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Effect of room modelling and sensor position on performance assessment of variable air volume control systems

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Variable air volume (VAV) systems are commonly used for air conditioning in buildings. The testing of different control strategies and controllers for this application has been a main concern in several simulation studies. In these simulations much attention has been paid to the accuracy of the models of the VAV system while very simple models are often used at the room level. These room models assume that the air in the room is perfectly mixed, even when the prevailing conditions are not at all homogeneous. Another important issue when testing controllers is the method used to assess the control performance. Since it makes no sense to use very detailed room models, if the method of performance assessment is insensitive to the way in which the room is modelled, both issues must be treated simultaneously. The paper considers the problem of assessing the performance of VAV systems that use ceiling diffusers. The study includes the development of a convection model for a room. Its complexity is reduced to a minimum to allow detailed dynamic simulation of a whole building, complete with its VAV system and other building services (e.g., sun-blinds, lighting, etc.). Since airflow in a room depends strongly on the type of diffuser that is used, the study is carried out for both round and slot diffusers. Results are presented that show that the room model and the position of the sensor affect the performance in different ways depending on the diffuser type and the operating mode. It is concluded that there are only small differences in terms of thermal comfort but significant differences in terms of overall energy consumption. The effect of sensor position on energy consumption is found to be a function of steady state temperature differences.

1 Introduction

The testing of control strategies and controllers for variable air volume (VAV) systems is often carried out via simulation[1–3] or emulation techniques.[4] The accuracy and applicability of the models of the central plant and air distribution system has already reached a high level. On the other hand, the modelling of conditions in the rooms of the building is still simplified to some extent. While conduction and radiation modelling is sufficiently detailed, convection modelling is usually poor and often assumes that the air in the room is perfectly mixed.

In this paper the effect of convective room modelling on the results of controller tests is studied. The modelling of room convection affects the temperature measured by the control-

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ler's sensor, the heat losses from the room and bypass effects. A zonal room model is developed for two characteristic diffuser types. A two-stage approach to comparing the 'well-mixed' and zonal room models is taken. First, simulations are carried out for a single room. A comparison is made of the responses to a step change in the internal heat gains. Second, the models are incorporated into a simulation of an entire multi-zone building equipped with a VAV system, in order to carry out more realistic performance tests over longer periods of time. Different methods of assessing the control performance are compared to see if the use of a more detailed zonal model is justified or if a simple 'well-mixed' room model is sufficient. The effect of sensor position on energy consumption and on controller tuning is examined in the last part of the paper.

2 Assessing the performance of a VAV control system

The use of computer simulation to assess the performance of a VAV control system is a challenging problem. A number of different methods have been used to classify controllers according to their ability to control room conditions.[5] However, the room models used for these tests assume that the condition of the air inside the room is homogeneous. Particular problems arise when more detailed convection models are used that are capable of modelling non-homogeneous conditions inside the room.

Key issues for realistic controller tests are the definition of the:

- *Test procedure*
  A procedure (test conditions and load profiles) has to be chosen that is realistic but, at the same time, challenging for the controllers under test. A step change in the temperature set-point would provide a challenging test but it is not sufficiently realistic. Actual weather conditions combined with occupancy profiles would be more realistic but are probably not challenging enough. Step changes in the internal heat gains in the room (convective and radiative) are both realistic and challenging, if steps are taken to represent 'worst case' disturbances.
- *Location of control sensor*
  This issue can be ignored if the convection model assumes homogeneous conditions in the room. Otherwise a location has to be defined for the control sensor (see Figure 1). In reality, it is most likely that the sensor (and controller) will be located at a convenient height on one of the internal walls, even though the conditions in the occupied zone are to be controlled. Depending on the airflow pattern in the room, this means that the sensor could be placed in a natural convection boundary layer or in the jet or plume produced by an air diffuser or emitter.
- *Reference position for assessing the control performance*
  The reference position or positions (see Figure 1) selected for the performance assessment can affect the results. A realistic single position is in the occupied zone at the height where it is most important for the thermal comfort to be closely controlled. On the other hand it might be argued that a controller should be judged on its ability to control the temperature it senses. It must therefore be decided whether the controller and sensor are to be considered as separate units or the combination of controller and sensor is to be tested. In the latter case, it is necessary to differentiate between the sensor location and the reference position(s) selected for performance assessment.
- *Criteria to be used for performance assessment*
  A representative variable has to be chosen for the assessment of controller performance: for example, the air temperature, the resultant temperature or a comfort index. However, simplified models do not always provide all of the data needed to calculate the comfort indices (air velocity, clothing, activity, etc.) and assumptions have to be made that have an influence on the results. The choice of a

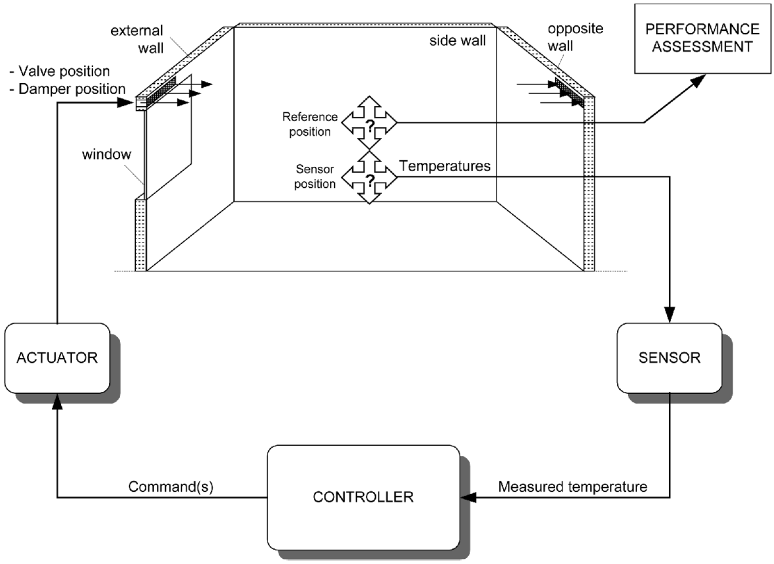

Figure 1 Assessing the performance of a controller

particular index might therefore lead to conclusions that are not universal. The most appropriate way of dealing with time-varying or spatially varying performance criteria may also affect the results. The maximum value or the mean value, over a specified time period and over each of the rooms or zones in the building under consideration may be used.

