A STUDY ON HVAC SCHEME DESIGN SIMULATION METHODOLOGY

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ABSTRACT

Upon the decision made on scheme design, ie zoning of buildings and type of HVAC systems used; the rest of the system design require the capability to calculate the size and capacity of equipment. For the scheme design phase, traditional method relies heavily on the experience or common sense to zone the building and to select the type of air conditioning system. This paper proposed a simple approach of finding the hourly room temperatures conditions under the influence of a system operating ideally and comparing with the user selected design conditions, ie the degree of satisfaction can be obtained as a index to evaluate the performance of the system.

KEYWORD

Scheme design, HVAC Simulation, full year analysis

Introduction

Heating, ventilating and air conditioning (HVAC) scheme design is an important stage for the design of building environmental control system. It is the vital link between the previous conceptual design stage (building envelope's thermal performance analysis) and the next detail design stage (equipment selection, sizing, pipe and duct network analysis, etc.). The decision made in the scheme design stage could affect the operation performance, initial and operation cost of the systems. To carry out the design for this stage, the following parameters have to be determined:

- Air system type: eg variable air volume (VAV), constant air volume (CAV), fan coil unit (FCU), VAV + reheat, or new system type; eg FCU + seasonal VAV primary air systems, variable refrigerant volume (VRV) or water loop heat pump, etc.
- System zoing: eg interior and exterior zones, or zoning based on orienatations.
- Terminal unit capacity: FCU capacity, VAV box air flow range, air handling unit (AHU) air volume, etc.
- 2-pipe or 4-pipe water distribution systems, etc.

The current practice of scheme design normally adopted the typical design days analysis together with experiences of the designer. It may be the reason that poor designs bring the operational shortfall as reported in the following papers. Xue[1] reported that through a large number of on site survey and measurements, there are a large number of offices experiencing under/over cooling/heating penonumona, which happenes particularly in changeover seasons. Li[2] reported the zoning arrangement of a system serving both the machine room and the consultation room in a hospital. As the heat gains from the machine is much higher than the consultation room, the above unbalanced cooling/heating situations happened again.

Yan[3] reported that some hotels in Beijing had installed 4-pipe systems, but only two of the four pipes were used. It caused a large waste on initial cost of the entire systems and it also hindered the operation efficiency causing a further energy wastage on running the system.

From the above reports, it can be seen that considering thoroughly the annual system performance during the scheme design stage is of great helpful in the light of performance and energy efficiency optimization.

Scheme design whole year simulation process

It is obvious that the summer and winter design days calculations cannot guarantee the problem for year round operations, particularly for that during changeover seasons. The 8760 hours simulation, that considers all possible working conditions for an entire year, could provide a better picture of the system performance than the traditional design days analysis. For example, when determining the suitable zoning of the system, 8760 hours simulations for different zone arrangements can clearly show how many hours of the room temperature could not be maintained by the different system zoning, this can be a better expression in helping the designer to select a proper zone arrangement.

As shown in Figure 1, after the designer has decided the scheme details, simulations could be used to obtain the hourly results for the whole year for analysis. If the designer is not satisfed with the results, the scheme details could be modified and simulation can be applied again until the final goal is reached. The scheme design stage is actually a loop that consistes of design, simulation, revising design, and simulation again, until the scheme has reached the design criteria. Simulation has taken away all the labourous and complicated calculations from the designer, and the results from the simulation could aid the designer to making valuable decisions.

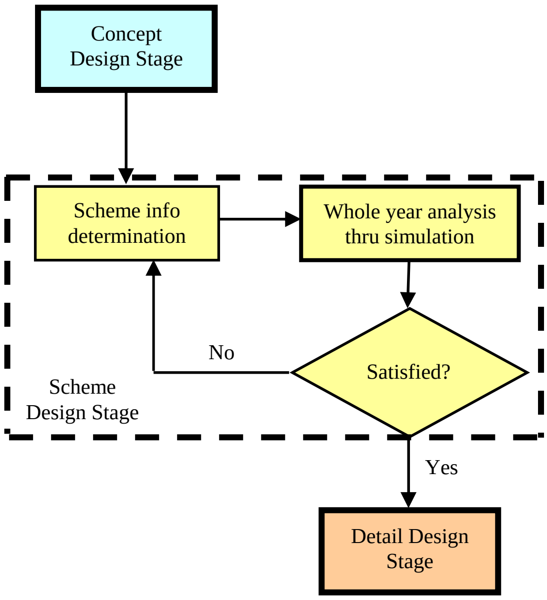

Figure 1 Detail layout of scheme design analysis process

This paper focuses on the problems that need to be solved in simulating the HVAC system performance in the scheme design stage. It proposes a methodology to simulate the performance of the HVAC system in this design stage. A case study is included in this paper to demonstrate the applications of the methodology for practical use. It could be shown by the case study that designers could use this methodology to optimise the best system types and zoning arrangement with enhanced technological supporting analytical results.

