Hensen, J.L.M., Hamelinck, M.J.H. & Loomans, M.G.L.C. (1996).
Modelling approaches for displacement ventilation in offices.
Proceedings of the 5th International Conference Roomvent '96, July, pp. 1-8. Yohohama: University of Tokyo

Modelling Approaches for Displacement Ventilation in Offices

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ABSTRACT

After briefly indicating HVAC performance evaluation criteria, displacement ventilation in offices is used a case study to demonstrate merits and drawbacks of various computer modelling approaches for HVAC design and performance prediction.
The main conclusions are that each approach has its own (dis)advantages, different approaches should/could be used depending on the question to be answered, and that there is quite a lot of future work needed.

KEYWORDS: computer modelling and simulation, performance prediction, HVAC systems

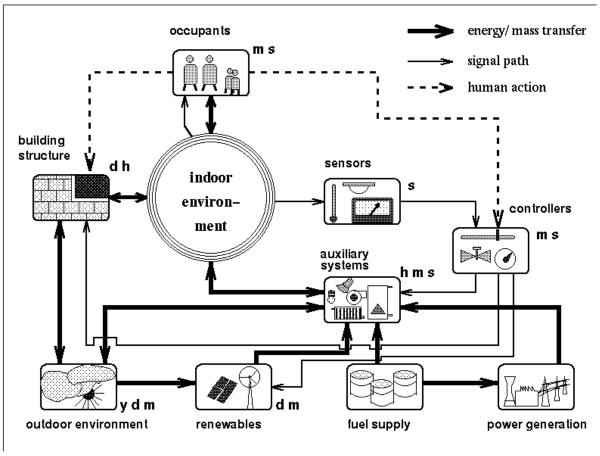

Figure 1 The building as an integrated, dynamic system

1. INTRODUCTION

Apart from the general need for energy efficiency and protection of the environment, building design engineers, environmental engineers and mechanical engineers encounter a vast range of additional 'challenges' in everyday practice. Very often these are related to time-variant interactions occurring between the various dynamic sub-systems which comprise a building as indicated in Figure 1.

For many practical problems it is essential to establish an integrated view because otherwise either important energy/mass flow paths are omitted or - which is even more common - important interactions between the various energy/mass flow paths are not taken into consideration. Performance prediction of displacement ventilation systems in office buildings is a practical example.

2. EVALUATION CRITERIA

Unlike for many industrial applications, the advantages of displacement ventilation over mixing ventilation are much less obvious in the case of offices.
When comparing various heating, ventilating and air-conditioning (HVAC) systems, the main evaluation criteria used by 'environmental engineers' are:
- energy consumption
- thermal comfort
- indoor air quality

In terms of energy it is not only cooling or heating demand which is of interest, but also electricity consumption by the distribution system (ie the fans). (Indeed, in terms of primary energy the latter is actually often much larger than the energy consumption for heating or cooling !)

In terms of thermal comfort we are interested not only in the common parameters such as air temperature, mean radiant temperature, relative humidity, and air velocity, which define whole body thermal comfort, but also in parameters related to local thermal comfort such as vertical temperature gradient, and turbulence intensity.

In terms of indoor air quality, interesting parameters are ventilation efficiency and contaminant concentration distribution in the room.

3. MODELLING APPROACHES

One of the most powerful tools currently available for analysis and design of complex systems, is computer modelling and simulation. Up to quite recent, simulation in the context of building design and building performance evaluation concentrated mainly on the building side of the overall problem domain (see eg Clarke 1985). We now see that modelling and simulation of HVAC systems is rapidly gaining more and more interest in both the building and environmental engineering communities.

Comparing different HVAC modelling approaches in general is the main subject of another paper (Hensen 1996). Here we want to focus on the air flow modelling part of HVAC system simulation. As indicated in Table 1., room air flow modelling approaches range from highly abstract and conceptual, via intermediate approaches, to very explicit approaches.