- *Design of the temperature sensor*
  The design of the temperature sensor used with the controller is also a major factor that can influence the results of the performance tests. The measurement will depend on the heat transfer characteristics of the sensing element, the sensor enclosure and the room. The location of the sensor in its enclosure, the design of the enclosure, and convective and radiation heat transfers from any electronic hardware installed inside the enclosure, all affect the output of the sensor. The convective, conductive and radiative components of the measurement must be specified if simulation is to be used for performance assessment.

3 Zonal modelling of the room

Zonal modelling[6–8] has been shown to have many advantages in comparison with other approaches such as ‘well-mixed’ models and models based on computational fluid dynamics (CFD). In a zonal model, the internal room air volume is divided into a relatively small number of sub-volumes and only the conservation of mass and conservation of energy equations are solved. Such models are able to represent the convective phenomena associated with nearly all types of HVAC equipment and, because the model complexity is kept at a reasonable level, they allow entire multi-zone buildings to be simulated dynamically. They are also able to address the issues associated with the testing of controllers that were raised in the previous section.

For this study, a three-dimensional zonal model of the room was developed in which the main zone and wall sub-volumes are each divided into three horizontal layers (see Figure 2). The sub-volume labelled ‘ceiling layer for jets’

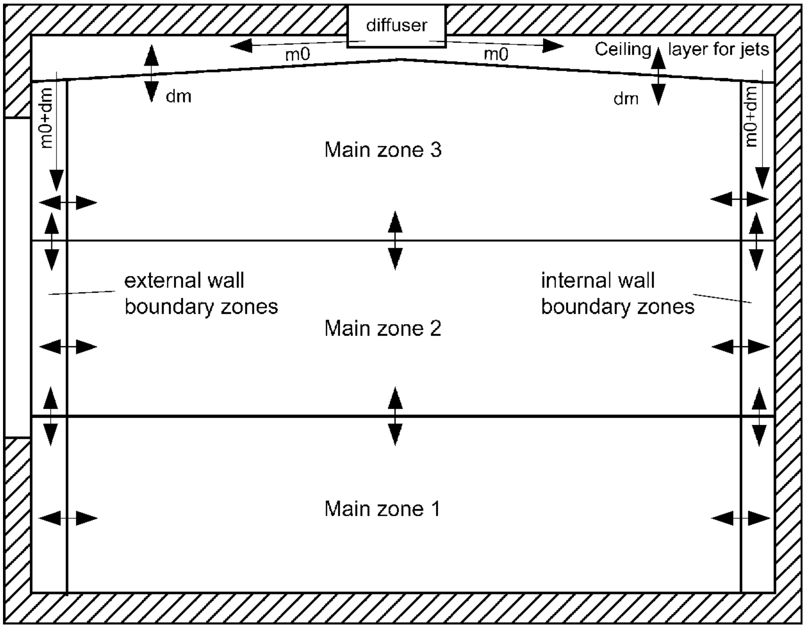

Figure 2 Structure of the zonal room model for a room with a round ceiling diffuser

represents all of the air arriving from plumes or jets. The air layers at the centre of the room represent the conditions in the occupied zone. The air volumes near the walls represent zones of natural, mixed or forced convection at the internal and external walls. Standard correlations[9,10] are used to calculate the airflow rates in the jets or plumes and near the walls, and to estimate the airflow between these zones and the main zones. Flow balances over all zones are then used to determine the airflow between the main zones. The zonal model, on which the room model is based, has been validated against experimental data obtained from a test room.[10]

The behaviour of two types of ceiling mounted diffusers[11] is simulated in this study: the slot diffuser and the round diffuser. The throw of the jet is selected so that it is slightly bigger than the characteristic length of the room (the distance between the diffuser outlet and the next wall intersecting the jet trajectory, or the intersection of the jets from two adjacent diffusers) to prevent the jet dropping into the occupied zone when the system is in cooling mode. Standard correlations,[11] which define the relationship between the airflow rate in the main characteristic regions of the jet[12] and the distance from the outlet of the diffuser, are used to calculate the air entrainment, *dm*, between the upper main zone and the 'ceiling layer for jets' (see Figure 2). The values of the parameters of the equations are derived from a particular manufacturer's data, as it is difficult to present general results even for a single type of diffuser.

The ceiling jet will affect the temperature in the boundary zones at the walls. When there is a cooling demand in the room, the cold jets are characterized by positive buoyancy forces (i.e., they will accelerate downwards) and the jet flow is assumed to fall down, along the wall, to the lowest main zone sub-volume. Different convective phenomena occur in heating mode because a warm jet arriving at a wall is subject to negative buoyancy forces. The vertical depth to which this warm jet penetrates the room along the wall depends on the temperature and flow rate of the jet.[13] The jet will affect the temperature measured by the control sensor if it reaches the sub-volume in which the sensor is located. Otherwise, it is assumed that the sensor will be immersed in a natural convection boundary layer.

