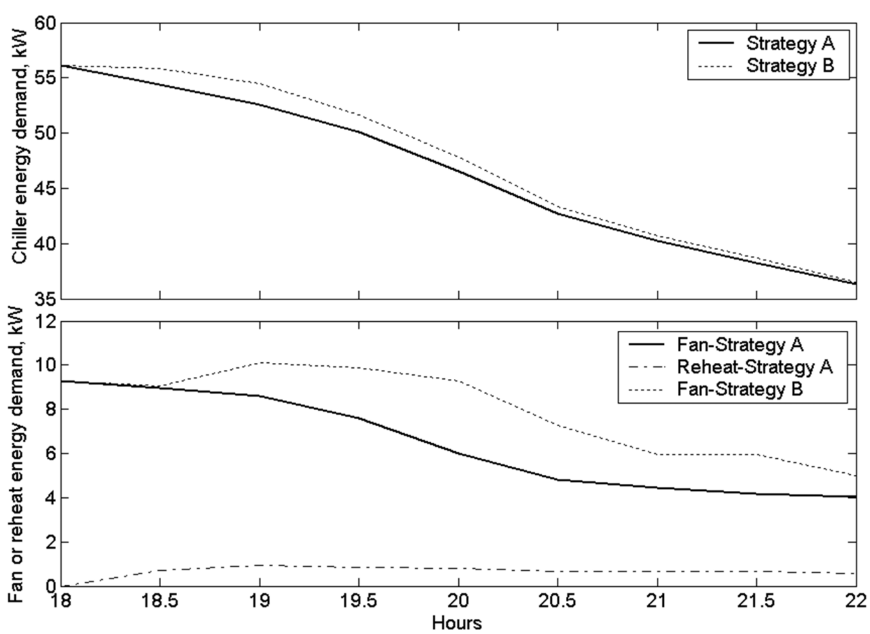

Figure 18. Reheat, fan, and chiller energy use for a period after 6:00 p.m. when the reheat is used.

Optimization of Zone Temperature Setpoints

Typically, zone air temperatures are maintained at constant setpoints in the comfort zone during occupied periods. However, during unoccupied times, the setpoints are set up for cooling and set back for heating in order to reduce energy use. A strategy using the optimization of the individual zone temperature setpoints, combined with other controller setpoints during occupied periods, could further reduce system energy use. The variation of individual zone air temperature setpoints has an effect on the system controller setpoints, such as outdoor airflow rate, supply air temperature, and supply duct static pressure. To evaluate the effect of individual zone air temperature on supply duct static pressure, an individual zone with a relative high load must be examined because this pressure is determined according to this zone's airflow rate. At 2:00 p.m., since the load acting on Z3 (see Figure 10) is relatively high with the resulting high airflow supplied to this zone, the duct static pressure setpoint that must respect the "zone airflow rate constraint" represented in Equation 3 is determined accordingly. When the temperature setpoint of this zone is set to 22.8°C, the duct static pressure setpoint is 170 Pa. If this individual zone temperature setpoint is increased from 22.8°C to 23.5°C, the airflow rate of zone Z3 to satisfy the load is decreased and, consequently, the duct static pressure setpoint could be 150 Pa. This results in an amount of fan energy saving that depends also on the required fan airflow rate. In order to evaluate the effect of individual zone temperature on the outdoor airflow rate and the air supply temperature setpoints, an individual zone with a relatively low load (Z1) must be examined. The temperature setpoint of the zone (Z1) could be decreased in order to increase the zone airflow required to satisfy this zone load and, consequently, to decrease the outdoor airflow rate (Y from Equation 5).

