Thermal Comfort, Uniformity, and Ventilation Effectiveness in Patient Rooms: Performance Assessment Using Ventilation Indices

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ABSTRACT

This paper describes a study of the performance of a ventilation system in a typical patient room using CFD modeling and calculations of various ventilation indices. The results show that the use of basboard heating is necessary in extreme weather conditions. In particular, good occupant conditions are virtually impossible to achieve without baseboard heating. Further, in general weather conditions, a ventilation rate of 4 ACH provides adequate conditions, though an increase to 5 or 6 ACH is optimum.

INTRODUCTION

The provision of good thermal comfort for patients in hospital rooms is an obvious necessity because of the length of time that the patient is resident in the room. Current AIA guidelines indicate that the minimum air change rates (ACH) to be used in these rooms should be 2 ACH (AIA 1996-97). However, a recent study has indicated that this minimum is too low in extreme winter conditions, especially where no baseboard heating is included.

This paper describes a study undertaken to assess the performance of the ventilation system as applied to a typical patient room using the technique of airflow modeling coupled with the calculation of various ventilation indices. The airflow modeling in this study uses the technique of computational fluid dynamics (CFD).

The paper describes the physical room considered in this study, a brief overview of airflow modeling, as well as a description of the various diffuser validation exercises undertaken to ensure that the representations of diffusers in the CFD models are accurate compared to available manufacturers' data. There is then an introduction to a number of different ventilation indices that determine the performance of the ventilation system in terms of thermal comfort, uniformity, and ventilation effectiveness. The ventilation indices are then applied to different ventilation system cases to assess the most appropriate ventilation system to provide good indoor air quality.

The results of this study are also intended to be linked to a concurrent study into minimizing the risk from airborne organisms in hospital isolation rooms. While the isolation room is not exactly the same in terms of dimensions, the two studies share enough common features—for example, there is a single bed in the room, the glazing features are similar in each case, there are similar amounts of furniture in the room, etc.—that the conclusions drawn from the isolation room study will be viable in this study, and vice versa.

DESCRIPTION OF PHYSICAL PROBLEM

The patient room considered for this study (Figure 1) measures 12 ft (3.66 m) by 20 ft, 2 in. (6.15 m) with a floor-to-ceiling height of 9 ft, 6 in. (2.9 m). The room has one of the shorter walls on the perimeter of the building with an extensive glazed area covering about two-thirds of that wall. The glazing is of high quality with a heat transfer coefficient of 0.29 Btu/ft^2·h·°F (1.65 W/m^2·K). Only relatively small levels of heat loss/gain are associated with the floor and glazing wall fabric; the other walls and ceiling are assumed adiabatic.

The room contains a bed and other typical furniture. For the purpose of assessment, the key areas for assessing the indoor air quality are around the bed and the couch, and the ventilation indices are calculated between floor level and 6 ft (1.83 m) above floor level. Figure 1 shows the room configured with the patient lying on the bed and a visitor seated on the couch. The supply air temperature is controlled to give an exhaust temperature of 73.4°F (23°C).

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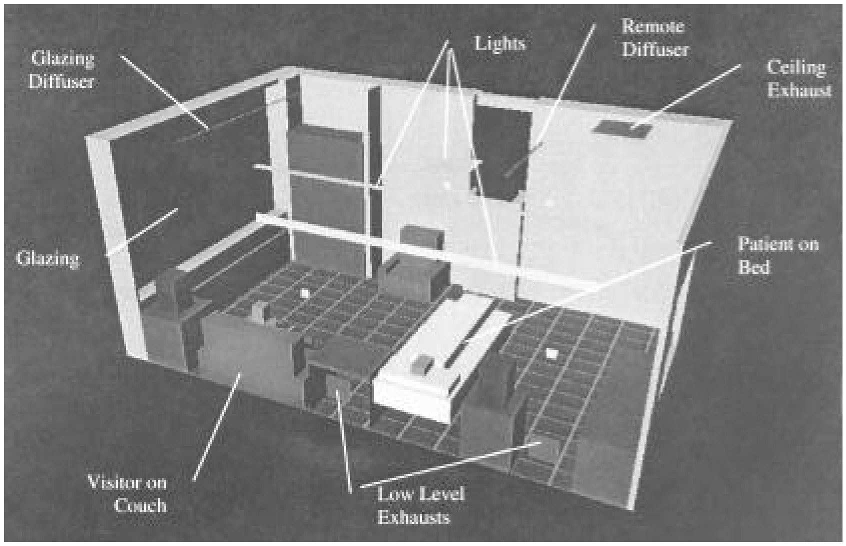

Figure 1 *Patient room configuration.*

UNDERSTANDING THE AIRFLOW AND HEAT TRANSFER

Airflow modeling based on computational fluid dynamics (CFD), which solves the fundamental conservation equations for mass, momentum, and energy in the form of the Navier-Stokes equations is now well established:

$$\frac{\partial}{\partial t}(\rho\varphi) + div(\rho\vec{V}\varphi - \Gamma_{\varphi} grad\varphi) = S_{\varphi}$$

Transient + Convection – Diffusion = Source

where

ρ = density,

$\vec{V}$ = velocity vector,

φ = dependent variable,

Γ_{φ} = exchange coefficient (laminar + turbulent),

S_{φ} = source or sink.

How Is It Done?

Airflow modeling solves the set of Navier Stokes equations by superimposing a grid of many tens or even hundreds of thousands of cells that describe the physical geometry, heat, contamination sources, and the air itself. Figures 2 and 3 show a typical research laboratory and the corresponding space discretization, subdividing the laboratory into tens or hundreds of thousands of cells.

The simultaneous equations thus formed are solved iteratively for each one of these cells to produce a solution that satisfies the conservation laws for mass, momentum, and energy. As a result, the flow can then be traced in any part of the room, simultaneously coloring the air according to another parameter such as temperature.