The model of room convection is coupled with three other simulation models: the model of the heat conduction through the building envelope, the radiation model and the model of the air-conditioning plant. The conduction model, which is based on a nodal representation of each layer, allows each element of the envelope to have up to four different material layers. Five different envelope elements are modelled: floors, ceilings, internal walls, external walls and windows. The radiation model uses a mean radiant temperature node to calculate the radiative heat transfer between the surfaces of the envelope. The simulated air-conditioning plant consists of a central air-handling unit, with supply and return fans, and pressure independent VAV boxes with reheat coils that provide air to the diffusers in each of the rooms in the building. In cooling mode, the simulated room temperature controller varies the set-point for a velocity controller, which regulates the airflow rate through the VAV box. The velocity controller adjusts the damper position to control the airflow rate into the room. In heating mode, the simulated airflow rate is held at its minimum value and the room temperature is controlled by varying the position of the valve controlling the hot water flow rate through the simulated reheating coil. The models of the air-conditioning equipment and controls were developed as part of a previous project on the design of integrated building control strategies.[14] Design data are used to evaluate the parameters of the models used in the simulation modules.

The simulation based on a 'well-mixed' model, which acts as a benchmark during the tests described in the following sections, uses the same conduction, radiation and air-conditioning plant models as the simulation based on the zonal room model but assumes perfect mixing of the air in the room (i.e., infinite flow rates between the sub-volumes of the zonal model).

4 Test cases and test procedure

Two series of tests are carried out:

- simulation of a single room under closed loop control with a step change in the internal heat gains;
- simulation of a multi-zone office building using real weather data and internal gains.

The effect of using different sensor positions is studied in both cases. Each of the simulations is carried out using: (i) the 'well-mixed' room model; and (ii) the zonal room model. The following sensor positions are consider in the latter case:

- at the centre of the room at a height of 1.5 m (position A);
- on a wall at a height of 1.5 m, potentially in the trajectory of the jet (position B);
- on a wall at a height of 1.5 m, outside the trajectory of the jet (position C).

The room is 5 m in length, 5 m in width and 2.6 m in height. The duration of the single room tests is just over 1 h. The whole building tests are carried out for a typical day in summer and a typical day in winter so that the control performance can be assessed and the estimated energy consumption compared. In all cases, the resultant temperature at the centre of the room (position A) is used to assess the control performance.

5 Single room tests

All of the inputs to the room model, except the internal heat gains, are kept constant during the tests so that the results can be interpreted more easily. The test conditions are listed in Table 1. The internal heat gains are chosen so that the step change is as large as possible without losing control of the room temperature, for all positions of the control sensor.

Prior to the tests, the proportional-plus-integral (PI) controller, which is used for room temperature control, was tuned manually using a simulation based on the 'well-mixed' model

Table 1 Test conditions for the single room tests

Test condition	Cooling mode	Heating mode
Zone set-point temperature (°C)	24	21
Temperature adjacent rooms (°C)	24	21
Supply air temperature (°C)	12	18
External air temperature (°C)	35	−5
Short-wave radiation (W/m^2)	0	0
Internal gains before the step change (W)	350	600
Internal gains after the step change (W)	850	50

of room (see Section 7). Different values of the controller parameters were used for the heating and the cooling mode tests. The simulated control sensor measures the average of the air temperature and the mean radiant temperature at the sensor position, and has a time constant of 5 min.

5.1 Cooling tests

Figures 3 and 4 show the time variation of the resultant temperature at the centre of the room, and the corresponding inlet airflow rate, during the slot diffuser and round diffuser cooling mode tests, respectively.

With the slot diffuser (see Figure 3), the resultant temperature predicted by the 'well-mixed' model is very similar to that predicted by the zonal model when the control sensor is in position A or C. However the 'well-mixed' model is unable to predict the fabric heat gains correctly and the airflow rate into the room is much smaller, though it is similar to that predicted by the zonal model when the control sensor is in position B. A significant difference is observed in the steady state temperature predicted by the zonal model when the sensor is placed in the jet at position B. This effect is due to the influence of the cold air in the jet on the control sensor. The size of this temperature difference will depend on the air temperature at the outlet of the diffuser. The test results obtained from the zonal model, when the sensor is located in positions A and C, are quite similar.

As can be seen in Figure 4, with the round diffuser all of the test results are very similar. The higher air entrainment with this type of diffuser leads to greater airflows between the sub-volumes, which creates more homogeneous conditions throughout the room. The resultant temperature and inlet airflow rate variations predicted by the 'well-mixed' model are therefore similar to those predicted by the zonal model. The location of the control sensor is unimportant.

There is very little difference in the shape of the transient responses observed during the tests with either diffuser.

The use of a 'well-mixed' room model is therefore acceptable when there are high airflow rates throughout the room. When the flow rates are lower, the imperfect mixing increases the error in predicting the airflow rate needed to cool the room, which could cause the predicted energy consumption to be underestimated significantly.

5.2 Heating tests

Figures 5 and 6 show the time variation of the resultant temperature at the centre of the room, and the water flow rate in the reheating coil, during the slot diffuser and round diffuser heating mode tests, respectively.