Another advantage of optimum zone temperature setpoints is their reduction of energy use while maintaining building thermal comfort. When the load distributions between zones change significantly, the individual determination of optimal zone temperature setpoints according to proper loads could decrease energy use significantly. If a certain average building thermal comfort is required, two methods could be applied: (1) the setpoints of zone air temperatures are optimized to give the required building thermal comfort (building PPD), (2) the setpoints of zone air temperatures are set to the same value to give the required building PPD. It should be noted that the building PPD is equal to the mean zone PPD respecting the zone PPD constraint. As discussed above, the variation of individual zone temperature setpoints has an effect on system controller setpoints, such as outdoor airflow rate, supply air temperature, and supply duct static pressure, and, consequently, an effect on energy use. The following two strategies have been tested and evaluated:

- *Strategy A*: same as the strategy described above so that the zone air temperature setpoints are optimized.
- *Strategy D*: same as strategy A, except that the zone air temperature setpoints are kept constant at 23.7°C.

Figure 19 shows the energy demand for strategies A and D. It is clear that, for the required building thermal comfort, the energy demand using optimal zone temperature setpoints (strategy A) is less than for strategy D using identical setpoints for all zones without considering load distributions between zones. Optimal solutions for strategy D are obtained by running the one-objective optimization program with constant zone temperature setpoints equal to 23.7°C. Optimal solutions for strategy A are obtained by running the two-objective optimization program with the optimization of individual zone temperature setpoints. To compare strategy A with strategy D, only one solution having the same PPD as strategy D is selected at each period.

Finally, strategy A, using the optimization of zone temperature setpoints during occupied periods, allows controlling the daily building thermal comfort by using two-objective optimiza-

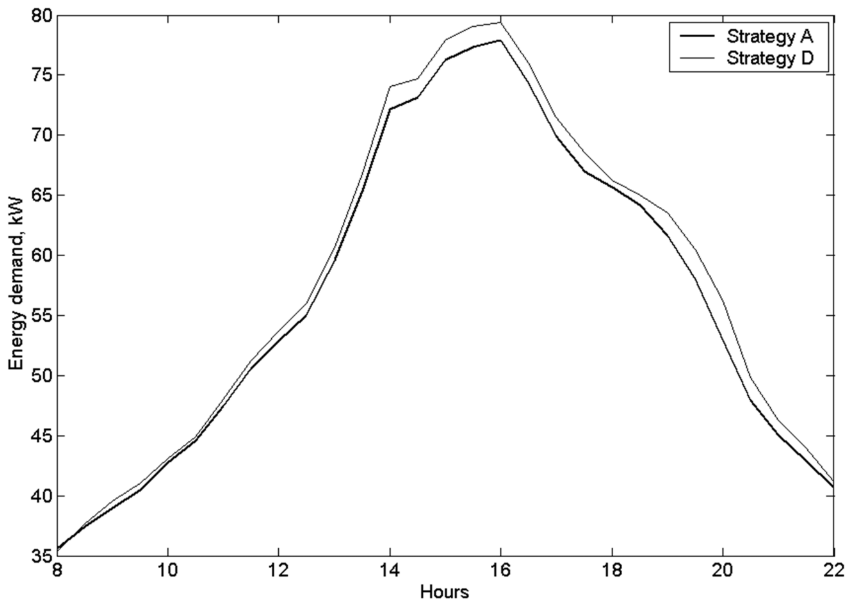

Figure 19. Energy demand for similar and different individual zone temperature setpoints (strategy D and strategy A, respectively).

tion and also enables further savings in energy use. From Figure 6, the energy demand is 41 kW for a PPD of 9.6% and 45.6 kW for a PPD of 5%. This means that to improve building thermal comfort from a PPD of 9.6% to a PPD of 5% at 5:00 p.m., the energy required is 4.6 kW. In the morning, this energy is only 1.6 kW. To obtain the required daily building thermal comfort, it is necessary to use the optimal selection tool that could select a low PPD in the morning and a high one in the early afternoon.

TWO-OBJECTIVE OPTIMIZATION

Thermal comfort is presented in this study as the objective function, but the optimization problem could be solved by defining the thermal comfort criterion as the constraint. Only one optimal solution would then be obtained, having a high PPD limit. The advantage of using two-objective problems is to minimize daily energy use by fluctuation of building PPD during occupied periods, taking into account the required energy demand. This is achieved by using the selection tool as follows.