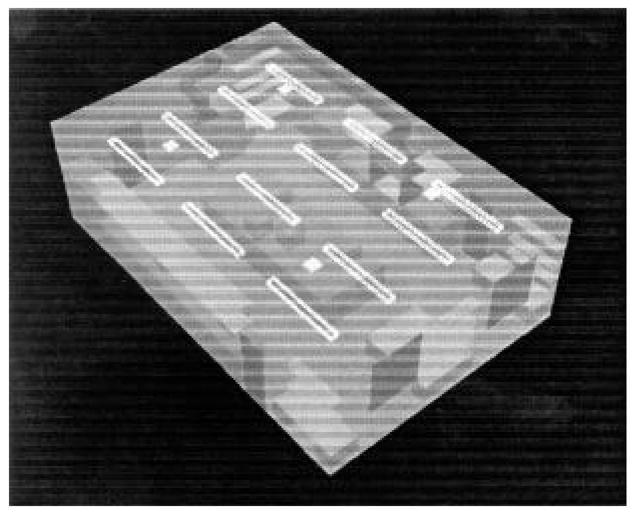

Figure 2 *Geometric model of a laboratory.*

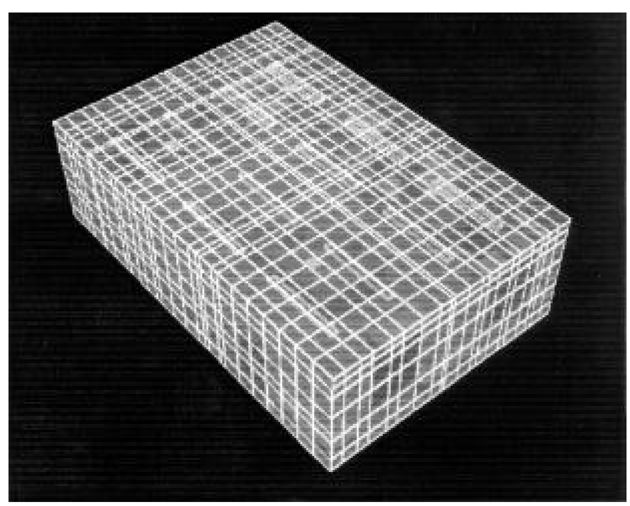

Figure 3 *Superimposed grid of cells for calculation.*

Validation of Airflow Modeling Methodology

The methodology was used extensively in a previous publication by Memarzadeh (1998), which considered ventilation design of animal research facilities using static microisolators. In order to analyze the ventilation performance of different settings, numerical methods based on computational fluid dynamics were used to create computer simulations of more than 160 different room configurations. The performance of this approach was successfully verified by comparison with an extensive set of experimental measurements. A total of 12.9 million experimental data values were collected to confirm the methodology. The average error between experimental and computational values was 14.36% for temperature and velocities, while the equivalent value for concentrations was 14.50%.

To further this research, several meetings were held to solicit project input and feedback from the participants. There were more than 55 international experts in all facets of the animal care and use community, including scientists, veterinarians, engineers, animal facility managers, and cage and rack manufacturers. The pre-publication project report underwent peer review by a ten-member panel from the participant group, selected for their expertise in pertinent areas. Their comments were adopted and incorporated in the final report.

The publication was reviewed by a technical committee of the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) and data accepted for inclusion in their 1999 Handbook.

VALIDATION OF THE SUPPLY AIR DIFFUSER MODELS

Three diffuser types are considered in this study:

- A multiple linear slot diffuser.
- A low induction diffuser.
- A radial diffuser.

Test data from manufacturers provide throw and drop for the diffusers at a given flow rate. The test room configurations used by the manufacturers were modeled to provide a prediction of throw and drop and the jet characteristics (such as jet velocity and thickness) adjusted to achieve good agreement with the manufacturers' data. The flow rates chosen for the tests were representative of the values used in this study.

Figures 4-6 show a typical linear slot diffuser and the validation plots for such a diffuser, operating in both one-way and two-way mode. The vertical line represents the throw as given by manufacturers' data that can be compared with the isovel for the respective terminal velocity.

Figures 7 and 8 display a typical low induction diffuser, as well as the validation plot for the diffuser. The horizontal line represents the throw as given by manufacturers' data, while the contour represents the numerical equivalent line.

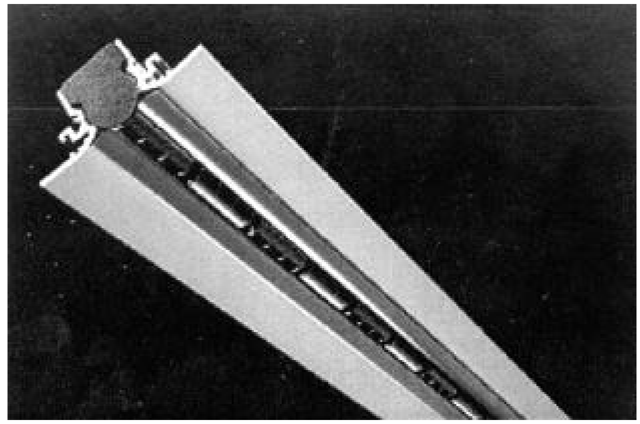

Figure 4 *Linear slot diffuser.*

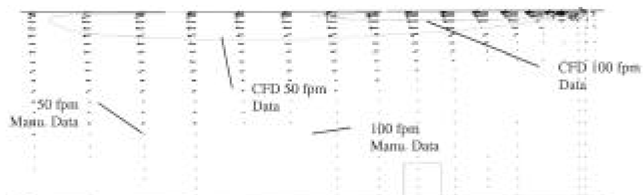

Figure 5 *Comparison of CFD and manufacturers' data for linear slot diffusers operating in one-way mode (zoomed in).*

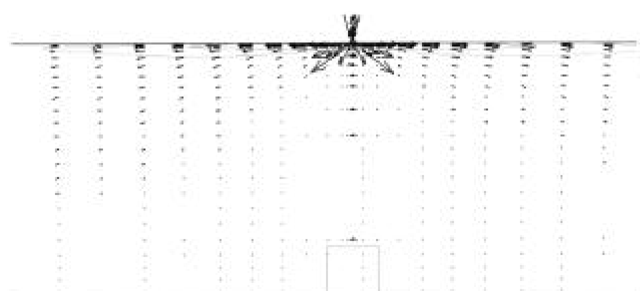

Figure 6 *Comparison of CFD and manufacturers' data for linear slot diffusers operating in two-way mode (zoomed in).*

Finally, Figures 9 and 10 show the radial diffuser considered, as well as the validation plot for the radial diffuser. The three horizontal and vertical lines represent the 100, 75, and 50 fpm manufacturers' throw data, respectively. Other than the match at the horizontal 100 fpm line, the agreement is very good between the numerical and manufacturers' data.

OVERVIEW OF VENTILATION INDICES

This section describes several ventilation indices that can be used to evaluate the performance of the ventilation system in the patient room. The indices give good indications as to the level of thermal comfort, uniformity, and ventilation effectiveness for the room. The calculation of the indices allows for