Currently, there are many building energy simulation tools available in the industry that could help designers to dynamically simulate the performance of HVAC systems, eg DOE-2[4] and EnergyPlus[5] from the US; and HASP[6] from Japan, etc. The discussion below highlights the scheme (or system) simulation approaches for DOE-2 and EnergyPlus.

DOE-2

DOE-2 provides CAV, VAV and other 14 types of built-in "standard" HVAC systems for scheme simulations, however new systems could be difficult to be defined. DOE-2 employed the sequential solution strategy "Load - Systems - Plants – Economic" (LSPE), as shown in Figure 2. Within this solution strategy, system simulationhas to be done with all steps linked together at the same time. It will be very difficult to simulate any one step alone since designers seldom know the detail performance characteristics of the cooling and heating plant, fan and pumps, etc., at the early design stage. Because of these needed design data, it would be better for DOE-2 to be used for the detail design stage in simulating and evaluating the energy consumption of a building.

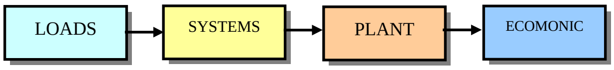

Figure 2 LSPE simulation solution strategy

For practical considerations, the room temperatures could not always be controlled at the room temperature set point when calculating the room loads. There's difference between the room loads and preliminary calculated loads. DOE-2 uses weighting factors method to adjust and correct the changes of loads. It defines three different building types (light, medium and heavy weight), corresponds to three different weighting factors. However, these building types are very different with the actual building envelope data in practice, it makes a significant difference between the calculated and the actual room temperature.

The thermal model applied in DOE-2 is single leaf wall response factors, which does not consider the heat balance of the room. Once the room temperature changes, the heat exchange of the building envelope is calculated by using steady state heat transfer equations in order to simpify the simulation. All these would make the calculation of the room temperature deviates to the actual physical conditions.

DOE-2 simplified the temperature control and room temperature as a linear step function. This funtion uses to adjust room loads and system cooling and heating capacities during the partload conditions. As this step function does not reflect accurately the physical process of the HVAC systems, it may also bring a larger deviation to the calculated results.

EnergyPlus

EnergyPlus provides a modular approach to define systems. Users could flexibly to "loop" system components in setting up systems or to create new system types from this modular approach. In order to improve the shortfalls of DOE-2, EnergyPlus considers the heat balance of the room in calculating the room loads. It also improves the sequential LSPE solution strategy as used by DOE-2 to simultaneous solution strategy of LSPE in providing feedbacks from different solution stages for a much more accurate result.

EnergyPlus is very powerful for simulating the performance of building environmental systems, while it still has some limitation on scheme simulation since it has the very similar shell as DOE-2. In EneryPlus, it needs to couple with building loads and plant simulations in order to carry out systems simulation. Purchased Air or Purchased Water can be used to simplify the HVAC part of the simulation, while it is still difficult to work with different practical design stages, and thus limits its contribution for the design process.

To reduce the work of calculations and to increase the computational speed, EnergyPlus uses the values of the previous time step, eg the outdoor air temperature, surface temperature and temperature of the adjacent rooms, etc, to replace the values of the current time step in solving the heat balance equations to find the room temperatures, this may increase the simulation error. And it may also be difficulty in reaching convergence in largy system simulation, due to the sensitivity of the length of time and space of each calculation step to the calculation result.

Identifying the problems for HVAC system simulations

To understand the building and its HVAC system's hourly data for the whole year, the following data must be defined:

- Building and weather data;
- Environmental control system scheme;
- Fans and pumps performance data;
- Air duct and water pipe network design,
- System control strategy.

However, in the scheme design phase, the datas that could probably be known are:

- Building and weather data;
- Environmental control system scheme: it includes the air system types selection, zoning arrangement, supply air flow range for each room, terminal reheat required or not, etc.

The datas that probobly could not be known are:

- AHU, cooling and heating plants detail characteristics and performances;
- Air ducting and water piping network details;
- System controls and regulations details, etc.

The main concern of the scheme design stage is whether the selected HVAC systems could maintain the room temperatures in the specified design conditions throughout the year. Since the room temperatures and humidities are primarily dependent on the supply air volume and the supply air conditions, the room conditions, controlled by the HVAC systems, have no direct association with the characteristics and performances of the AHUs, air ducting and water piping networks; and their control strategies. From these standpoints, the following assumptions for the scheme simulation methodology are proposed:

1. Assuming an ideal AHU that can supply any possible supply air conditions to the spaces. With this assumption, the energy consumption of the AHU is only associated with room temperatures and outdoor air conditions. Detail performance characteristics of the equipment will not be taken into account in this methodology. This ideal AHU forms the benchmark for comparing various schemes that the same AHU performance is assumed for all cases. The ideal AHU data obtained, eg supply and return air temperatures, can also be served as a reference for equipment selection in the latter stage;
2. Ideal control applied for the system is assumed , which means that the system can be regulated to any conditions perfectly. For example, the supply air flow rates of VAV systems can be controlled to any value within the specified range for each room, and no considering on the characterstics of VAV box controller is taken here. This approach can separate from the controller design with the scheme design. The latter one concerns the hourly system operations of the whole year, while the former one is considered within the time step of minutes or seconds.

