Building Simulation 2005

Ninth International IBPSA Conference
Montréal, Canada
August 15-18, 2005

SIMPLIFIED MODEL-BASED OPTIMAL CONTROL OF VAV AIR-CONDITIONING SYSTEM

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ABSTRACT

A simplified optimization process (SOP) for determining set points is proposed and evaluated using the monitoring data and model of an existing VAV system. Controller set points, such as supply air temperature, supply duct static pressure, and chilled water supply temperature, are determined by this proposed SOP in order to minimize energy use while respecting thermal comfort. Zone air temperatures are also considered in order to obtain further energy savings. The proposed SOP uses the simplified VAV system model and certain monitored variables available in the existing control system. The evaluation results on the existing VAV system obtained by the SOP are consistent with that reached through the detailed optimization process (DOP) proposed elsewhere. It is concluded that the SOP proposed here could be implemented to determine on-line controller set points without requiring detailed calculations, including the detailed VAV model and optimization program.

INTRODUCTION

Great efforts have been made in order to minimize the energy use associated with the operation of the HVAC system. One of these has involved the improvement of the VAV system performance through the optimization of controller set points (ASHRAE 2003). Much research has been done with a view to such optimization on multi-zone VAV systems (Nassif, Kajl, and Sabourin 2004a and Wang and Jin 2000). However, these require accurate VAV model and optimization algorithms. In addition, certain set points determined by these methods are not verified in real time by monitored data for representative local-loop control; an example is the calculated (or optimized) chilled water temperature, which is not verified on-line through the control valve opening. This paper thus proposes a simplified optimization process (SOP) for determining set points such as supply air temperature, supply duct static pressure, and chilled water supply temperature. The advantage of the SOP proposed over what has been done in previous research endeavors is that the SOP does not require a detailed VAV model and optimization program. Another advantage is that the monitored data for representative local-loop control is checked on-line, following which the controller set points are updated. In this case, the SOP ensures proper operation by opting for real situations with minimum energy use.

The simplified optimization process (SOP) is evaluated using the monitoring data and model of an existing VAV system. The simulated energy use is compared to that for the existing VAV system. The simulation results obtained by the SOP are also compared with the results obtained by our optimization process developed elsewhere (Nassif, Kajl, and Sabourin 2005), which we refer to as the *detailed optimization process* (DOP). The comparison indicates that the controller set points determined by the SOP are close or equal to the optimal ones (obtained by the DOP).

PROPOSED OPTIMIZATION PROCESS

This paper proposes a simplified optimization process (SOP) for determining the controller set points of a multi-zone VAV system. The SOP consists of: (*i*) controller set point strategies and (*ii*) a simplified VAV model, as shown in Figure 1. These two parts are presented and discussed in the next two chapters. The SOP is based on the simulation of the response of the VAV system performance to the proposed changes of controller set points. At each simulation step, three controller set point values are proposed and studied using the VAV model; this is done in order to select one value for each controller set point corresponding to the best performance of the VAV system, which then becomes the best controller set point. The three proposed controller set point values (proposed controller set points) are obtained as follows:

- The proposed values of supply air temperature set points include the current value and the values obtained by increasing or decreasing the current value by a small fixed amount (see Equation 13).
- Verifications of the proposed values are done by using the measured data, such as supply and zone air temperatures and zone airflow rates (see Equations 11, 12 and 14) in order to respect thermal comfort.

- The chilled water supply temperature and duct static pressure set points are then determined accordingly using strategies presented later in the chapter, *Strategies for Controller Set Points*.

Strategies are developed for determining the best duct static pressure and chilled water supply temperature set points at any proposed supply air temperature set point. In these strategies, the measured data for representative local-loop control, such as the positions of the cooling coil valve and the VAV box dampers, are used to incrementally update the controller set points (see Equations 8 and 10). The simplified VAV model developed and presented in the next chapter determines the change in the energy demands of system components for the three controller set point proposed values. The selected (or the best) controller set points corresponds to the lowest energy demands.

The SOP is evaluated by comparing the simulated energy use with the results obtained by: (*i*) the monitoring or model of an existing VAV system and (*ii*) our detailed optimization process (DOP) developed elsewhere (Nassif, Kajl, and Sabourin 2005). This DOP includes: (*i*) the detailed VAV model, (*ii*) the two-objective genetic algorithm optimization program, and (*iii*) an indoor thermal load prediction tool. The objective functions of the DOP are thermal comfort and energy use. To be able to compare the DOP with the SOP, the former must be modified to a one-objective optimization problem. The modified DOP presented here handles the total energy use as the objective function and the thermal comfort as the constraint. In addition, the load prediction tool was also simplified in the DOP presented in this paper. A simple load tool is applied, based on the assumption that indoor sensible loads are equal to the amount of cooling that terminal boxes provide as a product of the zone airflow rate and the difference in temperature between the supply and the zone air. Figure 1 shows the schematics of detailed and simplified optimization processes (DOP and SOP). In the DOP, the genetic algorithm program sends the controller set points to the detailed VAV system model at each optimization period (i.e., 5 minutes), where the energy use (objective function) is simulated and returned to the genetic algorithm program. The detailed VAV model is used to determine the energy use.