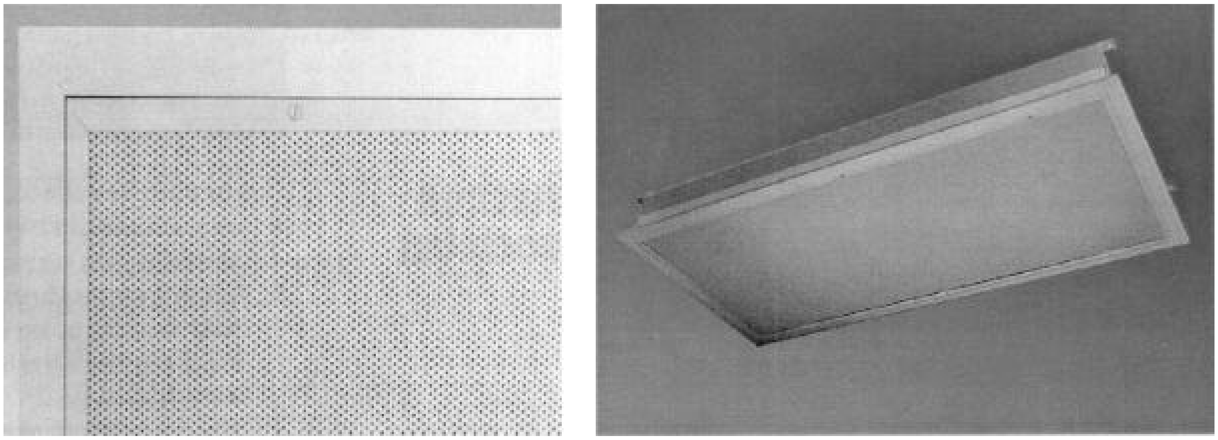

Figure 7 *Low induction diffusers.*

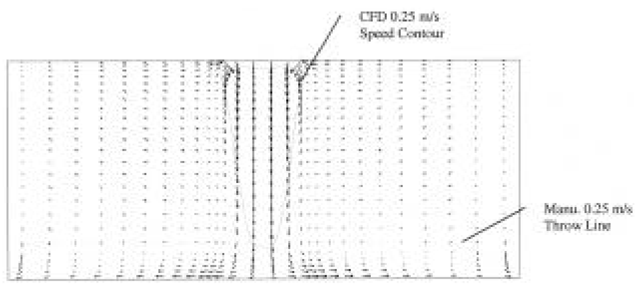

Figure 8 *Comparison of CFD and manufacturers' data for low induction diffuser.*

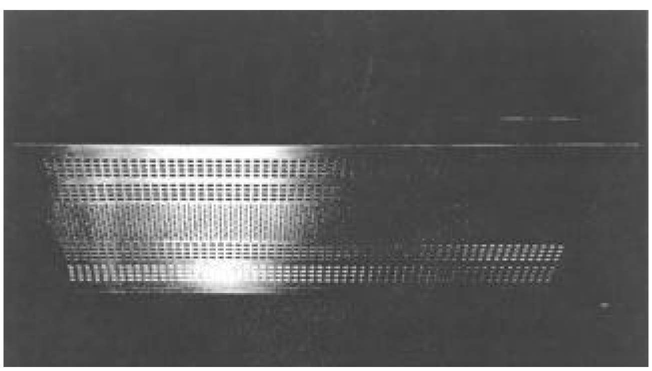

Figure 9 *Radial diffuser.*

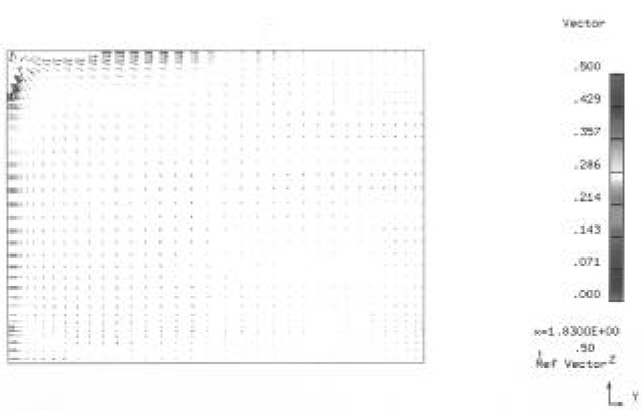

Figure 10 *Comparison of CFD and manufacturers' data for radial diffuser operating at* $\Delta T = 5°F$.

a more consistent means of assessing these parameters than through manual interpretation of the results.

Thermal Comfort

One method of assessing the thermal comfort is to use the equations for predicted mean vote (PMV) and predicted percentage dissatisfied (PPD) produced by Fanger and given in the ASHRAE Handbook (ASHRAE 1997). These equations are based on an empirical investigation of how people react to differing environments. It is well known that different people will have a different perception of the climate produced in a building, and that any given climate is unlikely to be considered satisfactory by all. In fact, it is considered that satisfying 80% of occupants is good, so a PPD of less that 20% is good. PMV and PPD provide a measure of the likely response of people. The predicted mean vote is an index from –3 (representing a response of very cold) through 0 (representing a thermally neutral response) to +3 (representing a response of very hot). The predicted percentage dissatisfied is directly related to the predicted mean vote, and so some people suggest that one is redundant. However, from an engineering stance, it is useful to have both immediately available. While PPD provides the information as to whether the environment is likely to be acceptable, PMV tells us what the problem is—whether it is too hot or too cold when the number dissatisfied is too large.

The equations implemented in the analysis shown here are taken from Fanger's equations for PMV and PPD as given in BS EN ISO 7730: 1995.

Definitions

$$\begin{aligned}PMV &= (0.303e^{-0.036M} + 0.028)\{(M - W) - 3.05 \times 10^{-3}\\ &\quad [5733 - 6.99\,(M - W) - p_a] - 0.42\,[(M - W) - 58.15]\\ &\quad 1.7 \times 10^{-5} M\,(5867 - p_a) - 0.0014\,M\,(34 - t_a) - 3.96\\ &\quad \times 10^{-8} f_{cl}\,[(t_{cl} + 273)^4 - (t_r + 273)^4] + f_{cl} h_c\,(t_{cl} \pm t_a)\}\end{aligned}$$

where

$$\begin{aligned}t_{cl} &= 35.7 - 0.028\,(M - W) - I_{cl}\,\{(3.96 \times 10^{-8} f_{cl}\\ &\quad [(t_{cl} + 273)^4 - (t_r + 273)^4] + f_{cl} h_c\,(t_{cl} - t_a)\}\end{aligned}$$

$$h_c = 2.38\,(t_{cl} - t_a)^{0.25} \text{ or } h_c = 12.1v^{0.5},$$
whichever is greater

f_{cl} $= 1.00 + 1.29\, I_{cl}$ for $I_{cl} \leq 0.078$ m^2 kW^{-1}
or $f_{cl} = 1.05 + 0.645\, I_{cl}$ for $I_{cl} > 0.078$ m^2 kW^{-1}

PPD $= 100 - 95e^{-n}$

where

n $= 0.03353$ PMV^4 + 0.2179 PMV^2

List of Symbols

PMV = predicted mean vote
PPD = predicted percentage dissatisfied
M = metabolic rate (W/m^2 of the body area)
W = external work (W/m^2 of the body area, = 0 in most cases)
I_{cl} = thermal resistance of clothing (m^2·kW^{-1})
f_{cl} = ratio of clothed surface area to nude surface area
t_a = air temperature (°C)
t_r = mean radiant temperature (°C)
v = air velocity relative to the body (m/s^{-1})
p_a = partial water vapor pressure (Pa)
h_c = convective heat transfer coefficient (W·m^2·K)
t_{cl} = clothing surface temperature (°C)

The number of parameters used to produce these measures begins to show how complex the human response is to the environment. PMV and PPD include air temperature, mean radiant temperature, air velocity, vapor pressure, clothing level, metabolic rate, and external work rate.

Within the remit of airflow modeling, the values of PMV and PPD can be calculated for each cell in turn and so a volume weighted average can be produced for the entire space. Clearly the averaging process runs the risk of hiding local areas of poor PMV and PPD. These can be investigated by inspecting statistical information, such as the minimum and maximum values and standard deviation. Also, two distinctly different conditions may produce acceptable PMV and PPD, such as low air temperature with low air speed or high air temperature with high air speed. In such a case, an occupant moving from one to the other would almost certainly register discomfort as a result of the change. A similar response could also be perceived due to temperature stratification where the body experiences too high a variation in conditions. To consider such variations, a different method can be used that is designed to measure uniformity.

Uniformity

A long-standing measure of uniformity is the Air Diffusion Performance Index (ADPI). ADPI (ASHRAE 1997) is a parameter that measures the uniformity of the space in terms of the proportion of the volume with velocity lower than 0.35 m/s (70 ft/min) and draft temperature between −1.7°C (3°F) and +1.1°C (2°F) from the mean temperature.

The draft temperature is defined as follows:

$$T_d = T_p - T_m - (7.66 \cdot (v_p - 0.15))$$

where

T_p = temperature at a point (each cell) in °C,
T_m = mean air temperature in °C,
v_p = velocity at the point in m/s.