At each period, this tool selects the solution requiring the least energy use (extreme right solution in Figure 6). The additional energy use required for improving thermal comfort PPD from this selected solution to the next is determined and compared with the permission setpoint recorded by the operator in the selection tool (*E* kW/PPD). If this further energy use requirement is less than the permission setpoint, the next solution will be selected; otherwise, the first solution is selected, and so on, for all the solutions. To evaluate the optimization results of the existing AHU-6 and modified HVAC using the one-objective and the two-objective problem, different permission setpoint values are settled in the selection tool during occupied periods for the two investigated systems. Daily energy use and daily thermal comfort (PPD) are calculated and illustrated as a function of the permission setpoint in Figure 20 for the modified HVAC and

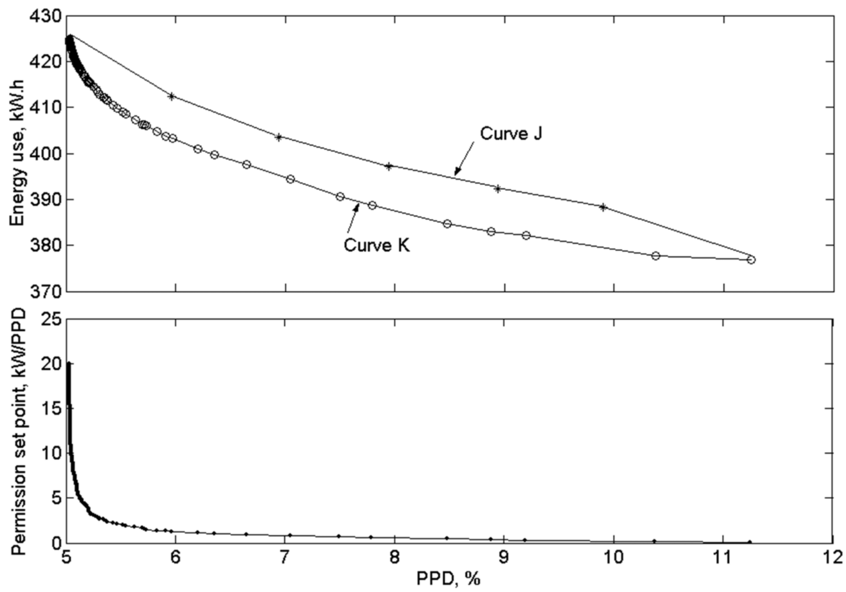

Figure 20. Optimal daily energy use obtained using the two-objective selection tool (curve k) and one-objective optimization (curve J) for modified HVAC system.

in Figure 21 for the existing system. When the permission setpoint is set at 1.18 for the modified HVAC (Figure 20), the daily PPD obtained is 6%, and the daily energy use is 403 kWh by using the two-objective optimization problem. However, the daily energy use is 413 kWh for the same daily PPD using the one-objective optimization problem. In both figures, curves K and J represent optimal solutions using the two-objective and one-objective optimization problems, respectively. While the permission setpoint is within the 0.1-20 range, the energy saving represented by the area between the two curves can be obtained by using the two-objective problem with the selection tool, as compared to the one-objective problem. If the value of the permission setpoint is chosen outside this range, the extreme solutions will be selected. In this case, it could be the same as in an objective problem by defining the thermal comfort criterion as the constraint. The optimal daily solution curve varies from day to day according to load and outdoor conditions. If the building load varies significantly during the day, as with the modified HVAC unit, the energy saving will be greater using the two-objective problem. If the online cumulative energy use during the day or current month exceeds the required level (peak energy use), the permission setpoint could be reset online to a lower value. In this concept, the monthly or daily energy use could be controlled to not exceed the required level by varying the permission setpoint.