Base on the above assumptions of idealising the equipment and control strategy, the scheme simulation results are more objectives and can be used for comparing various scheme configurations.

Mathematical models and solutions

According to the assumption of ideal equipment and control strategy, the the following problems need to be resolved by using scheme simulation

1. How to determine the supply air temperature that can satisfy most or all indoor air design ranges in different rooms;
2. For CAV systems without terminal reheaters, how to determine the hourly temperatures of rooms by using the calculated supply air temperature and design supply air flow rates;

3. For VAV systems with terminal reheater, How to define the supply air flow rates and terminal reheat capacities for each room so that the room design temperatures can be maintained.

Since most of HVAC systems rarely applied indenpent humidity control, in order to simplify the ccalculation, the determination of supply air flow rates, supply air temperature and terminal reheat capacity will not consider humidity effects throughout of this paper.

Room Heat Balance Model

the room temperature in the room *k* at a certain time could be written as:

$$\begin{aligned} t_k(\tau) = {} & t_{bzk}(\tau) + \sum_j \Phi_{j,\mathbf{0},k} t_j(\tau) + \Phi_{hvac,k} q_{hvac,k}(\tau) \\ & + \Phi_{hvac,k} C_p \rho G_{out,k}(\tau)(t_{out}(\tau) - t_k(\tau)) \\ & + \sum_j \Phi_{hvac,k} C_p \rho G_{jk}(\tau)(t_j(\tau) - t_k(\tau)) \end{aligned} \quad (1)$$

where

$t_k(\tau)$ = temperature of room *k*,

$t_{bzk}(\tau)$ = base temperature of room *k* (it excludes the effects of hvac systems, natural ventilation, ventilation and heat transfer from adjacent rooms),

$t_j(\tau)$ = temperature of room *j*,

$\Phi_{j,\mathbf{0},k}$ = the lumped affecting factor of room *j* to the temperature of room *k*,

$q_{hvac,k}(\tau)$ = cooling or heating power input to room *k*,

$\Phi_{hvac,k}$ = the lumped affecting factor of cooling or heating power to the temperature of room *k*,

$G_{out,k}(\tau)$ = Outdoor air ventilation rates,

$t_{out}(\tau)$ = Outdoor air temperature,

$G_{jk}(\tau)$ = Ventilation rates from adjacent room *j* to room *k*.

If the design conditions of supply air temperature, supply air volume and terminal reheat rates can be determined, the hvac system cooling or heating loads for system *l* of room *k* can be expressed as:

$$q_{hvac,k}(\tau) = C_p \rho G_{s,k}(\tau)(t_s(\tau) - t_k(\tau)) + q_{term,k}(\tau) \quad (2)$$

where:

$C_p \rho G_{s,k}(\tau)(t_s(\tau) - t_k(\tau))$ = cooling or heating loads of system *l* for room *k*,

$G_{s,k}(\tau)$ = supply air volume for room *k*,

$t_{s,l}(\tau)$ = supply air temperature for system *l*,

$q_{term,k}(\tau)$ = cooling or heating rates of terminal reheater for room *k*.

Substituting equation (2) into equation (1), gives:

$$\begin{aligned} t_k(\tau) = {} & t_{bzk}(\tau) + \sum_j \Phi_{j,\mathbf{0},k} t_j(\tau) + \Phi_{hvac,k} C_p \rho G_{s,k}(\tau) \\ & + \Phi_{hvac,k} q_{term,k}(\tau) \\ & + \Phi_{hvac,k} C_p \rho G_{out,k}(\tau)(t_{out}(\tau) - t_k(\tau)) \\ & + \sum_j \Phi_{hvac,k} C_p \rho G_{jk}(\tau)(t_j(\tau) - t_k(\tau)) \end{aligned} \quad (3)$$

Building Heat Balance Model

Since the supply air temperature for each room served by the same HVAC system would be the same, and all rooms in a building are connected by inter-zonal ventilations and heat transfer, The air temperatures of individual rooms must include inter-zonal effects of heat transfer and ventilations. To link up all rooms in a building and to find the room temperatures for each room, by re-writing equation (3), we can have:

$$\begin{aligned} & [a_{kk}(\tau) + G(\tau)] t_k(\tau) + \sum_j a_{kj}(\tau) t_j(\tau) + b_{kk} q_{term,k}(\tau) \\ & - G(\tau) t_s(\tau) + c_k(\tau) = \mathbf{0} \end{aligned} \quad (4)$$

The variables in equation (4)are explained as follows:

$$\begin{aligned} a_{kk}(\tau) = {} & \mathbf{1} + \Phi_{hvac,k} C_p \rho G_{out,k}(\tau) \\ & + \sum_j \Phi_{hvac,k} C_p \rho G_{jk}(\tau) \end{aligned} ;$$

$a_{kj}(\tau) = -\Phi_{hvac,k} C_p \rho G_{jk}(\tau) - \Phi_{j,\mathbf{0},k}$;