For analysis using airflow modeling, it is simple to apply this methodology to each of the calculation cells and then produce a volume weighted total for the proportion of the occupied zone that passes this test. Although the test explicitly fails high velocities above 0.35 m/s (70 ft/min) and large temperature variations, the test also implicitly tests for low velocities. The formula allows for airspeed to offset temperature variations in the calculation of the draft temperature. However, as airspeed falls, the temperature must also fall to compensate for the rise in draft temperature due to low velocity. Still air represents a rise in draft temperature of 1.1°C (2°F) compared with a typical room velocity of 0.15 m/s (30 ft/min), while a rise of velocity to 0.35 m/s (70 ft/min) represents a fall in draft temperature of 1.5°C (2.8°F) from that for a typical room velocity.

Ventilation Factors

Although less well established and more difficult to interpret, some commonly used ventilation factors (Brouns and Waters 1991) are the local mean age of the air and the Local Air Change Index.

The local mean age of the air, $\bar{\tau}_p$, is defined as the average time taken for air to travel from the inlet to any point p in the room and may be written as

$$\bar{\tau}_p = {}_0\!\int^{\infty} t A_p(t)\; dt$$

where $A_p(t)$ represents the age distribution curve for air arriving at point p.

The lower the local mean age, the less likely the air is to feel stale or stuffy.

This leads to a second parameter, the Local Air Change Index (LACI), ε_p, which is the age relative to the supply rate and is defined as follows:

$$\varepsilon_p = \tau_n / \bar{\tau}_p$$

where τ_n is the nominal time constant of the room (the reciprocal of the ventilation air change rate). A value of LACI of unity (1) represents the equivalent of a piston flow with a smaller number representing less effective ventilation. Values greater than unity are possible in the space, for example, in situations where the exhaust is close to the supply, but the mean for the space cannot exceed unity.

Although an actual value is difficult to identify for a pass/fail criterion, these parameters can easily be used to compare the relative performance of different ventilation systems.

DESCRIPTION OF CASES CONSIDERED

The majority of cases (Cases 1 to 29) in this paper consider two extreme design conditions as follows:

1. Maximum summer day solar loading for a south-facing patient room. External ambient is 31.5°C (88.7°F). Heat gains were considered from the solar loading, heat transfer through the glazing and room fabric, lighting, and miscellaneous heat gains from items such as a television and occupancy. Total heat gain = 2470 W.

2. Minimum winter night temperatures. External ambient is –11.7°C (10.9°F). Heat transfer losses were considered through the glazing and room fabric and infiltration from the glazing with heat gain from occupants only. Total heat loss = 580 W.

A smaller number of cases (Cases 30 to 36) were considered at two less extreme, more typical weather conditions. The ventilation systems considered in these cases used lessons learned from the previous extreme weather conditions for their design. The two conditions were as follows:

3. Summer day with external ambient at 17.06°C (62.7°F). As the external temperature is lower than the internal temperature, the heat gains, such as heat transfer through the glazing, now become losses. Heat gains omitted solar loading and miscellaneous heat gains. Total heat gain = 752 W.

4. Winter day with external ambient at 2.7°C (36.8°F). Heat transfer losses were considered through the glazing and room fabric, but loss through infiltration was omitted. Total heat loss = 130 W.

The cases considered in this study are shown in Table 1. The cases were centered predominantly on winter cases because these were likely to have more problems than the summer cases. In particular, the winter cases are more likely to demonstrate poor mixing compared with summer cases. It should be noted that the remote linear diffuser was only defined to throw vertically downward in cases with low ACH. In particular, cases 1-9 and 30-36 all have ACH values of 2 to 6 ACH. For all other cases, the remote linear diffusers were operated in two-way mode. This was to prevent high-momentum jets occurring close to the patient. The use of baseboard heaters was considered for winter cases, but for summer cases, all cooling was specified to occur as a result of the supply diffusers. The baseboard heaters considered were 0.46 m (18 in.) high and 2.35 m (7.7 ft) long. For the most extreme case, the heater dissipated 196 W/m (203 (Btu/h)/ft).

The locations of the diffusers for the different ventilation systems are shown in Figures 11-14. The linear slot diffusers used were 1.22 m (4 ft) long, each with two slots of ½ in. width (1.3e-2 cm). The low induction diffusers were 2 ft × 4 ft, and the radial diffuser was 1 ft × 4 ft.

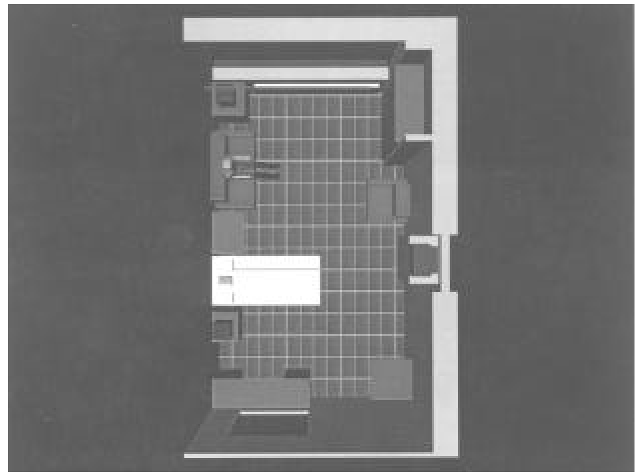

Figure 11 *Plan view showing diffuser locations for linear/linear diffuser combination.*

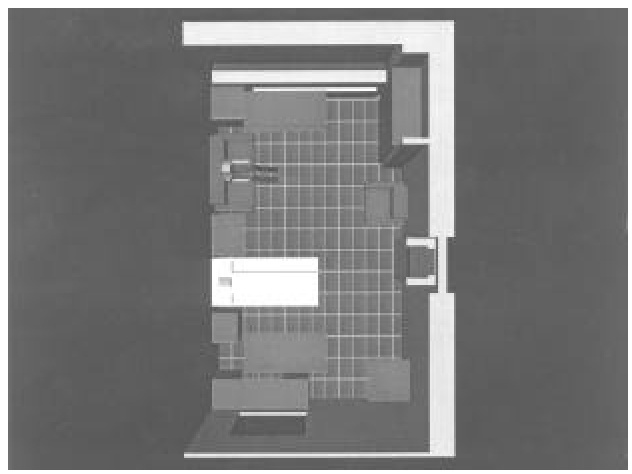

Figure 12 *Plan view showing diffuser locations for low induction/low induction diffuser combination.*

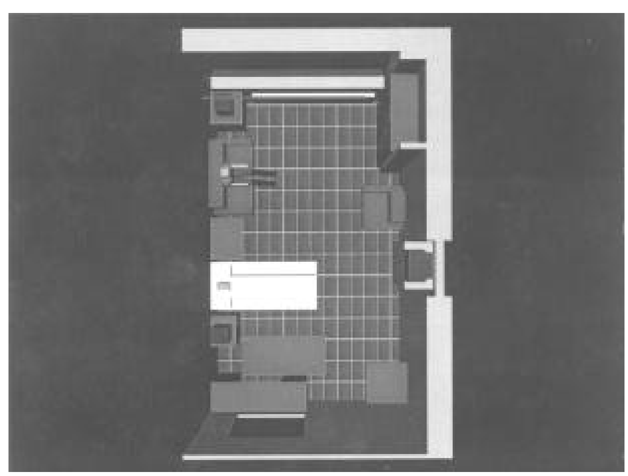

Figure 13 *Plan view showing diffuser locations for linear/low induction diffuser combination.*