In the heating case, the resultant temperature and the water flow rate predicted by the 'well-mixed' model are very different to those predicted by the zonal model. The differences are larger in the case of the round diffuser than in the case of the slot diffuser. The difference between the results obtained when the control sensor is placed in the jet trajectory (position B) and when it is placed at position A, is also larger

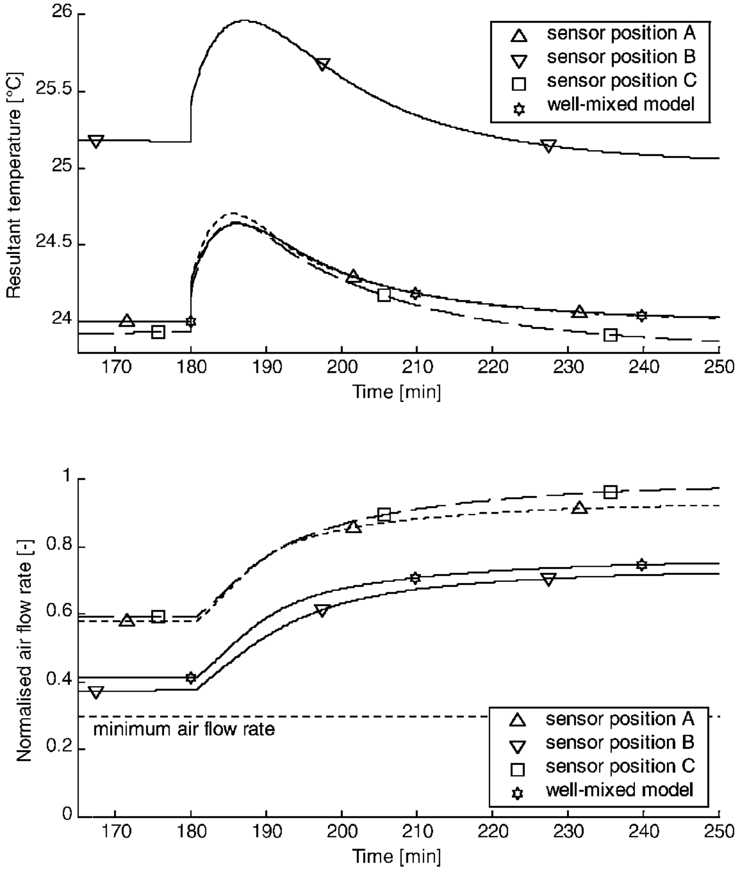

Figure 3 Results of a cooling test in a single room with slot diffuser

for the round diffuser than for the slot diffuser. The most likely explanation is that there is greater mixing of the jet and room air, and thus lower buoyancy forces associated with a negatively buoyant wall jet. It should be noted that the effects of high and low entrainment of air into the jet are opposite to those observed for the cooling case. The results obtained when the sensor is placed in the boundary layer (at position C) are similar to those obtained when the sensor is located at the centre of the room. This is due to the relatively low temperature gradients at the internal walls. Higher gradients might have created larger differences but they were not considered in this study.

The 'well-mixed' model predicts very different values for the air temperature and water flow rate, with either diffuser. The use of this type of model will therefore have a significant effect on the results of any performance test.

6 Whole building performance tests

A simulation of a five-storey building is used to examine the impact of the type of room model and the control sensor position on the overall performance of a building control system under more realistic test conditions.

The simulated building[15] has a north–south orientation and is divided into groups of rooms, here called zones. There are nine rooms on the north and nine rooms on the south side of each floor. All of the rooms have similar dimensions to those of the room used in the single-room

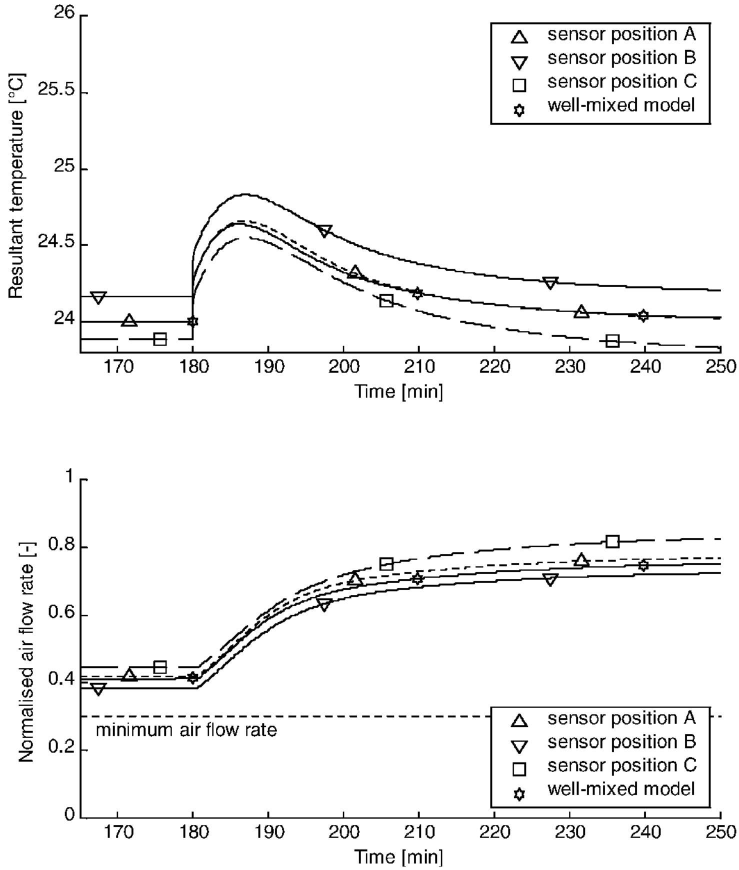

Figure 4 Results of a cooling test in a single room with round diffuser

tests. The zones with a south facade are zones 1, 3, 5 and the zones with a north facade are zones 2, 4, 6. Zones 3 and 4 include rooms with similar occupancy patterns on three of the five floors. There is no glazing on the east and west facades.