$b_{kk} = -\Phi_{hvac,k}$;

$G(\tau) = \Phi_{hvac,k} C_p \rho G_{s,k}(\tau)$;

$c_k(\tau) = -t_{bzk}(\tau) - \Phi_{hvac,k} C_p \rho G_{out,k}(\tau) t_{out}(\tau)$

If a building has *n* number of rooms and *m* number of different sets of air conditiong system, the room temperatures for each room can be written in the form of equation (4) and the following matrix could represent the whole building:

$$(A + B) \times T_r + B \times Q_{term} - G \times P \times T_s + C = \mathbf{0} \quad (5)$$

where:

$$T_r = \begin{bmatrix} t_1(\tau) \\ t_2(\tau) \\ \cdot \\ \cdot \\ t_n(\tau) \end{bmatrix} \qquad Q_{term} = \begin{bmatrix} q_{term,1}(\tau) \\ q_{term,2}(\tau) \\ \cdot \\ \cdot \\ q_{term,n}(\tau) \end{bmatrix}$$

$$G = \begin{bmatrix} \Phi_{hvac,1} C_p \rho G_{s,1}(\tau) & \mathbf{0} & \cdot\ \cdot\ \cdot & \mathbf{0} \\ \mathbf{0} & \Phi_{hvac,2} C_p \rho G_{s,2}(\tau) & \cdot\ \cdot\ \cdot & \mathbf{0} \\ \cdot & \cdot & \cdot & \cdot \\ \cdot & \cdot & \cdot & \cdot \\ \cdot & \cdot & \cdot & \cdot \\ \mathbf{0} & \mathbf{0} & \cdot\ \cdot\ \cdot & \Phi_{hvac,n} C_p \rho G_{s,n}(\tau) \end{bmatrix}$$

In equation (5), the martix P has elements of either 0 or 1. It reflects the association between systems and rooms. The rows in P is the room number n, the columns in P is the system number in m. For every row in the matrix P, there would be only one element could equal to 1. For example $P_{i,j} = 1$, then room i is associated with system j.

Upon the supply air temperature T_s, supply air flow rates for each room G and the capacities of terminal reheaters for each room Q_{term} have been determined, using equation (5), the room temperatures for each room T_r could be obtained:

$$T_r = (A+G)^{-1} * (G * P * T_s - B * Q_{term} - C) \qquad (6)$$

The next procedure will be, how to define the supply air temperatures for all systems T_s, supply air flow rates for all rooms G and terminal reheaters capcaities Q_{term} for all working conditions. The following examples describe the algorithm to define these parameters for the CAV system and VAV system.

Constant air volume (CAV) system simulations

Since The supply air volume is constant in the CAV system, in order to maintain the room temperature at part load conditions, the supply air temperature has to be ajusted during the scheme design stage to offset the heat gains or losses in a room and with minimal energy consumptions.

As the thermal conditions can be very different for different rooms at the same time, it is a very difficult solve analytically with the coupling of different room demands for the proper supply air temperature. Optimisation technique is employed to search for the best supply air temperature. The supply air temperature of the system could be controlled in a range of $t_s \in [t_{s,min}, t_{s,max}]$. A target function for CAV systems could be established as below:

$$x(t_s) = \alpha * \underset{i=1,n}{Max}(\Delta t_{r,i}) + |Q_{AHU}|$$

α is the weighting factor of the above function.

$\Delta t_{r,i}$ is the deviation to the room temperature setpoint:

$$\Delta t_{r,i} = \begin{cases} t_{r,i} - t_{rset\,max,i}\,, & t_{r,i} > t_{rset\,max,i} \\ \mathbf{0}\,, & t_{rset\,min,i} < t_{r,i} < t_{rset\,max,i} \\ t_{rset\,min,i} - t_{r,i}\,, & t_{r,i} < t_{rset\,min,i} \end{cases}$$

Q_{AHU} is the ideal sensible heat capacity for the air handling unit (AHU):

$$Q_{AHU} = C_p \rho \begin{bmatrix} (G_{supply} - G_{fresh}) * (t_s - t_{return}) \\ + G_{fresh} * (t_{supply} - t_{fresh}) \end{bmatrix}$$

α is the weight factor between room comfort and energy efficiency. By selecting a reasonable value of α, the selection of the system supply air temperatures could be set firstly to meet the room temperature design ranges then to miniumize energy consumption of the systems.

The expression of optimisation equation for CAV systems:

$$Min : x(t_s)$$

$$s.t. \qquad (A+G) * T_r + B * Q_{term} - G * P * t_s + C = \mathbf{0}$$
$$t_s \in [t_{s,min}, t_{s,max}]$$

Using the optimisation equation above, the optimised supply air temperature of the systems could be calculated. Room temperatures for each room can also be determined by using equation (6).

Variable air volume (VAV) systems simulations

Since VAV systems, can adjust both the supply air flow rates and the supply air temperature to offset the thermal loads for different rooms, the calculated best supply air temperature should not only meet the room temperature requirement, but also low down the energy consumption for both cooling/heating power and fan power.