TABLE 1
Cases Considered and Results of Index Calculations

Case	Summer (S)/ Winter (W)	Supply Flow Rate (cfm)	Supply Temp. °F (°C)[1]	General Exhaust Flow Rate (cfm)	ACH	Exhaust Location	Supply Diffuser Combination (Glazing/ Remote)	Base Heater (Y/N)	ADPI (%)	PPD (%)	PMV	LMAA (Around Patient)	LACI (Around Patient)
1	W	75	78.3 (25.7)	40	2	Ceiling	Linear/Linear	Y[2]	63.4	5.8	–0.18	28–30	0.93–1.04
2	W	75	"	40	2	Low[3]	Linear/Linear	Y[2]	63.2	5.4	–0.12	26–29	1.0–1.08
3	W	150	85.8 (29.9)	70	4	Ceiling	Linear[4]	N	34.5	11.5	–0.53	20–25	0.52–0.95
4	W	150	75.9 (24.4)	70	4	Ceiling	Linear/Linear	Y[2]	61.0	6.55	–0.26	6–16	0.9–2.5 (4.5 Peak)
5	W	150	85.8 (29.9)	70	4	Low[3]	Linear/Linear[5]	N	44.2	6	–0.18	12–19	0.72–1.12
6	W	150	75.9 (24.4)	70	4	Low[3]	Linear/Linear	Y[2]	67.4	6.1	–0.22	7–13	1.1–1.94 (2.2 Peak)
7	W	225	75.0 (23.9)	145	6	Ceiling	Linear/Linear	Y[2]	76.8	6.8	–0.29	5–11	1.05–1.98 (2.4 Peak)
8	W	225	81.7 (27.6)	145	6	Low[3]	Linear/Linear[5]	N	49.0	6	–0.19	8–11	0.87–1.17
9	W	225	75.0 (23.9)	145	6	Low[3]	Linear/Linear	Y[2]	83.0	6.4	–0.25	7–10	0.93–1.57 (Peak 4.3)
10	W	225	"	145	6	Ceiling	Low Ind./Low Ind.	Y[2]	69.0	6.7	–0.28	8–12	0.81–1.11
11	W	225	81.7 (27.6)	145	6	Low[3]	Linear/Low Ind.[5]	N	47.7	6.1	–0.2	7–13	0.74–1.42
12	W	225	75.0 (23.9)	145	6	Low[3]	Low Ind./Low Ind.	Y[2]	68.7	6.8	–0.29	8–10	0.94–1.28
13	W	225	"	145	6	Ceiling	Radial	Y[2]	90.5	6.8	–0.28	6–11	0.87–1.51
14	W	225	"	145	6	Low[3]	Radial	Y[2]	91.1	6.6	–0.27	7–11	0.88–1.37
15	W	300	79.5 (26.4)	220	8	Ceiling	Linear/ Linear[5]	N	21.3	12.7	–0.56	4–15	0.52–1.76
16	W	300	73.4 (23.7)	220	8	Ceiling	Linear/Linear	Y[2]	75.1	7.8	–0.36	6–9	0.84–1.18
17	W	300	79.5 (26.4)	220	8	Low[3]	Linear/Linear[4]	N	33.9	8.6	–0.38	5–12	0.58–1.41
18	W	300	73.4 (23.7)	220	8	Low[3]	Linear/Linear	Y[2]	74.6	7.4	–0.33	7–9	0.78–1.12
19	W	380	78.3 (25.7)	300	10	Ceiling	Linear/Linear[5]	N	40.6	11.8	–0.55	5–12	0.51–1.26
20	W	380	"	300	10	Ceiling	Linear[4]	N	43.0	10.4	–0.48		
21	W	380	74.3 (23.5)	300	10	Ceiling	Linear/Linear	Y[2]	85.5	7.8	–0.36	4–7	0.83–1.41
22	W	380	78.3 (25.7)	300	10	Low[3]	Linear/Linear[5]	N	57.8	9	–0.42	5–6	0.86–1.19
23	W	380	74.3 (23.5)	300	10	Low[3]	Linear/Linear[5]	Y[2]	86.8	7.5	–0.34	5–7	0.79–1.15

TABLE 1 (Continued)
Cases Considered and Results of Index Calculations

Case	Summer (S)/ Winter (W)	Supply Flow Rate (cfm)	Supply Temp. °F (°C)[1]	General Exhaust Flow Rate (cfm)	ACH	Exhaust Location	Supply Diffuser Combination (Glazing/ Remote)	Base Heater (Y/N)	ADPI (%)	PPD (%)	PMV	LMAA (Around Patient)	LACI (Around Patient)
24	S[6]	380	52 (11.1)	300	10	Ceiling	Linear/Linear[5]	N	89.1	12.6	–0.58	4–5	0.98–1.4
25	S[6]	380	"	300	10	Low[3]	Linear/Linear[5]	N	85.3	9.9	–0.46	5–7	0.81–0.99
26	S[6]	380	"	300	10	Ceiling	Linear/Low Ind.[5]	N	87.8	14.2	–0.65	5–7	0.74–1.14
27	S[6]	380	"	300	10	Low[3]	Linear/Low Ind.[5]	N	88.0	10.3	–0.49	6–7	0.74–0.96
28	S	455	56.2 (13.4)	375	12	Ceiling	Linear/Linear[5]	N	85.4	10.4	–0.48	4–6	0.74–1.26
29	S	455	"	375	12	Low[3]	Linear/Linear[5]	N	85.2	10.2	–0.48	4–6	0.82–1.04
30	W	150	73.9 (23.3)	70	4	Low[3]	Linear/Linear[7]	Y[8]	74.4	6.1	–0.23	9–13	1.06–1.26 (Peak 2.38)
31	W	225	73.8 (23.2)	145	6	Low[3]	Linear/Linear[7]	Y[8]	89.9	6.6	–0.27	8–9	0.95–1.4 (Peak 2.0)
32	W	150	73.9 (23.3)	70	4	Low[3]	Linear/Linear[9]	Y[8]	80.1	6.1	–0.22	13–15	0.9–1.05
33	W	225	73.8 (23.2)	145	6	Low[3]	Linear/Linear[9]	Y[8]	85.2	6.7	–0.28	8–11	0.85–1.19
34	S	150	57.4 (14.1)	70	4	Low[3]	Linear/Linear	N	83.0	8.0	–0.37	10–15	0.86–1.22 (Peak 2.6)
35	S	225	62.8 (17.1)	145	6	Low[3]	Linear/Linear	N	88.7	8.1	–0.38	8–10	0.95–1.35 (Peak 2.86)
36	S	150	57.4 (14.1)	70	4	Low[3]	Linear/Linear[9]	N	89.4	7.8	–0.36	13–14	0.86–0.97

Notes:

[1] Quoted value based on supply air compensating for heat gain/loss. In reality, air supply temperature in model adjusted slightly in some cases to ensure 73.4°F (23°C) at exhaust, particularly cases involving ceiling exhausts with no baseboard heating.

[2] Baseboard heater will dissipate 80% of total heating load; in this case, dissipation will be 460 W (80% of 580W).

[3] Single ceiling-level exhaust split into two 1 ft × 1 ft exhausts located 1 ft from floor level on either side of the bed on the patient-side wall.

[4] Only glazing diffuser considered.

[5] Glazing linear slot diffuser directed toward glazing only to account for heat gain/loss.