Table 1 Room air flow modelling levels

level		type
A	fully mixed zones	CONCEPTUAL
B	fully mixed sub-zones, combined for gross indication of main inter-zonal air flows and temperatures	|| || V
C	detailed intra-zonal flow and temperature field approach	EXPLICIT

The **LEVEL A** approach assumes fully mixed zones. In principle it should be possible to incorporate some vertical temperature profile depending on the type of HVAC system and other factors, but in practice these profiles are very hard to predict.
Advantages of this approach are the ‘user friendliness’ in terms of problem definition, and the straight forward internal representation and calculation procedure. This approach can easily be incorporated in a mass flow balance approach which would allow prediction of bulk flows through the whole building as caused by wind, temperature differences, and/or mechanical systems.
The main disadvantage is that - depending on the type of HVAC system - it does not represent the real temperature and flow conditions within a room very well.

In the intermediate **LEVEL B** approach a room is represented by a combination of fully mixed sub-zones, and the main inter-zonal flows have to be - more or less - specified by the user. Advantages are the relative ease for the user to define the problem, again the easy incorporation into a whole building air flow network calculation method, and it is possible to get some ‘global’ indication of temperature profiles. Main disadvantages are that - as a user - it might not be possible to pre-define the main inter-zonal flows, and, in case there is for instance an important radiation exchange between the sub-zones, there could be some internal representation and calculation problems.

In the **LEVEL C** approach, computational fluid dynamics is used for the room air flow modelling part.
The main advantage of this approach is the ‘richness’ of the results in terms of information detail regarding the flow and temperature field within a room.
Main disadvantages are the huge user effort in terms of problem definition, and the computational effort. It is practically only possible to make predictions for specific points in time, and it is quite difficult to integrate this method within general building energy simulation (see for instance Chen 1988, Niu 1994, Clarke et al. 1995).

4. CASE STUDY

One of the ways to elaborate the above is by means of a case study. The background of the case being considered here, is that recent years have shown a considerable interest in displacement ventilation systems. (For principles and applications see eg. Jackman 1991.)
Displacement systems have a higher ventilation efficiency than mixing systems. Since, in case cooling is needed, heat dissipation may be regarded as another indoor air "contaminant", displacement ventilation is expected to have a lower cooling energy consumption than mixing ventilation systems.
Although displacement ventilation is a proven technology for factories and workshops with a high degree of heat dissipation or air contamination, the advantages are not so clear cut for office type applications.

The case study concerns a "standard office module" located in The Netherlands (temperate sea climate). This module is an office (4.5 *m* deep x 3.6 *m* wide x 2.7 *m* high) with only one external wall (3.6 *m* wide x 2.7 *m* high). The West facing facade has a window area of 40% (double glazing). (For a more elaborate description, see Cox and Elkhuizen 1995.) The following sections discuss how, and to what extent, the previously described modelling levels can assist environmental engineers in the process of comparing displacement versus mixing ventilation system.

4.1. Level A: Fully Mixed

Although we did not actually use this method it will be obvious that in terms of cooling energy predication this method will not be able to distinguish between displacement and mixing ventilation since it is not possible to take the vertical temperature gradient into account. It will be quite possible however to estimate the differences in fan electricity consumption. In terms of thermal comfort, it should be possible to make predictions regarding whole body thermal comfort, but not regarding local thermal discomfort. In terms of indoor air quality this method will not be able to give any information other than global, whole building information regarding bulk air flows including contaminant dispersal.

4.2. Level B: Intermediate Approach

When a room with displacement ventilation is cooled, there will be a vertical air temperature gradient due to the upward air flow. Like most other building energy simulation systems, the current ESP-r version (ESRU 1996) assumes complete mixing of the zone’s air.

Thermal stratification was approximated by sub-dividing the office in stacked (fully mixed) sub-zones separated by "fictitious" floors. In case of displacement ventilation (in cooling mode) the dominating flow in the room can be approximated as indicated in Figure 2. Heating or cooling of the room is controlled on the basis of the air temperature sensed at a height of 1.2 *m*. (A more elaborate description of the modelling can be found in Hensen and Hamelinck 1995.)