The building is equipped with a VAV system, including primary plant (boiler, chiller, etc), a single air-handling unit (including supply and extract fans as well as a mixing box) and pressure-independent terminal boxes with reheat coils in each room. The building also has an innovative lighting and shading control system.[14]

The behaviour of all of the rooms in the same zone will be very similar and, to simplify the simulation, only one room in each zone of the building is simulated. The design heat gains and airflow rates associated with each zone are therefore divided by the number of rooms in that zone and only six rooms are simulated: each representing a typical room in each of the six zones of the building. Real weather data are used to simulate summer and winter conditions. The internal gains follow a predefined profile during an occupancy period, which extends from 8 am to 6 pm.

6.1 Impact of room model and sensor position on the performance assessment

The control performance tests are carried out for one typical day in summer and one typical day in winter. Representative results are presented for zone 3, a zone with a southerly orientation.

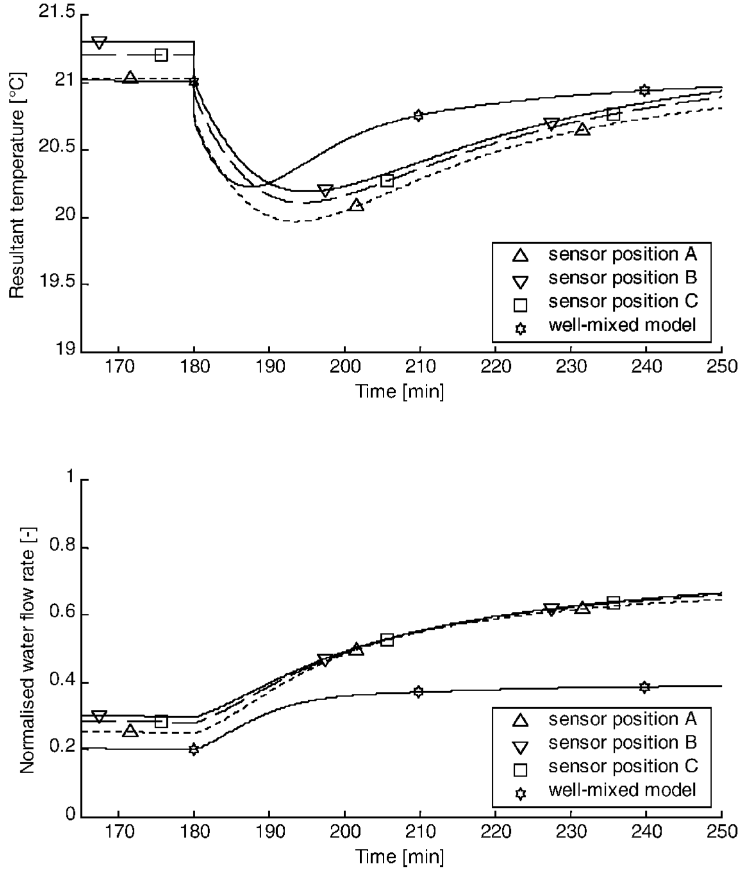

Figure 5 Results of a heating test in a single room with slot diffuser

During the summer test period, the greatest differences are observed when the slot diffuser is used (Figure 7). The cold jet has a significant influence on the measured temperature and the behaviour of the simulated zone, when the sensor is at position B. The results are similar for the other sensor positions. In all cases, there are significant differences between the predicted airflow rates.

The values of the performance indices, which were obtained during the summer test period, are given in Table 2. The mean resultant temperature ϑ_{res}, the maximum peak-to-peak value of resultant temperatures $\Delta\vartheta_{res}$, the mean and maximum values of the PPD, and the root mean square (RMS) and maximum control error, are tabulated for both types of diffuser. The means and maxima are taken both over the occupancy period and over the six zones.

The values obtained for the slot and round diffusers are similar except for those cases where the sensor is placed at position B. The differences in the PPD values obtained with the two diffuser types and for the different sensor positions are not very large. The performance predicted by the 'well-mixed' model, for both the slot and the round diffusers, is similar to that predicted by the zonal model, when the control sensor is placed at any of the three positions.

Significant differences are observed when the tests are performed in winter, as can be seen in Figure 8. Firstly, the 'well-mixed' model predicts a much larger rise in the resultant temperature after the boiler turns on at 6.30 am than is