Similar to CAV, a target function for VAV systems could be written as below:

$$x(t_s) = \alpha * \underset{i=1,n}{Max}(\Delta t_{r,i}) + \beta * |Q_{AHU}| + G_s$$

where: α and β are weighting factors

By selecting reasonable values of α and β, the optimisation procedure of the system can be firstly to satisfy room temperature ranges. The next optimisation target is the cooling/heating power of the AHU and then finally the supply air flow rates. By using the optimisation method such as golden section search, it will be very quickly to get the optimized result.

The expression of optimisation equation for VAV systems can be writen as:

$$\begin{aligned} &Min : x(t_s) \\ &s.t. \quad (A+G)*T_r + B*Q_{term} - G*P*t_s + C = \mathbf{0} \\ &\qquad t_s \in [t_{s,min}, t_{s,max}], \qquad G \in [G_{min}, G_{max}] \end{aligned}$$

In equation (6), when the system supply air temperature is determined, two other unknowns exist, they are supply air flow rates and room temperatures. By defining either one of them, the other could be calculated. For VAV systems, when the system supply air temperature is defined the following calculation strategy can be applied to find the room supply air flow rates and the room temperatures:

1. Calculating the room temperature t_{r1} by defining the room supply air flow rates at its minimum value G_{min};
2. If t_{r1} is inside the design range of room temperature, it idicates that the air flow at G_{min} could satisified the demand for the room. Then at the current system supply air temperature, the room supply air flow is set at G_{min};
3. If $t_{r1} < t_{rsetmin}$, it indicates that when the room supply air flow is at G_{min} , the room temperature is too cold. Then, set the room supply air flow to G_{max}, and recalculate the current room temperature t_{r2}:
   - If $t_{r2} < t_{r1} < t_{rsetmin}$, it indicates that room temperature decreases when increasing supply air flow rates. The room supply air flow rates should be G_{min}, and the room temperature is t_{r1};
   - If $t_{r1} < t_{r2} < t_{rsetmin}$, it indicates that the room temperature increases when increasingng supply air flow rates, but it still cannot reach the design conditions. Thus the room supply air flow rates should be G_{max}, and the room temperature is t_{r2};
   - If $t_{rsetmin} < t_{r2}$, it indicates that the room temperature increases when increasingng supply air flow rates, and the flow rate can be controlled to reach the room design conditions then the room temperature can be set at $t_{rsetmin}$, and substitute it in the calculation to calculate the room supply air flow G;
4. $t_{rsetmax} < t_{r1}$, it indicates the room air temperature is too warm when the room supply air flow is at G_{min}. By adjusting the supply air flow to the maximum G_{max}, the current room temperature t_{r3} could be obtained:
   - If $t_{rsetmax} < t_{r1} < t_{r3}$, it indicates, by increasing the air flow, the room temperature is raised. The room supply air flow rates is then G_{min}, and the room temperature is denoted as t_{r1};
   - If $t_{rsetmax} < t_{r3} < t_{r1}$, it indicates, by increasing the air flow, the room temperature is decreasing, but it still cannot reach the design conditions. The room supply air flow sets as G_{max} and the room temperature is t_{r3};
   - If $t_{r3} < t_{rsetmax}$, it indicates that the room temperature decreases when increasingng supply air flow rates, and the flow rate can be controlled to reach the room design conditions. then the room temperature can be set at $t_{rsetmax}$, and substitute it in the calculation to find the room supply air flow G.

From the above procedure, the system supply air temperature, room supply air flow rates and room temperatures can be determined. The best system supply air temperature and air flow rates are determined by the optimisation methods.

Case Studies

A fictitious office building is used for demonstrating the analysis procedure as described above for comparing the annual performance of various HVAC systems.

Building Details

The building plan is shown in Figure 3. The office building has a total floor area of 480 m^2. The exterior zone is 160 m^2 (denotes Room2 in Figure 3), the interior zone has an area of 256 m^2 (denotes Room1 in Figure 3). The corridor is 64 m^2. The floor height is 3.6 m. The U-values for external walls and double-glazing external windows are 1.4 $W/(m^2K)$ and 3.1 $W/(m^2K)$ respectively. The external wall window ratio is fixed at 0.5.

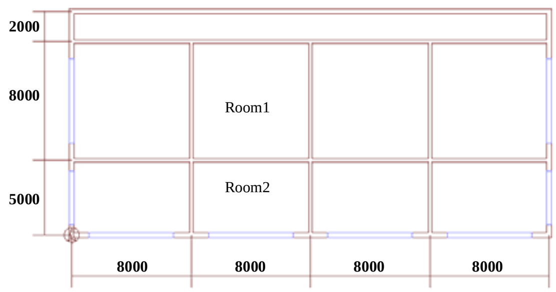

Figure 3 Layout plan of the Case Study Building (Unit: mm)

The office building is located in Beijing, China. The densities of occupancy, lighting and small power are taken as 0.1 person/m^2, 10 W/m^2 and 10 W/m^2 respectively. The schedules for occupancy, lighting and small power are shown in Figure 4. The indoor design temperature range is 22 °C – 25 °C and room design humidity range is 40 % - 60 %. The HVAC system is considered to be run continuously throughout the year.