VAV SYSTEM MODEL

The detailed VAV system model developed and validated (Nassif, Kajl, Sabourin 2004b) is used by the DOP, while the SOP uses the simplified one. When the SOP is evaluated, the detailed component models are also used for simulation purposes.

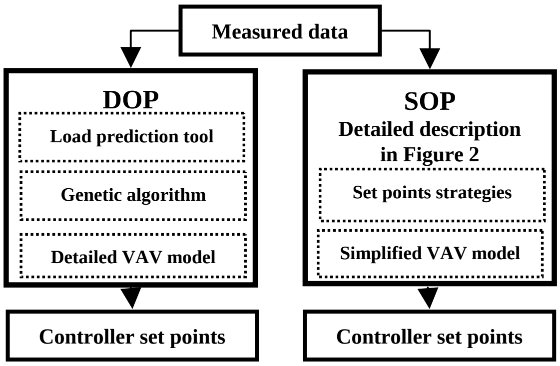

Figure 1 Schematics of the detailed and simplified optimization processes (DOP and SOP)

Simplified VAV component models

The simplified VAV component models used by the SOP are developed to be used in determining energy demands at any proposed controller set point. The variations of the fan, system heat, zone reheats, and chiller energy demands are determined through variations of the controller set points. During a small simulation time step, the thermal loads are assumed to be constant and the relation between the supply air temperature (*Ts*) and the zone airflow rate ($\dot{Q}z$), at a constant zone air temperature (*Tz*) could be presented as follows:

$$\dot{Q}z_{k-1}\cdot \rho \cdot c_p \cdot \left(Tz - Ts_{k-1}\right) = \dot{Q}z_k \cdot \rho \cdot c_p \cdot \left(Tz - Ts_k\right) \quad (1)$$

The specific heat c_p and air density ρ could be considered to be constant. The subscripts, *k* and *k-1*, respectively indicate the current and previous periods.

Fan energy demand ($\dot{W}f$) is a function of the fan airflow rate ($\dot{Q}f$) and of total static pressure. The latter is equal to the sum of the duct static pressure set point (*Ps*) and the remaining duct static pressure drop (VAV component duct), which is a function of the fan airflow rate and the flow coefficient C_f determined at design conditions. The relation between the current and previous fan energy demands is given by:

$$\dot{W}f_k = \dot{W}f_{k-1} \cdot \frac{\dot{Q}f_{k-1} \cdot \left(Ps_{k-1} + C_f \cdot \dot{Q}f_{k-1}^{\,2}\right)}{\dot{Q}f_k \cdot \left(Ps_k + C_f \cdot \dot{Q}f_k^{\,2}\right)} \quad (2)$$

The fan airflow rate ($\dot{Q}f_k$) is equal to the sum of the zone airflow rates ($\dot{Q}z_k$) that are calculated by:

$$\dot{Q}z_k = \frac{Tz - Ts_{k-1}}{Tz - Ts_k} \cdot \dot{Q}z_{k-1} \quad (3)$$

The zone and fan airflow rates are limited by their maximum and minimum values. When the VAV box dampers are not wide open, the zone air temperatures (*Tz*) could be assumed to be equal to their set points (i.e., 22.5°C). As shown in Equations 2 and 3, any change in the controller set points (*Ps* and *Ts*) generates a change in fan energy demand. It should be noted that the real-time measured data at the previous time (*k-1*) takes into account the variations of thermal loads and outdoor air conditions over time.

As mentioned above, the thermal loads and outdoor conditions are assumed to be constant during a small simulation time step. Thus, the chiller energy demand could be presented only as a function of chilled water supply temperature *Tw* (Wang and Jin 2000):

$$\dot{Wc}_k = \dot{Wc}_{k-1} \cdot \left[1 + Cc \cdot \left(Tw_{k-1} - Tw_k\right)\right] \quad (4)$$

The parameter *Cc* can be obtained at design system operation and is approximately constant for a chiller.

A comparison of the fan and chiller energy demands obtained by Equations 2 and 4 with those obtained with detailed fan and chiller models (DOP) shows the acceptable accuracy (within 5%), which improves when the variable changes (*Ps*, *Tw*, *Ts*) are small.