[6] Based on the cooling load, this is the first ACH to be checked without using cooling mechanisms other than the supply air (air i s supplied at 11.11°C [52°F]).

[7] Remote diffuser directed vertically downward.

[8] Baseboard heater will dissipate 80% of total heating load; in this case, dissipation will be 105 W (80% of 130W).

[9] Remote diffuser operating in two-way mode.

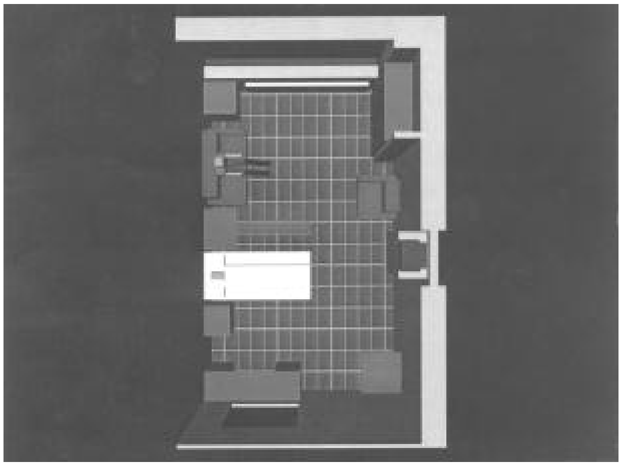

Figure 14 *Plan view showing diffuser locations for radial diffuser system.*

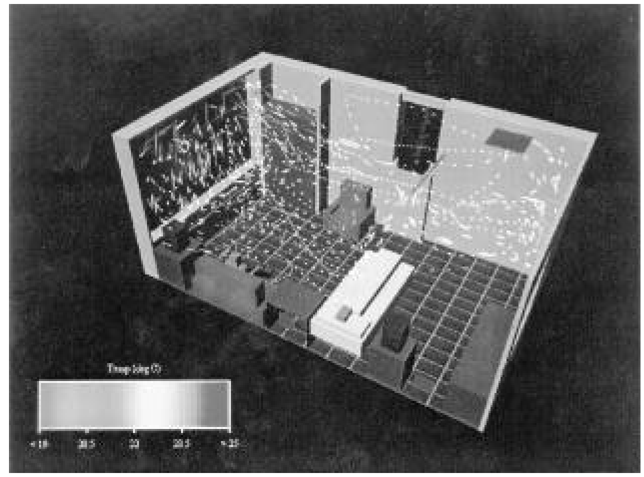

Figure 15 F*low pattern in Case 15 (no baseboard heating).*

RESULTS OF CASES CONSIDERED

The results of the ventilation index calculations are given in Table 1. This section discusses the results in terms of the different parameters considered.

Extreme Winter Cases—
No Baseboard Heating vs. Baseboard Heating

The results show that, in extreme winter cases where no baseboard heating is used, the ventilation system satisfies the Fanger indices, PPD and PMV. In particular, the PPD values for all these cases are below 20%, while the PMV values are all below ±0.5.

However, these cases show poor values for the uniformity index, APDI. In particular, the value of ADPI is below 50% for all cases except for Case 22, but here the ACH is 10. The reason that the values are low is that there is relatively poor mixing when no baseboard heating is used, resulting in the cases failing on draft temperature in the ADPI calculation. For example, Figure 15 shows the flow patterns from Case 15—the difference in temperature between floor level and mid-level of the room is apparent, indicating high stratification of the air.

The value for Case 8 (6 ACH) is higher than for those at higher ACH; compare, for example, Case 17 (8 ACH). The reason for this is that Case 8 uses a remote diffuser that is directed vertically downward, and this helps to increase the value of ADPI. However, the diffuser cannot be directed this way for ever increasing ACH; at some point, the patient would find this jet uncomfortable.

As well as the low values of ADPI, the poor quality of the no- baseboard-heating ventilation system conditions are emphasized on examination of the LACI values for these cases. In particular, again with the exception of Case 8, the values for LACI are low around the patient, even when high values of ACH are used. For these cases, although the values of LMAA around the patient are low, the low LACI values indicate that the air is supply inefficiently; see, for example, Case 20 (10 ACH).

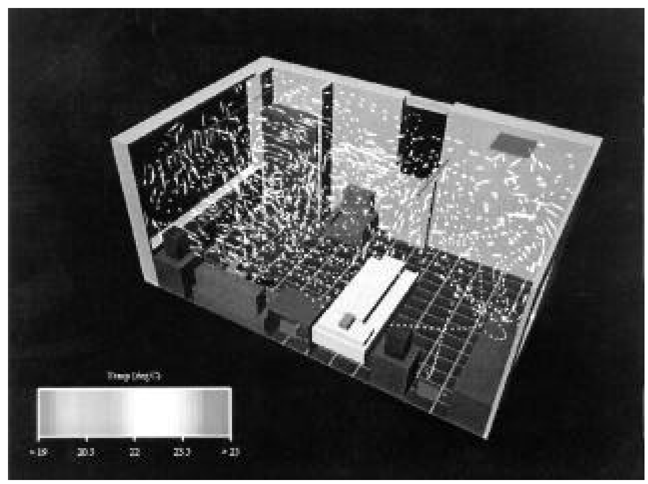

Figure 16 *Flow pattern in Case 16 (baseboard heating).*

The results of the ventilation index calculations are more favorable when baseboard heating is included. The values for the Fanger indices, PMV and PPD, are again very good. In particular, the PPD values for all these cases are below 20%, while the PMV values are all below ±0.5. Further, the value for ADPI does not drop below 60 for these cases, even at the lowest flow rates; see, for example, Cases 1 and 2, which consider 2 ACH. The improvement is attributed to better mixing, and so the systems no longer fail on the draft temperature calculation. For example, Figure 16 displays the flow field for Case 16. The plot indicates much more uniform temperatures and better mixing than Figure 15.

Finally, the values of both LMAA and LACI are very good in cases where baseboard heating is included.

Extreme Winter and Summer Cases—Variation of ACH

The extreme winter cases show good values for the Fanger indices, PPD and PMV, for all the values of ACH considered. In particular, the PPD values for all these cases are below 20%, while the PMV values are all below ±0.5.

For the uniformity index, ADPI, the picture is more complicated. As pointed out above, the values for ADPI are generally good for extreme winter cases in which baseboard heating is used and poor for cases where it is not. However, there are a couple of subtleties as ACH increases.

- The value of ADPI for Case 4 (4 ACH) is lower than that for Case 1 (2 ACH). The reason for this can be seen from the LMAA values for these two cases. In particular, in Case 2, the jet from the remote diffuser penetrates down into the room more than it does in Case 1. This results in a drop in the ADPI value because that calculation picks up the higher velocity jet. However, the presence of the remote diffuser jet dramatically reduces the LMAA value around the patient.
- Cases in which the value of ACH is 8 have lower ADPI and LACI values than cases in which the ACH is 6. This can be attributed to the effects of the remote diffuser mentioned above.

As expected, the best values for linear slot diffuser only cases are seen for ACH values of 10.

The extreme summer cases show good values for all the ventilation indices, irrespective of ACH. This is because, as the ACH values are generally high, mixing is very good for all these cases.

Extreme Winter and Summer Cases—Ceiling vs. Low Level Exhausts

The results show a general benefit in using low level exhausts over ceiling exhausts for extreme winter cases, particularly at low ACH. For example, compare the results for ADPI from Cases 4 and 6. The reason for this is that the low level exhausts provide extra mixing and so help break up the temperature stratification in the room.