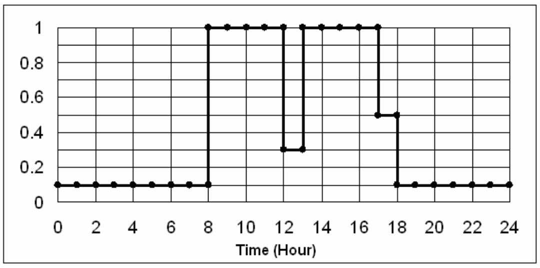

Figure 4 Occupancy, lighting and small power schedule

Load Calculation Results

From the simulated hourly load results, we can see that Room1 in Figure 3 is in the interior zone ,which is mainly affected by internal heat gains, only cooling demand is requires all over the year as shown in Figure 5. In the exterior zone of Room2 as shown in Figure 3, Since this room has a large window area, the hourly load in the room influence greatly by outdoor air temperature and solar radiations, as in Figure 6.

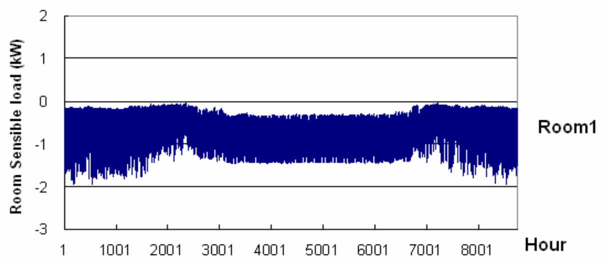

Figure 5 Room1 annual hourly loads

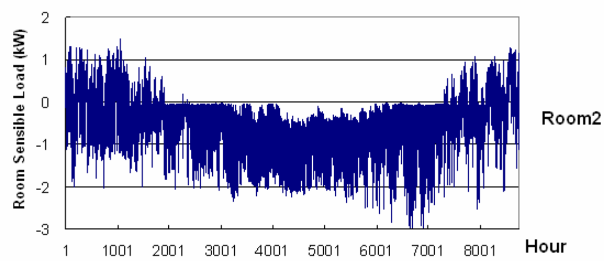

Figure 6 Room2 annual hourly loads

Preliminary scheme

The preliminary scheme for the building adopts a 4-pipe CAV system and its zoning arrangement is as shown in Figure 7. The supply air flow rates for each room is at 8 ach. There will be no terminal reheaters and the supply air temperature from the system is ranged from 14°C to 32°C.

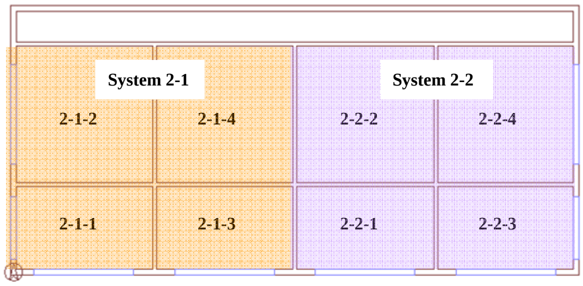

Figure 7 System zoning diagram

From the methodology presented above-, which had been fully incorporated in the current version of DeST, the dissatisfied hours, during which the room temperature calculated does not fall within the specified design temperature range, could be obtained for analysis as shown in Table 1. The hourly supply air temperatures for System 2-1 is shown in Figure 8. Figure 9 shows the hourly supply air temperature (SUPPLY_T) and room temperatures on 29 July.

Table 1 CAV systems annual dissatisfied hour

System No.	2-1	2-2
Hour of dissatisfied	1196	1003

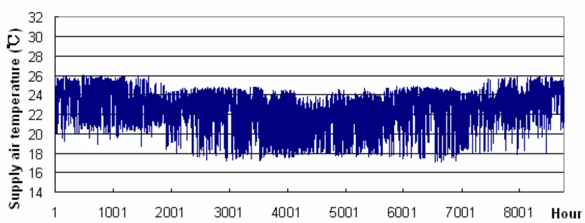

Figure 8 System 2-1, annual hourly supply air temperatures

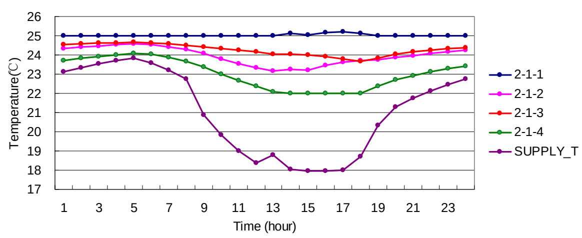

Figure 9 System 2-1, supply air and room hourly temperatures on 29 July

In Figure 9, the room temperature design range is betewwn the maximum value of 25°C (red line on the figures) and minimum value of 22°C (blue line on the figures). The room temperature at 1600 hr on 29 July of room 2-1-1 served by system 2-1 is found to be exceeding the maximum design room temperature of 25°C, while the supply air temperature is 17.99°C. Figure 10 shows the room temperatures for all rooms served by system 2-1.