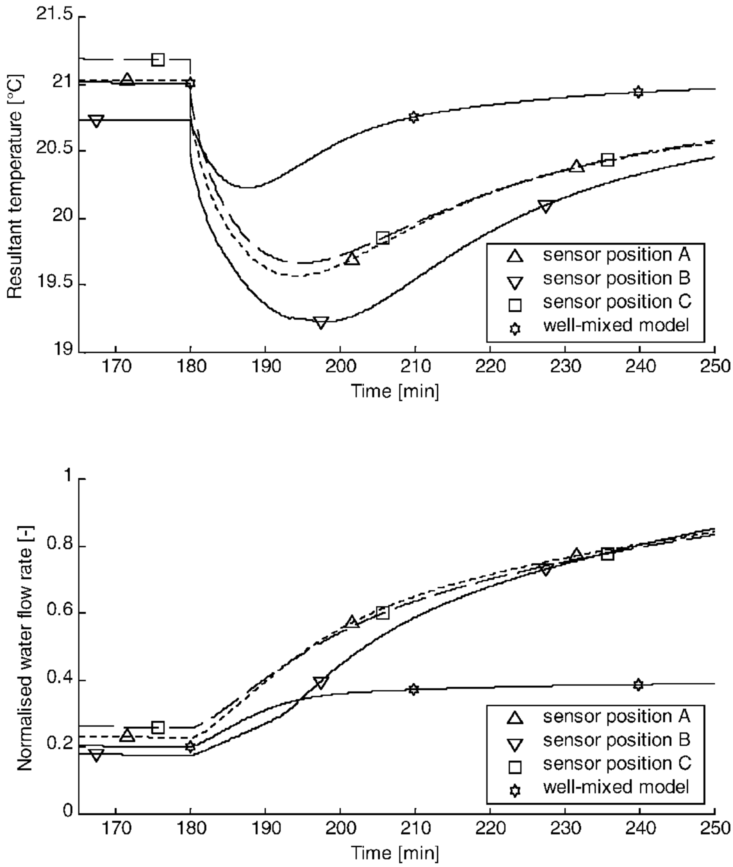

Figure 6 Results of a heating test in a single room with round diffuser

predicted by the zonal model (for all positions of the control sensor). This is a result of the significant temperature stratification, which occurs in the room when the terminal box is in heating mode. The stratification is correctly predicted by the zonal room model but not by the 'well-mixed' room model. The sudden increase in the internal gains, which occur at the start of the occupancy period, is the cause of the second rapid change in the resultant temperature predicted by the zonal model. The temperature stratification is also the reason why the zonal model predicts that the valve of the reheating coil will not begin to close until nearly 2 h later than is predicted by the 'well-mixed' model. This delay results in the zonal model predicting a larger overshoot in the resultant temperature during the early part of the occupancy period. The zonal model also predicts a slightly lower value of the resultant temperature in the afternoon, when the control sensor is at position B. The most likely explanation is that the warm, negatively buoyant wall jet does not begin to increase the temperature measured by the sensor until after midday.

The values of the performance indices, which were obtained during the winter test period, are given in Table 3. There are some differences in the assessment of the control performance when the 'well-mixed' model is used or the control sensor is in position B. However, the differences are again relatively small, particularly those associated with the sensor being located at position B.

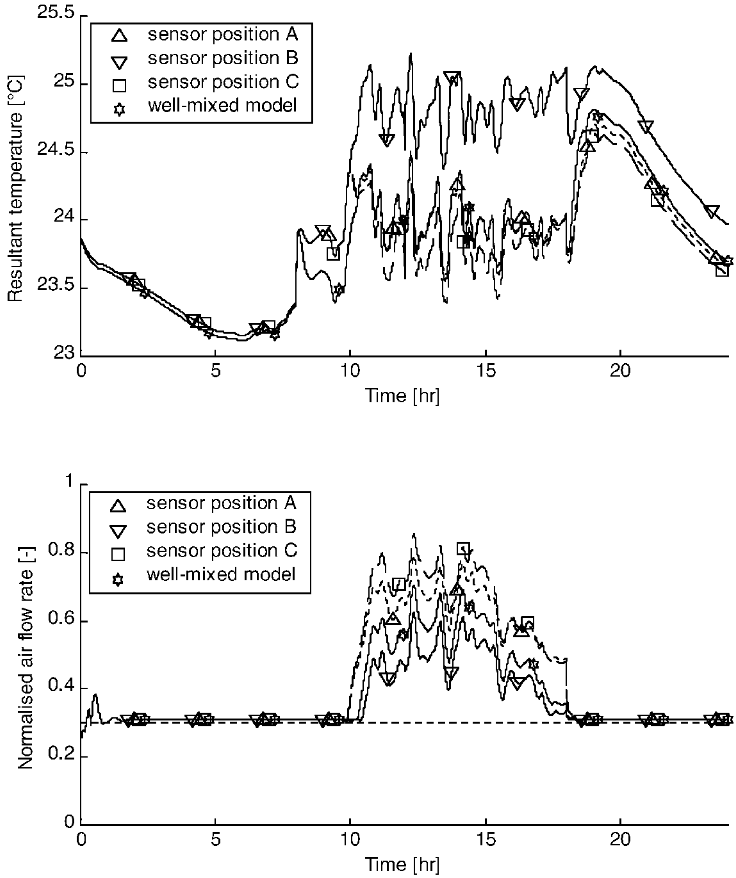

Figure 7 Results of the building test in summer with slot diffusers

Table 2 Control performance indices (summer test period)

	Mean ϑ_{res} °C	Max $\Delta\vartheta_{res}$ °C	Mean PPD %	Max PPD %	RMS E °C	Max E °C
Slot diffuser						
Sensor position A	23.96	1.01	5.16	5.88	0.17	0.63
Sensor position B	24.61	1.73	5.59	6.52	0.75	1.11
Sensor position C	23.83	0.79	5.35	6.43	0.22	0.63
'Well-mixed' model	23.91	1.12	5.31	6.38	0.27	0.71
Round diffuser						
Sensor position A	23.91	1.03	5.27	6.14	0.22	0.65
Sensor position B	24.02	1.18	5.18	6.14	0.27	0.65
Sensor position C	23.75	0.82	5.53	6.71	0.32	0.69
'Well-mixed' model	23.91	1.12	5.31	6.38	0.27	0.71