Since zone air temperature set points are kept constant, the zone air temperatures, and consequently the temperature obtained from the combination of outdoor and return air (*Tm*), could be assumed to be constant during a small simulation time step. Therefore, the system heat could then be determined:

$$\dot{Wh}_k = \dot{Qf}_k \cdot c_p \cdot \rho \cdot \left(Ts_k - Tm_{k-1}\right) \cdot \lambda \quad (5)$$

The actual zone air temperature will only differ from the set points (i.e. 22.5°C) in a limited number of zones (critical zones), as will be discussed later. Evidently, the heat system is not considered when the *Tm* is greater than *Ts*. The λ is the conversion factor to electricity energy demand. It is equal to 1 if the electricity heat system is used.

The zone reheat is turned on when the zone airflow rate reaches its minimum limit ($\dot{Qz}_{min}$), and air temperature in the zone is decreased to a minimum level (Tz_{min}). Zone reheats ($\dot{Wz}$) could be determined by Equation 1 (by adding local reheats):

$$\dot{Wz}_k = \dot{Wz}_{k-1} + \frac{\dot{Qz}_{max}}{\alpha_z} \cdot c_p \cdot \rho \cdot \left[\left(Ts_{k-1} - Ts_k\right) + \left(Tz_{min} - Tz_{k-1}\right)\right] \quad (6)$$

$$\alpha_z = \frac{\dot{Qz}_{max}}{\dot{Qz}_{min}} \quad (7)$$

where the $\dot{Qz}_{max}$ is the maximum zone airflow rate at design duct static pressure.

If the zone reheat is not used at a previous time ($\dot{Wz}_{k-1} = 0$), the effect of the supply air temperature decrease on the local reheat is seen only when the air zone temperature is close to Tz_{min}. If the difference (Tz_{k-1}-Tz_{min}) is less than the value of the change in the supply air temperature set point ($Ts_{k-1} - Ts_k$), the zone reheat will take a negative value, which then converts to zero. When the zone reheat is used at a previous time ($\dot{Wz}_{k-1} \neq 0$), the last term of the equation above becomes 0 ($Tz_{min}=Tz_{k-1}$).

Equations 2, 4, 5, and 6 show the variations in energy demand in response to the variations of the controller set points. It is known that the supply duct static pressure set point should be decreased for fan energy savings, and that the chilled water supply temperature should be increased for chiller energy savings. However, the supply air temperature set point, which has conflicting effects on these component energy demands, lays bare the optimization problem. Three supply air temperature values are proposed at each simulation time, and the associated total energy demand is simulated in each case. The selected value then corresponds to the least total energy demand.

STRATEGIES FOR CONTROLLER SET POINTS

As indicated earlier, the response of the VAV system performance could be simulated for any proposed controller set points. The supply air temperature set point increases or decreases by a small fixed value, while respecting the thermal comfort in the zones. The chilled water supply temperature and duct static pressure set points are then accordingly determined using the strategies developed below. These strategies ensure that the chilled water supply temperature and duct static pressure set points leads to best value and they provide a proper system operation. The zone air temperature set points in the SOP are not optimized but the zone air temperatures are investigated as presented next.

Zone air temperature

Typically, zone air temperatures are maintained at constant set points in the comfort zone during occupied periods. However, during unoccupied times, the set points are set up for cooling and set back for heating, in order to reduce energy use. A strategy using the optimization of individual zone temperature set points combined with other controller set points during occupied periods could further reduce system energy use (Nassif, Kajl, Sabourin 2005). The zones with the highest or lowest zone airflow ratios (*Ra*) are called critical zones. The *Ra* is the ratio of the zone airflow rate to the design maximum airflow rate. The number of critical zones selected with the highest *Ra* ratios is indicated by

N_{max} while the number of critical zones selected with lowest ratios *Ra* is indicated by N_{min}.

The strategy, which is applied by the SOP, involves keeping all zone temperature set points constant. However, the zone air temperatures in critical zones (N_{max}+ N_{mim}) may be moved away from their set points, but should always be kept within predetermined minimum or maximum levels (i.e. Tz_{max}= 24.5^{o}C and Tz_{min}=21^{o}C). This strategy allows further decreases or increases of the controller set points, and thus provides savings in energy use. In this case, the critical zones (extreme ones) have not priority to determine the controller set points with respect to thermal comfort in the zones. Many strategies could be applied to determine N_{max} and N_{min}. A standard deviation σ, of the normal distribution calculated by zone airflow ratios *Ra* could be used. The number of critical zones will then be high when the load distributions between zones are significant. When there is no thermal load distribution (zones perform as one zone), the N_{max} and N_{min} could be zero.

In order to compare the SOP with the DOP, the air temperature set points of the critical zones are optimized by the DOP, and are kept constant in other (non-critical) zones. In addition, N_{max}=3 and N_{min}=4 are assumed to be constant for the SOP and the DOP.