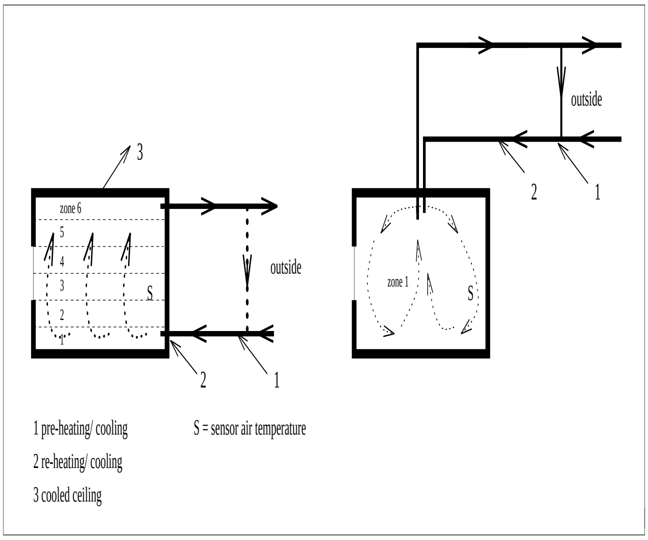

Figure 2 Modelling displacement ventilation and mixing system

4.2.1. Model Verification

The model was verified with experimental results (Cox 1990) from a displacement system in an office very similar to the standard office module described above.

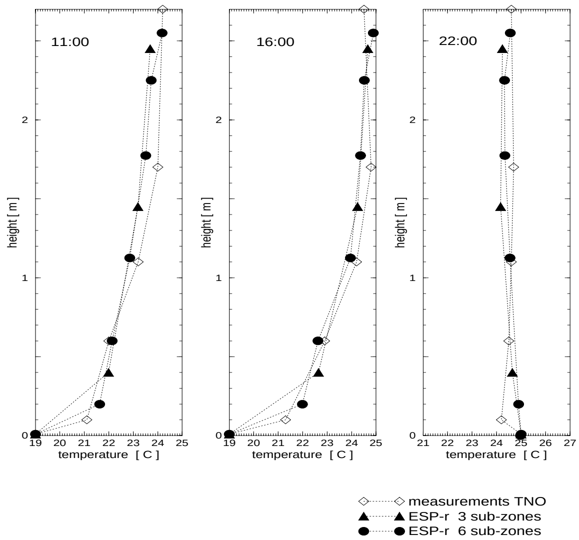

Figure 3 Vertical air temperature gradients for different times during 8 August 1989

In order to verify the model, a number of simulations were carried out with models incorporating respectively three and six stacked sub-zones. Figure 3 illustrates both measurements and "predictions" for vertical air temperature gradients during 8 August 1988.

From Figure 3 it can be seen that the predicted vertical air temperature gradients during office hours (due to the relatively cold incoming air) quite well represent the measured values. The maximum error in the predicted air temperatures is approximately 1 K.

4.2.2. Simulations & Results

Simulations were carried out in order to compare a displacement system with a mixing system in terms of annual energy consumption. As implied in Figure 4, the basis of comparison is the same air temperature at 1.2 m above the floor. During office hours (8:00 - 18:00) the set points (air temperature) were 20°C for heating and 22°C for cooling. In the period 6:00 - 8:00 the set point for heating increases (linear) from 15°C to 20°C. Outside these hours and during weekends the set point for heating is 15°C. In the case of a cooled ceiling the cooling set point for the air temperature is 1.5°C higher in order to achieve a comparable operative temperature.