Room 2-1-1 is located in the exterior zone and facing south-west. It is the warmest room in the afternoon. If the supply air temperature is lowered at this moment, despire the warmer room 2-1-1 could received more cooling but the cooler room 2-1-4 might be overcooled. Clearly, system 2-1 could not couple with the simultaneous cooling and heating demands for all the rooms.

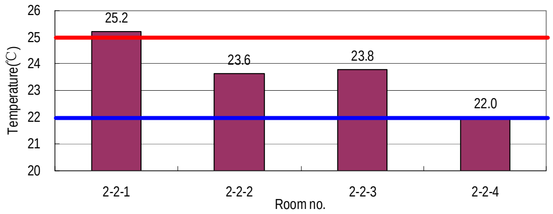

Figure 10 CAV system 2-1, room temperatures at 1600hr on 29 July

VAV systems

The first scheme proposed above has a large number of dissatisfied hour. It is therefore not possible to meet the design conditions for all rooms for the whole year. A change of system type is proposed to improve this situation. A VAV system is used and the range of supply air flow rates is set at 2 – 8 ach while all other design parameters remain unchanged.

The calculation was done for 8760 hours and the results are shown in Table 2. It is obvious that the VAV systems has a less dissatisifed hour

Table 2 VAV systems annual hour of dissatisfied

System No.	2-1	2-2
Dissatisfied hour	299	277

As at 16:00 hr of 29 July, the room temperatures of each room of system 2-1 are shown in Figure 11. The system supply air temperature is 15.67°C. Through the regulation of the supply air flow rates for each room (the red line and the blue line in Figure 12 are the upper and lower limits of the air change rates respectively), the system can eliminate the simultaneous over-cooling and over-heating of the rooms. All room design room temperatures can be met as served by system 2-1.

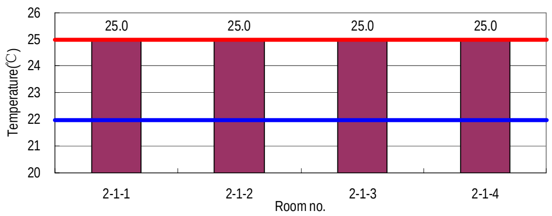

Figure 11 VAV System 2-1, room temperatures at 1600 hr on 29 July

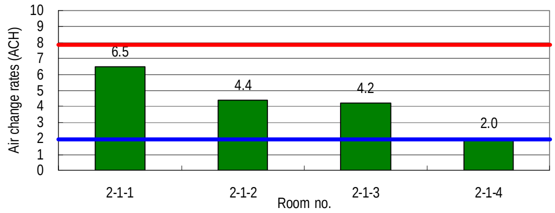

Figure 12 VAV System 2-1, air changes rates for rooms at 1600 hr on 29 July

By appling VAV sytem, ,the system dissatified hour reduces greatly. However, there are still some dissatified hours existes due to the large difference of room loads As in Figures 13 and 14, in the evening of 15 Feb, the load of Room 2-1-4 in the interior zone is very different with the others. Even when the minimum supply air flow rates is applied, the room temperature is still higher that the maximum design room temperature.

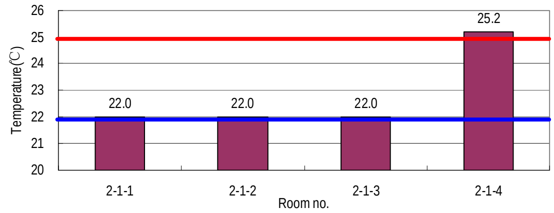

Figure 13 VAV system 2-1, room temperatures at 0600hr on 15 February

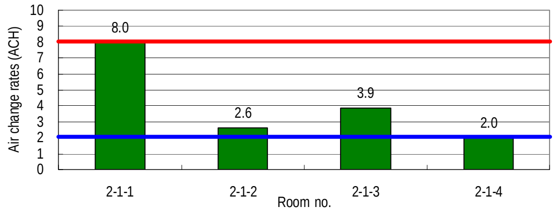

Figure 14 VAV system 2-1, room air changes rates at 0600 hr on 15 February

VAV + Terminal reheat systems

To improve the performance of the VAV system, a 1 kW reheater is added for each room, while all other system parameters remain unchanged. Applying the same analysis procedure mentioned above, the dissatified hour for the VAV + terminal reheat systems is shown in Table 3.

Table 3 VAV + terminal reheat systems annual hour of dissatisfied

System No.	2-1	2-2
Dissatisfied hour	4	0

It can be seen that the VAV + terminal reheat systems can fulfil almost all the rooms cooling/heating demand allover the year . For example, the supply air temperature at 08:00 am on 14 Nov for System 2-1 is 15.66°C, figures 15, 16 and 17 show the conditions for various rooms. The red line and the blue line shown n Figure 17 are the maximum and minimum range of reheating power respectively For the cooler room 2-1-1, the terminal reheater could bring the room temperature within the design conditions. .