Duct static pressure set point

The duct static pressure will ensure the proper operation of the zone VAV boxes under varying load conditions. For a fixed duct static pressure set point, all the VAV boxes tend to close as zone loads and flow requirements decrease. Significant fan energy savings are possible if the duct static pressure set point is reset such that at least one of the VAV boxes remains open. Several different strategies based on this concept are proposed (Englander and Norford 1992). Our strategy in this paper takes into consideration this concept as well as the effect of changing other set point. Assuming that the duct static pressure ensures proper operation at design conditions with at least one of the VAV boxes fully open, the design duct static pressure (Ps_{des}) is considered as optimal under design operation conditions. The new (or optimal) static pressure set point (*Ps*) at off-design conditions could be simplified by the first term of the next equation (Nassif, Kajl, and Sabourin 2005):

$$Ps_k = \left(Ra_k^2\right)_{highest} \cdot Ps_{des} + a_{ps} \cdot \left(0.98 - \theta_{k-1}\right) \tag{8}$$

The highest *Ra* value is considered above, excluding the critical zones, while their air temperatures are lower than maximum values (Tz_{max}). It is clear that the duct static set point determined above is less than the design value at off-design conditions ($Ra < 1$). A further decrease in the duct static pressure set point, and consequently in fan energy, could be also obtained if the highest *Ra* values in the critical zones N_{max} are not considered while the air temperatures are lower than the maximum values. In order to take into consideration the real-time system operation, the damper position (θ_{k-1}) of the VAV boxes is checked. When the θ_{k-1} is less than 99% and greater than 97%, the term a_{ps} is equal to zero. Otherwise, it is a fixed value {i.e. $a_{ps} =0.01\ Ps_{des} =2.5\ Pa$}.

Chilled water supply temperature set point

With a fixed-speed pumping chiller system, the chilled water temperature set point must be adjusted to maintain all supply air temperatures of AHUs with a minimal number of cooling coil control valves in a saturated (full open) condition (ASHRAE 2003). In Chapter 41 of ASHRAE 2003, the chilled-water temperature (*Tw*) can be reset in response to sudden changes in load and supply air temperature set point:

$$Tw_k = Ts_k - \frac{PLR}{PLR_0} \cdot (Ts - Tw)_0 \tag{9}$$

The equation above assumes that the chilled water supply temperature associated with the last decision control (indicated by index *0*) was optimal. Assuming that the design chilled water supply temperature set point under the design operation conditions is optimal; the design condition could be used as the last decision control:

$$Tw_k = Ts_k - PLR \cdot (Ts - Tw)_{des} + a_v \cdot (0.98 - \theta v_{k-1}) \tag{10}$$

For the investigated existing system, the difference $(Ts\text{-}Tw)_{des}$ at design conditions is equal to 5^{o}C. To consider the real data, the last term is added. When the cooling coil valve opening θv_{k-1} is less than 99% and greater than 97%, the term a_v is equal to zero. Otherwise, it is a fixed value {i.e. $a_v =0.01(Ts\text{-}Tw)_{des} =0.05^oC$}.

The part load ratio *PLR* (current cooling coil load to design one) could be calculated from the water or air side depending on measured data. In this paper, the sensible thermal ratio is calculated by using the difference between the mixing plenum air temperature and the supply air temperature.

Supply air temperature set point

As mentioned earlier, the supply air temperature set point has conflicting effects on component energy demands. When its value is high, it may allow a higher chilled water supply temperature set point and associated improvement in chiller efficiency. However, when it is low, it will decrease fan energy use. As a result, the supply air temperature set point should not be set too high as it may provoke under-cooling in certain zones. When the minimum airflow rate introduced into internal zones (zone reheat does not exist) is limited in order to meet ventilation requirements, the low supply air temperature set point may also cause over-cooling in certain zones.

The supply temperature set point must thus be properly selected in order to maintain the required comfort in each zone. In the SOP this is done through a verification of any proposed supply air temperature. The verification (as we will see in Equation 14) is meant to ensure that the proposed change in the supply air temperature set point (i.e. $0.1^{o}C$) is at a level that is lower that the maximum allowed change, without affecting thermal zone comfort. Thus, it is assumed that thermal comfort is respected when the zone air temperatures in non-critical zones are maintained at their required set points while those in critical zones are within maximum and minimum limits [Tz_{min} - Tz_{max}]. This previous supply air temperature set point could be increased by a maximum value $a_{i,max}$ and decreased by a maximum value $a_{d,max}$, and these parameters are determined by Equation 1, where $\{Ts_k=Ts_{k-1}+a_{i,max}$ and $\dot{Q}z=\dot{Q}z_{\max}$ for $a_{i,max}$ and $Ts_k=Ts_{k-1}-a_{d,max}$ and $\dot{Q}z=\dot{Q}z_{\min}$ *for* $a_{d,max}\}$.

$$a_{i,\max} = \min\{Tz\cdot(1-Ra)+Ts\cdot(Ra-1)+(Tz_{\max}-Tz)\} \quad (11)$$

$$a_{d,\max} = \min\left\{Tz\cdot\left(\frac{Ra}{\alpha_z}-1\right)+Ts\cdot\left(1-\frac{Ra}{\alpha_z}\right)+(Tz-Tz_{\min})\right\} \quad (12)$$

The parameters above are determined for each zone, and the minimum values are selected to take into account the fact that the zone air temperature Tz must not be higher than Tz_{max} (or lower than Tz_{min}). It should be noted that all variables in Equations 11 and 12 are taken from previous measured data (*k-1*).