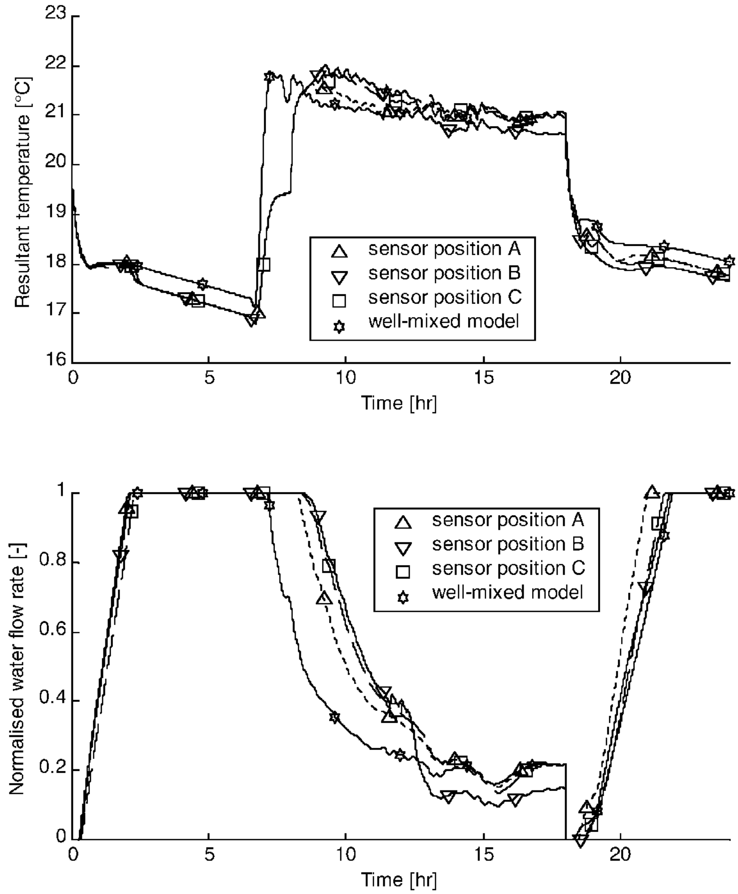

Figure 8 Results of the building test in winter with round diffusers

Table 3 Control performance indices (winter test period)

	Mean ϑ_{res} °C	Max $\Delta\vartheta_{res}$ °C	Mean PPD %	Max PPD %	RMS E °C	Max E °C
Slot diffuser						
Sensor position A	21.14	1.24	5.10	5.87	0.25	0.75
Sensor position B	21.35	1.47	5.27	6.27	0.44	0.96
Sensor position C	21.28	1.40	5.21	6.18	0.37	0.92
'Well-mixed' model	21.09	0.96	5.12	6.63	0.20	0.88
Round diffuser						
Sensor position A	21.15	2.20	5.13	7.17	0.27	1.48
Sensor position B	21.13	2.49	5.27	7.17	0.44	1.48
Sensor position C	21.30	2.42	5.27	7.18	0.41	1.48
'Well-mixed' model	21.09	0.96	5.12	6.63	0.20	0.88

Table 4 Energy consumption—summer test day

	Slot diffuser	Round diffuser
Sensor position A (kWh)	538.6	509.2
Sensor position B (kWh)	494.2	500.8
Sensor position C (kWh)	549.4	523.1
'Well-mixed' model (kWh)	526.7	526.7

6.2 Impact of room model and sensor position on building energy consumption

A comparison is made of the total energy consumption of the building predicted by the zonal models of the rooms, with different sensor positions, and by the 'well-mixed' room model. The results are obtained for both types of diffuser, for one day in summer and one day in winter. Table 4 shows the results obtained during the summer test period.

The energy consumption predicted by the 'well-mixed' room model is close to the mean of the values predicted for the slot diffuser and, for the round diffuser, it is similar to the value obtained using the zonal room model with the sensor placed in a boundary layer outside of the jet (position C). Not surprisingly, the lowest energy consumption is predicted by the zonal model with the sensor placed at position B, where the cold jet influences its measurement. The predicted energy consumption is greater when the control sensor is placed at position C because the sensor measures a temperature that is higher than the temperature at the centre of the room. This effect will depend on the conditions in the adjacent zones.

Table 5 presents the results obtained during the winter test period. In heating mode, the energy consumption predicted by the 'well-mixed' room model is significantly lower than those predicted by the zonal model. The main cause of the differences is the 'well-mixed' model's assumption of homogeneous conditions throughout the zone, which has a major impact on the calculation of the heat losses at the surfaces and the ventilation heat losses. The convective heat exchange at the internal surfaces of the room is sensitive to the unmodelled spatial variations in the air temperature. The 'bypass' effect, which is especially important in the heating case, is also not taken into account by the 'well-mixed' model.

Table 5 Energy consumption—winter test day

	Slot diffuser	Round diffuser
Sensor position A (kWh)	571.7	567.2
Sensor position B (kWh)	629.4	579.7
Sensor position C (kWh)	610.6	606.1
'Well-mixed' model (kWh)	531.1	531.1

7 Impact of sensor position and room model on the tuning of controllers

Manufacturers often supply VAV room temperature controllers with pre-defined values of the control parameters to avoid the risk of oscillatory or unstable operation. Clearly, the choice of room models or sensor positions can have no influence on the tuning of the controllers in such cases. However, in situations where simulation is used to tune the parameters of the controllers, the type of room model and the assumed control sensor position can have a significant influence on the outcome of the tuning process. The impact on the resulting control performance will be particularly important if the controllers are tuned to give tight control or the zone has a relatively small dominant time constant or low room ventilation rates are to be used. Aggressive tuning might even result in an oscillatory control loop when the VAV system is in heating mode and there is significant temperature stratification in the room.