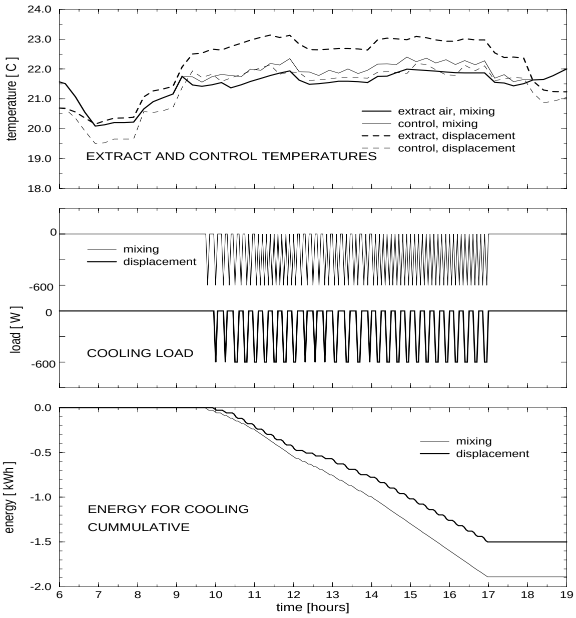

Figure 4 Temperatures, loads and energy consumption in case of cooling mode (for 25 August of a reference year)

Figure 4 shows some simulation results in terms of temperatures, loads and energy consumption. The energy consumption for cooling is lowest in case of the displacement system. This is actually only the case for casual gains lower than about 35 W/m^2. For higher casual gains, and a ceiling height of 3 ... 3.5 m, a displacement system needs an additional cooled ceiling, and the energy consumption for cooling will be higher than in case of a mixing system only.

If the fan electricity consumption is taken into account, than the difference between displacement and ventilation systems disappears, even at low casual gains.

As was done in another paper (Hensen and Hamelinck 1995), it is quite possible to predict whole body thermal comfort and local discomfort due to vertical temperature gradients. However it is not possible to predict discomfort related to turbulence intensity.

In terms of indoor air quality, this type approach will not be able to generate anything more detailed than global, whole building information regarding bulk air flows including contaminant dispersal.

4.3. Level C: Flow Field Approach

Unlike the previous approaches, computational fluid dynamics (CFD) enables prediction of the air flow and temperature field within a room. Therefore, it should be possible to predict the performance of displacement ventilation in an office. However, unlike the previous approaches, CFD does not allow prediction of for instance annual energy consumption.

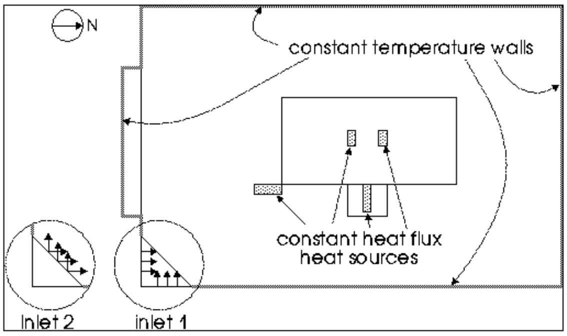

Figure 5 Boundary conditions reference case

For the current work, Fluent (Fluent 1995) was used for the CFD approach. The reference case was solved using the standard k-epsilon turbulence model. The boundary conditions are shown in Figure 5. The radiant heat transfer was accounted for by lowering the heat flux at the heat sources in the room, as determined from measurements (see Cox 1990). The grid distribution is crucial in CFD-modelling.

Boundary fitted coordinates were not used because of the rectangular geometry of the room. The flow at the inlet therefore entered the room at an angle of 45° to the grid lines. Two inlet models were used. For the reference case, inlet 1 was applied. In order to better approximate the heat transfer, the grid was refined near the heat sources. Due to the employed structured grid, this refinement was found throughout the calculation domain. Grid dependency was checked by increasing the number of grid points to a nearly doubled grid. The settings of the solution parameters were tuned from intermediate results and were applied as a solution strategy for subsequent CFD-simulations.

4.3.1. Model Verification

Results from the CFD-model were compared to the abovementioned measurement results for a displacement system in a similar office. A qualitative comparison of the flow in the room indicates a good agreement between the predicted and measured (visualized) flow field. Table 2 shows a quantitative comparison between measured velocities and (interpolated) calculated velocities for the reference case and the double grid case at five different points in the room.