Three values of supply air temperature set points proposed to simulate the performance of the VAV are given by:

$$\begin{aligned}&(i).\quad Ts_k = Ts_{k-1} + a_o\\ &(ii).\quad Ts_k = Ts_{k-1} + a_i\\ &(iii).\quad Ts_k = Ts_{k-1} - a_d\end{aligned} \quad (13)$$

The parameters in Equation 13 are determined as:

$$\begin{aligned}&(1).\ a_i = 0.1 \quad a_d = 0.1 \quad a_0 = 0\\ &(2).\ if\, a_i \succ a_{i,\max} \Rightarrow a_i = a_{i,\max}\\ &(3).\ if\, a_d \succ a_{d,\max} \Rightarrow a_d = a_{d,\max}\\ &(4).\ if\ a_{i,\max} \prec 0 \Rightarrow a_i = a_o = -a_d\\ &(5).\ if\ a_{d,\max} \prec 0 \Rightarrow a_d = a_o = -a_i\\ &(6).\ if\, a_{d,\max} \prec 0\ and\, a_{i,\max} \prec 0 \Rightarrow a_i = a_o = -a_d\end{aligned} \quad (14)$$

If the zone thermal comfort is ensured, then a_o is equal to zero and a_i and a_d are equal to a fixed value, $0.1^{o}C$. When there is a local zone reheat, the $a_{d,max}$ is not calculated due to fact that the supply air temperature could be decreased without affecting the zone air temperatures (zone temperature is maintained by local reheat).

When the $a_{i,max}$ is negative, it means that the airflow rate reaches its maximum value and the zone air temperature is higher than Tz_{max}. Thus, the supply air temperature set point should only be decreased (number 4 of Equation 14). When the $a_{i,d}$ is negative, it means that the minimum airflow rate has reached its minimum value and the zone air temperature is less than Tz_{min}. Thus, the supply air temperature set point should be increased (number 5 of Equation 14).

To maintain air temperatures at their set points in non-critical zones, the supply air temperature set points should be limited (using Equation 1):

$$Ts_{\lim} = 22.5 - D\cdot(Tz_n - Ts_k) \quad (15)$$

For a high limit, the D is Ra and the Tz_n is the temperature in the zone having the highest Ra, but excluding the critical zones (N_{max}). For a low limit, the D becomes Ra/α and the Tz_n is the temperature in the zone having the lowest Ra, excluding the critical zones (N_{min})

SIMPLIFIED OPTIMIZATION PROCESS CALCULATIONS

As mentioned earlier, the SOP consists of controller set point strategies and a simplified VAV model. The goal of the former is to propose three sets of controller set points at each simulation time step, while the latter aims to calculate the associated energy demands for each set, and thus allow the selection of the set corresponding to the least energy demand. Figure 2 shows the simplified optimization process calculations.

Step#1: The values of $a_{i,max}$ and $a_{d,max}$ are determined by Equations 11 and 12, using previous data of zone airflow ratios (Ra) and zone and supply air temperatures (Tz and Ts). The parameters of equation 13 are determined using Equation 14. The N_{max} and N_{min} proposed here are 3 and 4.

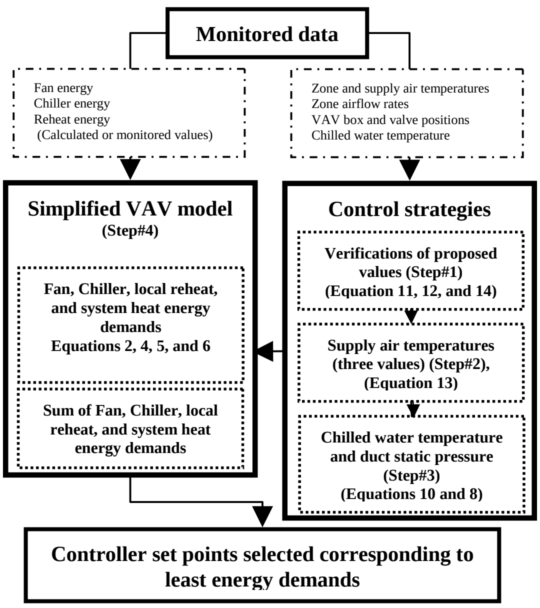

Figure 2 Simplified optimization process calculations

Step#2: The proposed supply air temperature set point values are determined by Equation 13 respecting the limits presented in Equation 15. Zone airflow rates (and ratios Ra_k) are determined by Equation 3. These values are limited within their upper and lower values.