CONCLUSION

The proposed optimization process was applied to an existing HVAC system that is installed at the École technologie supérieure. The setpoints, such as the zone air temperatures, the supply air temperature, the supply duct static pressure, the zone supply air temperature or reheat required, the minimum outdoor ventilation flow rate, and the chilled water supply temperature, are optimized for the existing and modified HVAC systems. The existing HVAC, abbreviated

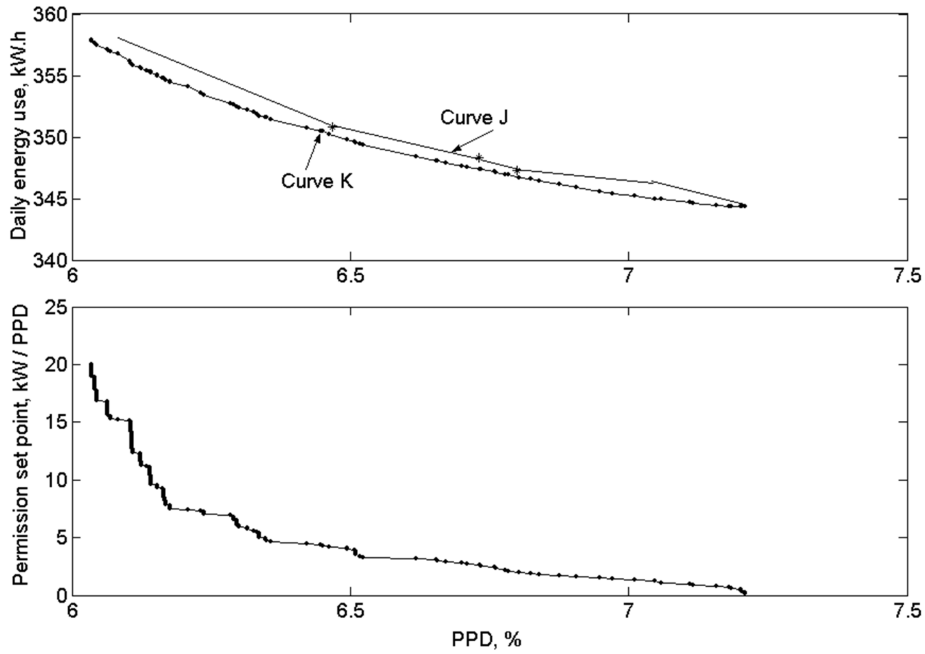

Figure 21. Optimal daily energy use obtained by two-objective selection tool (curve k) and one-objective optimization (curve J) for existing AHU-6.

AHU-6 and providing conditioned air to internal zones, was investigated. In this case, the optimization using a two-objective genetic algorithm was done for two summer months. The results then show that by comparing actual and optimal energy use, the optimization of a supervisory control strategy could save energy by 16% for two summer months while satisfying minimum zone airflow rates and zone thermal comfort. These results indicate that the optimization of the control strategy with required constraints could improve the operating performance of the existing HVAC system. However, three different control strategies are also evaluated for a modified HVAC system. The results show that a strategy that optimizes all controller setpoints, including zone temperature and zone reheat, performs better and provides more energy savings. Other results indicate that the application of a two-objective optimization problem could help control daily energy use or daily building thermal comfort while providing further energy use savings as compared to a one-objective optimization problem.

NOMENCLATURE

Acronyms

COP = coefficient of performance

ETS = École de technologie supérieure

GAP = genetic algorithm program

NSGA-II = elitist non-dominated sorting genetic algorithm

PMV = predicted mean vote

PPD = predicted percentage of dissatisfied

SBX = simulated binary crossover

Symbols

Q = airflow rate
η_c = distribution index
p_c = probability
p_m = mutation probability
P_S = static pressure
p_z = population size
ΔP_{duct} = pressure drop in supply duct
X = uncorrected fraction of the outdoor ventilation air in the supply system
Y = corrected fraction of outdoor ventilation air in the supply system
Z = ratio of required outdoor air to primary air in the critical zone

Subscripts

design = design condition
fan = supply fan
max = maximum value
sd = supply duct
z = zone

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