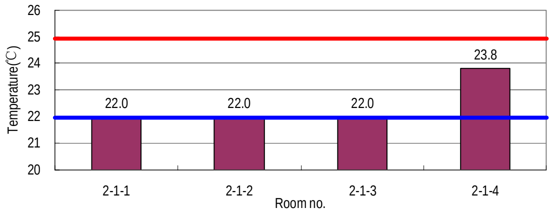

Figure 15 VAV + terminal reheat system 2-1, room temperatures at 0800 hr on 14 November

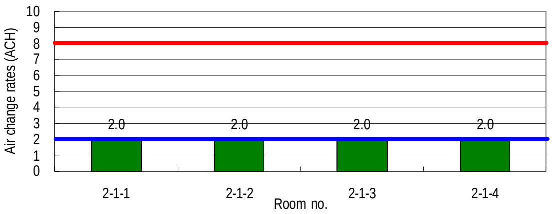

Figure 16 VAV + terminal reheat system 2-1, room air changes rates at 0800 hr on 14 November

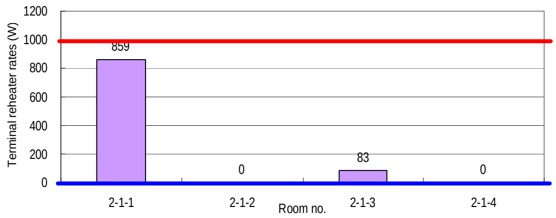

Figure 17 VAV + terminal reheat system 2-1, reheater rates at 0800hr on 14 November

Change of zoning arrangement

From the calculation results above, we can see that by changing the system type to VAV + terminal reheat we can greatly reduce the dissatisfied hour. However, the initial cost and operation cost of the system increases. So some other aspects should also be improved. As in Figure 18, the zoning arrangement is changed so that a system serves the interior rooms and another system serves the exterior rooms. The same CAV system is applied and all other design parameters remain unchanged in order to highlight the zoning change influence.

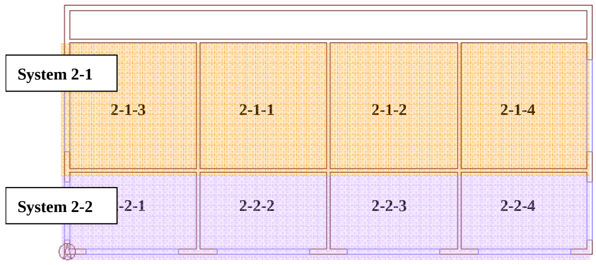

Figure 18 New zoning arrangement diagram

Table 4 shows the results of dissatisfied hour for the two CAV systems.

Table 4 New zoning CAV systems annual hour of dissatisfied

System No.	2-1	2-2
Hour of dissatisfied	10	3

It can be seen that by appling of proper system zoning scheme, without the change of the system type, the calculated dissatisifed hours for the new scheme has been great reduced and it could basically fulfill the design requirements for the entire year.

In figures 19 and 20, the results of the new zoning system arrangement of systems 2-1 and 2-2 at 1600 hr on 29 July indicate that the desired room temperature ranges can be met for all rooms.

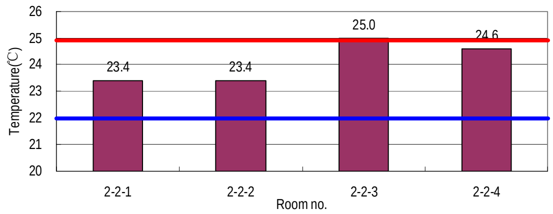

Figure 19 New zoning system 2-1, room temperatures at 1600 hr on 29 July

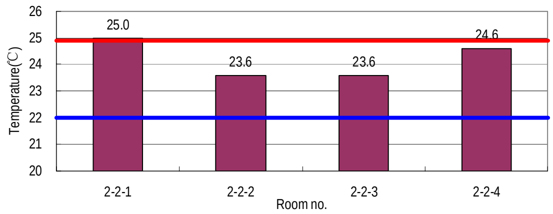

Figure 20 New zoning system 2-1, room temperatures at 1600 hr on 29 July

Conclusion

In the course of scheme design process, the work includes: defining system type, system zoning, terminal reheat capacity, water network arrangement, etc. These details contribute significantly to the initial and operation costs, increased efforts must be put in this stage to better understand the system behaviour. As for the current practice, it is still mainly based on experiences together with typical design day estimations. It may be the cause for various major inherent system design problems making the system operates in low efficient and somehow cannot meet the design requirements. Through detail analysis of the physical process of two major HVAC system types, this paper proposed a methodology with detailed models and solution procedures for finding the best system for the scheme design stage. With a case study, the methodology was tested and the results show the disatified hour to the design room temperatures provided a handy comparison index for different system types and zoning arrangements.

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