The difference between the ceiling and low level results for the summer cases are much less pronounced. This is because the summer cases generally have much better mixing and are considered at higher ACH values.

Extreme Winter and Summer Cases—Different Diffuser Combinations

Definitive conclusions are more difficult to draw for this test because of the smaller amount of cases that do not consider the linear slot diffuser only combination. However, the following points can be made regarding the different combinations:

- For extreme winter cases at 6 ACH, the radial diffusers appear to be the best diffuser type to use in combination with baseboard heating. The flow field pattern from the diffuser only for Case 14 is shown in Figure 17—mixing also occurs from the baseboard heater in the room. The reason for the success of this diffuser appears to be because there are no strong jets associated with it as there are with the linear slot diffusers, leading to a more even distribution of the air flow.

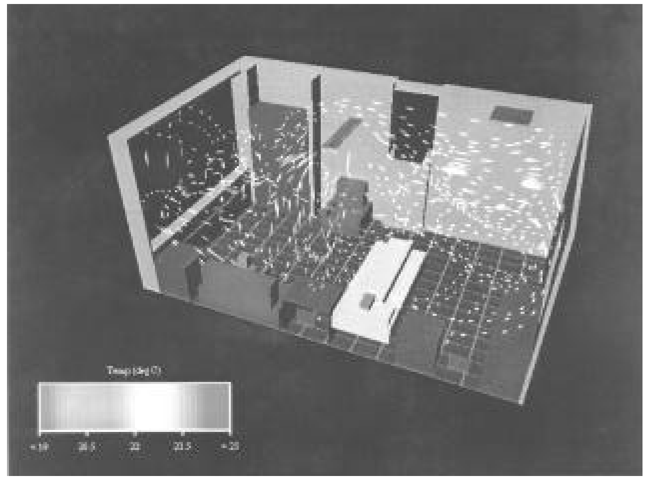

Figure 17 ***F****low pattern in Case 14 (radial diffuser).*

The one note of caution to make regarding this diffuser is that it is sensitive to dumping when the ΔT between the supply temperature and the mean room temperature is high (Memarzadeh 1998). This means, therefore, that this diffuser is not appropriate for use in summer cases in which all the cooling is done by the diffusers alone, as they are in this study. In these cases, the diffuser jet flow is likely to dump, creating a column of cold air in the center of the room above the patient.

- For the extreme winter cases considered at 6 ACH, the use of linear slot diffuser only combinations provide better conditions than those provided by linear slot/ low induction diffuser or low induction diffuser only combinations. The reason for this is that the remote linear slot diffuser provides better mixing than a low induction diffuser.
- In summer cases, the sensitivity of the results to the diffuser combination is reduced. This can again be attributed to better mixing in the summer cases compared to the winter cases.

Typical Winter and Summer Cases—General Results

As noted above, the lessons learned from the extreme weather condition cases were applied to more “typical” summer and winter day conditions. In particular,

- baseboard heating was applied in the typical winter cases,
- low level exhausts were used,

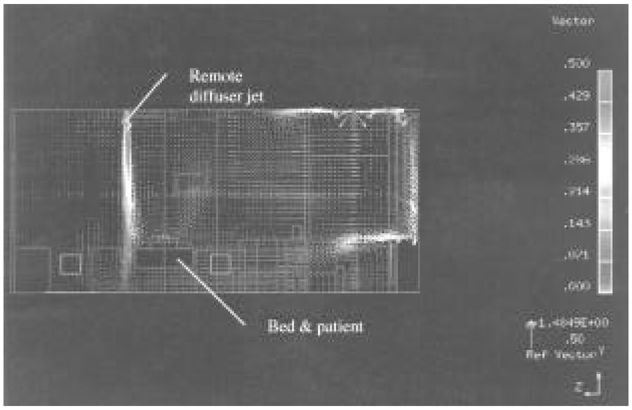

Figure 18 *Vector plot of flow field midway through center of remote diffuser (Case 30).*

- a remote diffuser was used at low (6 or below) ACH.

The results show that, for either typical summer or winter cases, the use of the remote diffuser directed vertically downward is dangerous. In the extreme winter condition, the warm remote diffuser jet is prevented from impinging too far into the room by the stratification of the air. In these typical cases, however, the supply temperature is much closer to, or lower than, the average room temperature, and the jet is likely to penetrate far enough to make the patient or physician uncomfortable; see, for example, Figure 18, which shows the flow field from Case 30.

A safer configuration is, therefore, to use the remote diffuser in two-way mode in typical scenarios. This means that for the typical winter cases considered here, a value of 4 ACH (though just barely) is required to ensure values of ADPI similar to those present in summer conditions. However, there are two important points to make here:

- For this particular configuration, increasing the ventilation rate to 6 ACH shows a clear improvement in LMAA.
- The configuration is marginal in terms of the use of baseboard heating. In particular, the total loss to be accounted for is only 130 W. If the decision were made not to use baseboard heating here, a higher value of ACH (5 or 6) should be used.

In summer conditions, the values of ADPI are very good for even the lowest ACH value considered (4 ACH in Case 34) due to good mixing conditions. In typical summer cases, therefore, the engineer designing the ventilation system in the room has some flexibility to keep the ACH low and pay the cost of cooling the air significantly or to increase the value of ACH and save on the cost of cooling the air.

Sensitivity of PPD and PMV to Metabolic Rate

In this study, values were assumed for the various parameters that contribute to the calculation of PMV and PPD, namely, air temperature, mean radiant temperature, air velocity, vapor pressure, clothing level, metabolic rate, and external work rate. It is, of course, important that the assumptions about these parameters be made correctly for the facility being considered. For the patient room, this is a challenging prospect since such parameters as clothing level and metabolic rate can be wildly different for medical staff and visitors compared with the patient.

From the formula for PMV, the most dominant parameter can be seen to be metabolic rate. In this study, the metabolic rate was assumed to be 1.2 met, a value consistent with light sedentary activity (from EN ISO 7730). However, as this value is altered slightly, even down to 0.8 met, a value consistent with reclining, the effect on the values of PMV and PPD are dramatic, as seen in Table 2. This could, therefore, be a typical example of using experimental data outside its range of applicability. Such low values of metabolic rate are probably not realistic since the patient would normally be virtually decoupled from the room environment by bedding, and the sensitivity demonstrates the dangers of applying such a subjective index.

CONCLUSIONS

The primary conclusions to be drawn from this study are as follows.

- In extreme winter cases, baseboard heating must be used in order to produce good conditions in terms of thermal comfort, uniformity, and ventilation effectiveness. Without baseboard heating, these conditions cannot be created, even at relatively high air change rates per hour (ACH).
- Although good values for thermal comfort and Air Diffusion Performance Index (ADPI) can be obtained at 2 ACH for extreme winter cases with baseboard heating, high values of local mean age of air (LMAA) indicate that the patient would experience somewhat stuffy conditions. The most effective value for extreme winter case ACH, therefore, appears to be 6, as this produces values of ADPI similar to those produced in summer conditions (above 80%) while also giving good values for LMAA and Local Air Change Index (LACI).
- The general results for LMAA are consistent with results from recent experimental studies (Han et al. 1999). In particular, this current study shows that the values for LMAA for summer (cooling) conditions are much lower than for equivalent winter (heating) conditions without baseboard heating.
- Low level exhausts produce better conditions than ceiling level exhausts for extreme winter cases at low ACH. This is because of better mixing conditions in the former cases.
- A single radial diffuser and baseboard heating appear to be the best ventilation system for extreme winter cases. Caution should be exercised in using this combination in summer cases where the cooling is only done through

the supply diffuser because of the risk of dumping.