Step#3: The chilled water supply temperature set point is calculated by Equation 10. *PLR* is determined using the measured mixing plenum and supply air temperatures (*Tm* and *Ts*) and the calculated fan airflow rate. The duct static pressure set point is determined by Equation 8.

Step#4: The fan, chiller, system heat, reheat, and consequently, total energy demands for the proposed values of controller set points, are determined by Equations 2, 4, 5, and 6 of the simplified VAV model. The best controller set points, corresponding to the least energy demand, are selected.

EVALUATION AND DISCUSSION

The SOP is evaluated on the existing HVAC system installed at the *École de technologie supérieure (ÉTS)* campus. Two air-handling units of multi-zone VAV systems (AHU-4 and AHU-6) are investigated. The AHU-4 meets the load for 68 west perimeter zones, while the AHU-6 meets the load for 70 interior zones. Controller set points are determined by the following three supervisory control strategies:

- Strategy $\mathbf{S_1}$: controller set points are exactly the same as in the existing system
- Strategy $\mathbf{S_2}$: controller set points are determined by the SOP
- Strategy $\mathbf{S_3}$: controller set points are determined by the DOP.

In the existing system (Strategy $\mathbf{S_1}$), the chilled water supply temperature and duct static pressure set points are constant at 7^{o}C and 250 Pa, respectively. The supply air temperature set point of the existing system (strategy $\mathbf{S_1}$) is set by the operator in the AHU-6 system (i.e. 14^{o}C). However, this set point is determined by applying the following strategy (for the AHU-4 system). The set point changes linearly within the 13 to 18^{o}C range, with the outdoor temperature between –20 and +20^{o}C. The set point calculated above is corrected by adding a value which varies linearly from –2 to +2^{o}C, which corresponds to the variation of the fan airflow ratio from 50% to 90%. The set point is always limited between 13 and 18^{o}C. However, in strategies $\mathbf{S_2}$ and $\mathbf{S_3}$, this set point is only limited at the low value (13^{o}C). The lowest supply duct static pressure and highest chilled water supply temperature set points are limited (150 Pa and 11^{o}C, respectively) The evaluations are done for three weeks under different weather conditions (summer, midseason, and winter), but are presented here only for three different days (day#1, day#2, and day#3). As mentioned above, a high supply air temperature may decrease chiller energy use and a low one may decrease fan energy use. The best selected supply temperature depends then on the value of the chiller and fan energy use. Thus, two following cases are assumed:

1. The chiller energy use is much higher than the fan energy use (about five times). The AHU-6 is studied in this case. There is one chiller serving all air handling units (including the AHU-6) that supply conditioned air to the ÉTS campus. It is assuming that the chilled water supply temperature is determined by the AHU-6 system. The monitoring data for two years showed that this assumption is quite realistic. As we will see next (see Figure 4), to minimize energy use, the supply air temperature set point tends to be high in order to save chiller energy, which becomes higher than the fan energy.
2. The chiller energy use is close to the fan energy use. The AHU-4 system is studied in this case, assuming that it is served by one small chiller. As will see next (see Day#1 in Figure 8), to minimize energy use, the supply air temperature set point tends to be low in order to save fan energy, which becomes higher than the chiller energy.

AHU-6 System

Figures 3 through 5 show energy demands, supply air and chilled water temperatures, and duct static pressure for three investigated strategies ($\mathbf{S_1}$, $\mathbf{S_2}$, and $\mathbf{S_3}$). It is noted that there are no local reheats installed in the interior zones served by the AHU-6 system. The controller set points and resulting energy demand determined by the SOP is very close to the value determined by the DOP. Since these two strategies start at 9:00 using real set point values (i.e., 14^{o}C for supply air temperature, 7^{o}C for chilled water supply temperature, and 250 Pa for supply duct static pressure), the controller set points determined by the SOP needs a certain amount of time to reach the optimal values determined by the DOP due to a small incremental change in supply air temperature. This could also happen when the thermal loads change significantly. The air temperatures in the zones must be maintained at the required set points. However, a certain amount of floating is allowed between maximum and minimum levels in critical zones. Figure 6 shows the zone air temperatures determined by the proposed SOP. It is able to maintain all required zone temperatures (except the critical zone temperatures) at the set point (22.5^{o}C). As has been stated, the values of *Ra* in the critical zones (N_{max}=3) are excluded in the calculation of the duct static pressure set point, which becomes less than 250 Pa, and further fan energy savings are obtained. Since the air temperatures in the critical zones are allowed to float within maximum and minimum values, there is consequently little restriction with respect to selecting the supply air temperature set point. This leads to further energy savings and a convenient system operation. For example, in Figure 6, the air temperature of one of the critical zones is at approximately 21^{o}C at 21:00. To maintain the temperature in this zone at 22.5^{o}C, the supply air temperature set point should be increased, which could lead to poor thermal comfort in other critical zones having higher airflow ratios.