Figure 9 shows an example of this phenomenon in the case of the single room heating mode test using a round ceiling diffuser. The test conditions are the same as those given in Table 1 for the heating mode tests. The controller was tuned manually using the simulation based on the zonal room model and the control sensor at position B. The tuning process resulted in the selection of a value of 0.55 for the proportional

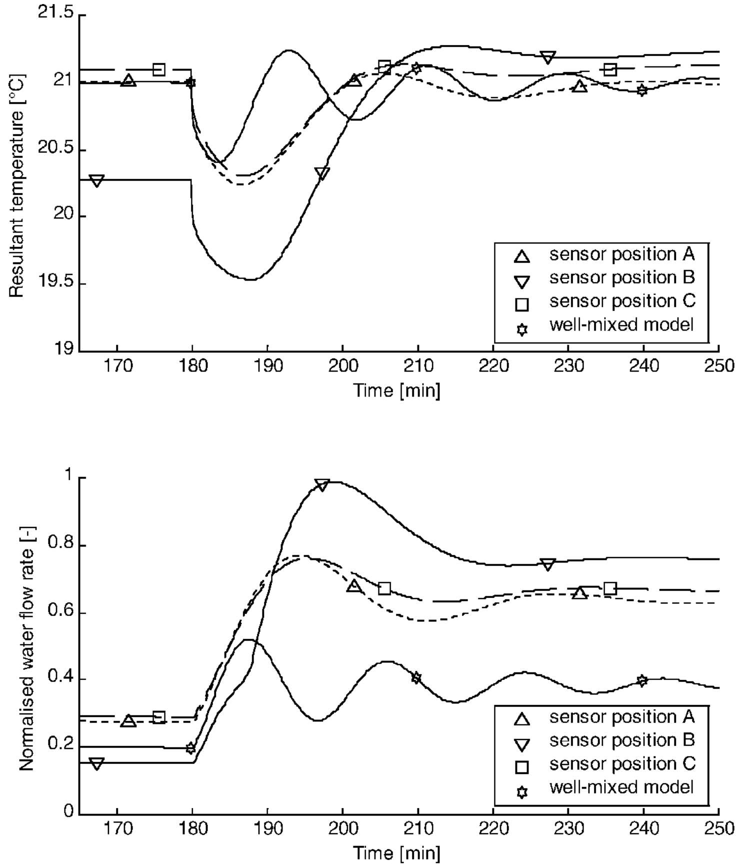

Figure 9 Results of a heating test in a single room after tuning of the controller using the zonal model with the sensor at position B

gain of the controller. Three effects are observed when the controller uses this value of gain:

- The response predicted by the well mixed model is highly oscillatory and the controller takes much longer to stabilize. As might be expected, manual tuning using the 'well-mixed' model produces in a much smaller value of proportional gain (0.13).
- The responses predicted by the zonal model are reasonably similar when the sensor is at the centre of the room or is in the boundary layer (positions A and C).
- The resultant temperature predicted by the zonal model with the control sensor in position B is below its set-point (21°C) before the step decrease in the internal gains occurs but is around the set-point afterwards. This effect is a result of the room temperature controller increasing the temperature of the air entering the diffuser, in response to lower internal gains, and the jet no longer reaching the control sensor.

8 Conclusions

The results of performance assessment based on computer simulation depends on the type of room model, the type of diffuser and the performance indices, which are used.

The accuracy of the predicted performance, based on a 'well-mixed' model of the room, is acceptable when the VAV system is in cooling mode. However, significant errors can occur when the VAV system is in heating mode. The values of the performance indices obtained during the heating tests do not correspond with those obtained using the zonal model, even when the control sensor is placed at the centre of the room.

The position of the control sensor can also have an influence on the predicted behaviour when the zonal model is used. The results have demonstrated that there are only slight differences in the predicted behaviour if the control sensor is placed at the centre of the room or outside of the possible trajectory of any air jets. If the sensor is placed in the trajectory of a jet, a steady-state difference is observed in cooling mode and differences in both the steady-state and transient behaviour are observed in heating mode.

Similar conclusions can be made as far as the predicted energy consumption is concerned. Large differences are observed in the heating case while smaller differences are observed in the cooling case. The use of a 'well-mixed' room model to predict the energy consumption is probably satisfactory in the cooling case.

The effect of the diffuser type on the predicted behaviour also depends on whether the VAV system is in heating or cooling mode. The two diffusers used in the study are characterized either by high or by low entrainment of room air into their jets. There is little temperature difference between the air in the jet and the room air with high entrainment (the round diffuser) and a large temperature difference with low entrainment (the slot diffuser). In cooling mode, the cold jet accelerates when it falls down the internal walls and the higher the temperature difference between the jet and the room air, the more effect the jet has on the control sensor. In heating mode, the lower the temperature difference between the air in the jet and the room air, and the higher the jet velocity, the more likely it is that the warm jet will have reached the control sensor, in spite of the negative buoyancy force. The diffusers were designed and sized according to the ASHRAE recommendations in this study. If the diffusers had not been designed and sized correctly, the room conditions would have been less homogeneous and the use of a zonal model of the room would have been more important, especially for the assessment of comfort in the occupied zone.

In general it can be concluded that the more homogeneous are the actual room conditions, the more accurate will be the results obtained using a 'well-mixed' room model. The use of a zonal room model is therefore recommended when there is likely to be significant temperature stratification, or if the control sensor is located in the trajectory of an air jet with a temperature that is very different from the mean room air temperature, or where the terminal boxes and diffusers are poorly designed.

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