TABLE 2
Effect of Metabolic Rate (Activity Level) on PPD and PMV

Case	Sedentary, Light Activity-----Relaxed Reclining					
	Met Rate = 1.2		Met Rate = 1.0		Met Rate = 0.8	
	PPD (%)	PMV	PPD (%)	PMV	PPD (%)	PMV
1	5.8	–0.18	23.1	–0.92	79.4	–2.07
2	5.4	–0.12	20.5	–0.85	75.6	–1.98
3	11.5	–0.53	35.5	–1.19	85.0	–2.26
4	6.55	–0.26	27.3	–1.03	84.5	–2.20
5	6	–0.18	23.5	–0.93	79.0	–2.07
6	6.1	–0.22	24.9	–0.97	81.8	–2.12
7	6.8	–0.29	28.6	–1.05	85.8	–2.23
8	6	–0.19	23.9	–0.94	80.1	–2.09
9	6.4	–0.25	26.8	–1.01	84.0	–2.18
10	6.7	–0.28	28.2	–1.05	85.4	–2.22
11	6.1	–0.2	24.4	–0.95	80.7	–2.10
12	6.8	–0.29	28.6	–1.06	85.9	–2.24
13	6.8	–0.28	28.4	–1.05	85.7	–2.23
14	6.6	–0.27	27.4	–1.03	84.7	–2.20
15	12.7	–0.56	44.5	–1.37	94.0	–2.65
16	7.8	–0.36	32.4	–1.14	89.1	–2.24
17	8.6	–0.38	34.0	–1.17	89.1	–2.38
18	7.4	–0.33	31.1	–1.11	88.0	–2.30
19	11.8	–0.55	43.9	–1.37	94.8	–2.63
20	10.4	–0.48	40.0	–1.29	92.6	–2.53
21	7.8	–0.36	32.3	–1.14	89.1	–2.34
22	9	–0.42	36.1	–1.21	91.0	–2.43
23	7.5	–0.34	31.3	–1.11	88.1	–2.31
24	12.6	–0.58	46.4	–1.42	95.8	–2.69
25	9.9	–0.46	39.0	–1.27	92.6	–2.51
26	14.2	–0.65	50.3	–1.49	97.0	–2.79
27	10.3	–0.49	40.3	–1.30	93.4	–2.54
28	10.4	–0.48	40.3	–1.29	93.1	–2.53
29	10.2	–0.48	39.8	–1.28	92.8	–2.53
30	6.1	–0.23	25.4	–0.98	82.6	–2.14
31	6.6	–0.27	27.8	–1.04	85.2	–2.21
32	6.1	–0.22	25.0	–0.97	82.2	–2.13
33	6.7	–0.28	28.3	–1.05	85.7	–2.23
34	8.0	–0.37	32.5	–1.14	89.6	–2.36

TABLE 2
Effect of Metabolic Rate (Activity Level) on PPD and PMV

35	8.1	–0.38	33.8	–1.16	90.2	–2.38
36	7.8	–0.36	33.1	–1.15	89.1	–2.34

- In the extreme summer cases considered here, there was not much variation in the results on changing ACH, diffuser combination, or exhaust location. This is because of the very good mixing conditions provided in summer cases.
- In the typical winter and summer cases considered here, the remote diffuser should not be directed vertically downward, as the jet could cause discomfort for the patient and/or physician. The remote diffuser should be operated in two-way mode.
- In the typical winter cases considered here, a value of 4 ACH would ensure values of ADPI in a room similar to those in summer conditions (above 80%). However,
    - an increase to 6 ACH improves the LMAA around the patient and
    - if baseboard heating were not used, the ACH would have to be increased slightly to improve the mixing. In this case a value of 5 or 6 ACH would be needed.
    - In the typical summer cases considered here, the values of ADPI were good for all values of ACH considered. This result gives the design engineer good flexibility in the design and operation of the ventilation system for these conditions.

From the concurrent study into minimizing the risk from airborne organisms in hospital isolation rooms, similar conclusions have been drawn with regard to the use of baseboard heating and ventilation systems in summer cases.

In particular, the study shows that the removal of particles potentially containing bacteria—either through ventilation through exhausts, sticking to the wall, or killing through ultraviolet germicidal irradiation (UVGI)—is dramatically increased when using baseboard heating compared with not using it.

Further, the isolation room study shows that the increase of ACH beyond 6 ACH for summer cases, and winter cases with baseboard heating, provides diminishing returns with regard to the number of viable particles. Therefore, this ties in well with the recommendation in this study of using 6 ACH for thermal comfort of the patient.

With regard to exhaust location, the isolation room study shows an improvement in using high level exhausts over low level exhausts. However, this conclusion is made for the particle release points considered.

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DISCUSSION

Michael A. Humphreys, Senior Researcher, Oxford Brookes University, England: It is good to see the question of thermal comfort in hospital rooms quantitatively assessed.

The calculations treat the “adiabatic” condition. Would the authors like to comment on the likely effect on the resulting temperatures and velocities, if due allowance were made for heat flowing in and out of the structure as a result of the diurnal cycle of outdoor temperature and of the incident solar radiation to the interior?

Farhad Memarzadeh: While the interior walls, floor, and ceiling were accounted for as adiabatic (it was assumed that the room was surrounded by rooms of similar nature), the effects of changes in outdoor temperature and incident solar radiation to the interior were included in the calculations through the glazing.

In particular, extremes of both summer and winter conditions were considered, as well as more typical values.

Humphreys: To calculate PMV, it is necessary to assume values for the thermal insulation of the clothing of the patient and the visitor and for their metabolic rates. What assumptions were made and how were they justified?

Memarzadeh: The metabolic rate was assumed to be 1.2 met, a value consistent with light sedentary activity (EN ISO 7730) and a combination of a variety of activities—see *ASHRAE Fundamentals 1997*, page 8.7, intermittent activity calculation.

The clothing level was set at 0.76 clo. I assumed that the visitor had trousers and a long-sleeve shirt (0.61 clo, *ASHRAE*

Fundamentals 1997, F8.8), and that he was in an armchair, which added another 0.15 clo (same source).

The research concentrated on the visitor rather than the patient because of difficulties in nailing down figures for the patient.

For example, (1) as pointed out in the paper (section 7.6), the patient is virtually decoupled from the environment by bedding; (2) the Fanger indices do not account for differing levels of illness; (3) the patient may only be wearing a hospital gown, but the bedding would add clo, as would the sheets, and/or a robe.

Further, if either the patient or the visitor is out of the bed or chair, then the Met rate would go up, and the clo value would go down. I, therefore, tried to go for representative values, acknowledging that they may not be appropriate for all instances.

Based on this uncertainty, a sensitivity test was added to the paper on the most dominant parameter, namely Met value. If lower values are taken for either the clo or Met values, the PPD increases quite significantly. As there was not enough data to use universal values for these parameters, I cautioned the use of applying such subjective indices as PPD and PMV (see section 7.6 of the paper), and I would recommend the use of LMAA, LACI, and ADPI (in combination) as a better marker of the ventilation system in a room.