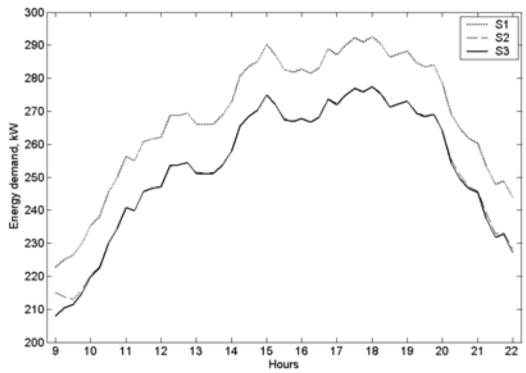

Figure 3 Energy demands in AHU-6 for summer day

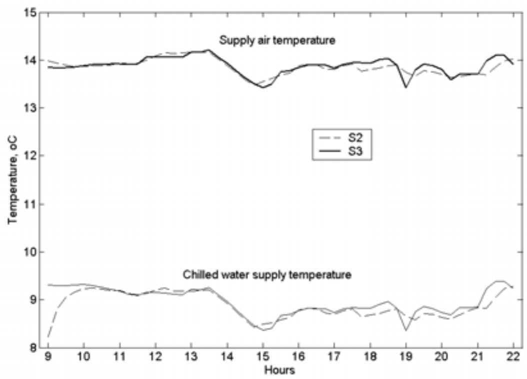

Figure 4 Supply air and chilled water temperature set points in AHU-6 for summer day

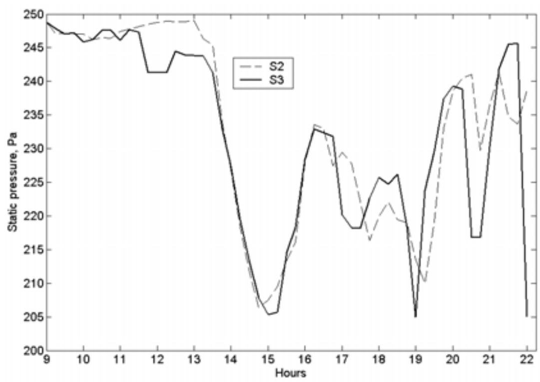

Figure 5 Duct static pressure set point in AHU-6 for summer day

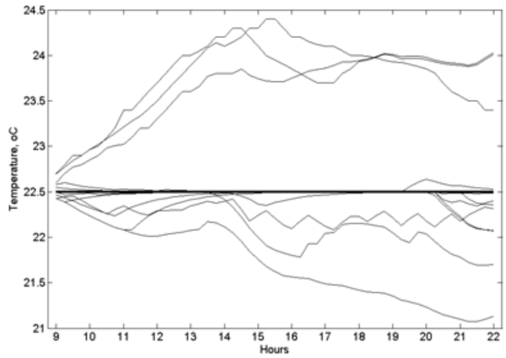

Figure 6 Zone air temperatures for strategy $\boldsymbol{S_2}$

The set points are also determined for midseason and winter days. Since the AHU-6 system serving the interior zones only yields cooling thermal loads, the results are not much different from what is discussed above, and are thus not presented here.

Comparing the results obtained by the three investigated strategies, we found that the SOP is able to successfully determine the set points that are close to those determined by the DOP considered to be optimal. The energy savings obtained for three weeks is 16.2% when the SOP is applied and 16.6% when the DOP is applied, versus the energy used by existing system.

AHU-4 System

The three investigated strategies are also evaluated on the AHU-4 system. Figures 7 and 8 show the energy demands and supply air and chilled water temperature set points. All zones served by the AHU-4 have the same orientation (south-west). Consequently, the values of *Ra* are quite the same, and thermal comfort restrictions are less significant. On Day#1, the supply air temperature and resulting duct static pressure set points for two strategies, $\mathbf{S_2}$ and $\mathbf{S_3}$, are the same. Given that the thermal loads, and consequently the zone airflow rates, are relatively low on Day#2 and Day#3, the duct static pressure set points are at their lowest value (i.e. 150 Pa), and so they are not illustrated. It should be noted that the cooling coil valve is closed before 11:00 on Day#2.

On winter days (Day #3), great energy savings are obtained by the DOP and the SOP. The fan energy demand does not vary significantly with the variation of supply air temperature, due to the saturation of the fan airflow at its low limit (40% of design value). The main energy demand is from zone reheats. The supply air temperature set points determined by the SOP and the DOP are higher than that for the existing system. As a result, the required local reheats are lower than for the existing one. The supply air temperature set points determined by the SOP and the DOP could be equal to the return and required outdoor air mixing temperature.

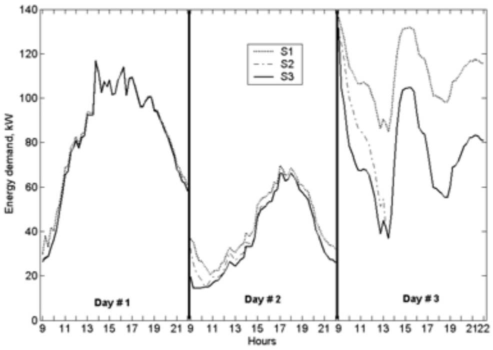

Figure 7 Energy demands of AHU-4 for three investigated days

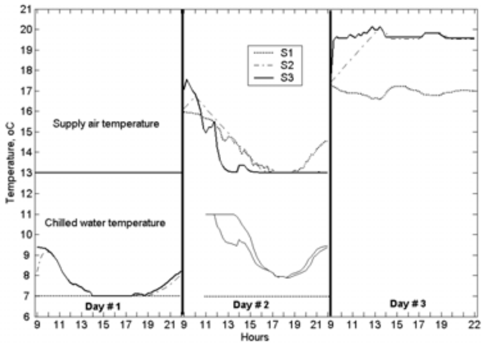

Figure 8 Supply air and chilled water temperature set points of AHU-4 for three days

We conclude that the performance of the SOP proposed in paper is close to that of the DOP. To improve the performance of the SOP, the dynamic incremental value could be used instead of a fixed value (0.1^{o}C) in order to increase its responses with any significant condition changes (Day#3 in Figure 8). For example, the fixed incremental value could be adjusted to take into consideration variations of the supply air temperatures during a sample previous period. The proposed strategy $\mathbf{S_2}$ (SOP) provides great energy savings compared to strategy $\mathbf{S_1}$ (existing one). As shown in Figure 7, the energy savings are highest on winter days.

The detailed VAV model used by the DOP is considered to be accurate. Thus, the controller set points, such as duct static pressure and chilled water temperature, ensure that at least one of the VAV box and cooling coil valves are wide open. However, since in reality, the VAV model is not completely accurate, these set points could not quite so optimal, in view of which, the SOP, by using the monitored data (VAV box and cooling coil valve openings) could perform better than the DOP unless the DOP were to use a very accurate and adaptive VAV model.

CONCLUSION

Evaluations of simulations done on the existing HVAC system show that the simplified optimization process SOP developed in this paper provides great energy savings compared to the strategy applied in the existing system. Energy savings obtained for three weeks could be 16.2% when the SOP is applied to the existing AHU-6 system. Comparing SOP with the detailed optimization proposed DOP, it is found that SOP is capable of successfully determining the set points that are close to those obtained by the DOP, and considered as optimal. The disadvantage of using the SOP instead of the DOP is that the controller set points determined by the SOP need a certain amount of time to reach the optimal values determined by the DOP when the outdoor conditions or thermal loads are significantly changed. This could be overcome by using a dynamic incremental value, which takes into consideration the variations of the supply air temperatures during a sample previous period. Thus, the proposed SOP could be implemented in order to determine the on-line controller set points without requiring detailed calculations, including the VAV model and optimization program.

REFERENCES

HVAC Applications 2003. ASHRAE handbook. American Society of Heating, Refrigerating and Air-conditioning Engineers, Atlanta, USA.

Brandemuel, M. J., Gabel, S., and Andersen, I. 1993. A Toolkit for Secondary HVAC System Energy Calculation. University of Colorado at Boulder.

Englander, S. L. and. Norford, L.K. 1992. Saving fan energy in VAV systems parts 2: Supply fan control for static pressure minimization using DDC zone feedback. ASHRAE Transactions, 98 (1), 19-32.

Nassif, N., S. Kajl, and R. Sabourin. 2005. Optimization of HVAC Control System Strategy Using Two-Objective Genetic Algorithm. *HVAC&R* 11(3): In press.

Nassif, N., S. Kajl, and R. Sabourin. 2004a. Two-Objective On-Line Optimization of Supervisory Control Strategy. *Building Serv. Eng. Res. Technol* 25(3): 241-251.

Nassif, N., S. Kajl, and R. Sabourin. 2004b. Modeling and Validation of Existing VAV System Components. *Proceeding of Esim, Canadian Conference on Building Simulation,* Vancouver, Canada.

Wang, S. and X. Jin. 2000. Model-Based Optimal Control of VAV Air-Conditioning System Using Genetic Algorithm. *Building and Environment* 35: 471-487.

Wang, S. 1999. Dynamic Simulation of Building VAV Air-Conditioning System and Evaluation of EMCS On-Line Control Strategies. *Building and Environment,* 34 681-705.