Table 2 Air velocity at five different points in the room

point	height	measurement	calculation reference	calculation double grid
	m	m/s	m/s	m/s
SouthWest	0.1	0.08	0.167	0.208
SouthWest	0.6	0.28	0.098	0.126
Middle	1.1	0.05	0.062	0.067
Middle	1.7	0.05	0.023	0.020
NorthEast	0.1	0.10	0.068	0.076

Differences are found especially at the points near the inlet. This results from the inlet modelling; a mean velocity has been prescribed at the inlet, whereas in reality small apertures are applied to create a highly turbulent non-uniform flow field. Furthermore, steep velocity gradients (du/dy) are calculated near the inlet. Distinct velocity differences are calculated using a doubled grid, so that a grid dependent solution is found for the reference case. This difference can be explained as a result of numerical diffusion (Patankar 1980), because of the position of the inlet. The standard Power Law interpolation scheme was replaced by a second order upwind interpolation scheme to estimate the influence of this numerical diffusion. This model could not be brought to a converged solution however.

At the other points indicated in Table 2, a reasonable agreement is found when the accuracy of the measurement system at low velocities is taken into account. At these points the double grid case shows a close resemblance to the results calculated from the reference case.

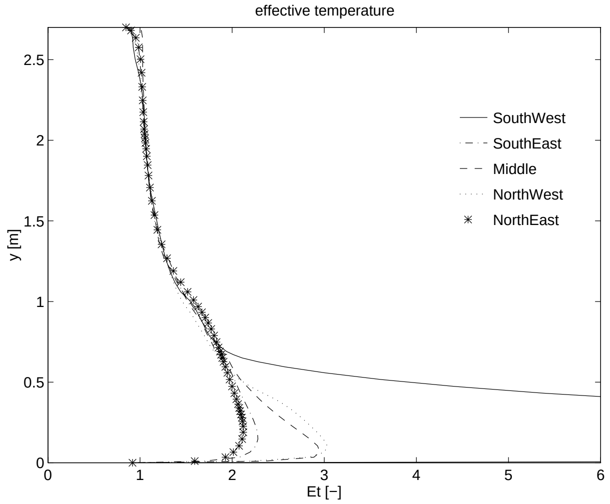

Figure 6 Predicted temperature effectivity at five positions

Figure 6 shows the vertical gradient for the temperature effectivity (E_t)) calculated for the double grid case at five positions in the room. The temperature effectivity is defined by:

$$E_t = (\eta_e - \eta_s)/(\eta_w - \eta_s)$$

where: η_e = air temperature at the exhaust,
η_s = air temperature at the inlet,
η_w = air temperature at specific location.

For a mixing ventilation system the temperature effectivity will be close to unity everywhere in the room since the exhaust temperature is assumed to be equal to the temperature in the bulk of the zone. An temperature effectivity above one, as in Figure 6, indicates a higher effectivity of the cooling system when compared to mixing ventilation. Figure 6 is in qualitative agreement with the measurement results. Table 3 gives a quantitative comparison between measured and predicted temperature effectivity.

Table 3 Measured and predicted temperature effectivity [-]

height	SouthWest			Middle			NorthEast		
m	meas.	ref.	double	meas.	ref.	double	meas.	ref.	double
0.1	4.28	13.87	28.59	2.35	3.30	2.90	1.79	2.32	2.03
0.6	2.93	2.52	2.43	1.57	2.28	1.99	1.50	2.16	1.92
1.1	1.24	1.52	1.17	1.28	1.56	1.37	1.26	1.63	1.39
1.7	0.95	1.22	1.10	0.98	1.18	1.10	1.01	1.18	1.10

Table 3 shows distinct differences between predictions and measurements, and between the standard and double grid simulation. However, absolute temperature differences (derived from the difference between measured and predicted temperature effectivity) are less than 1°C for the reference case and less than 0.5°C for the double grid case.

The difference between the reference and the double grid case results from an increased heat flux calculated at the (constant temperature) walls for the double grid simulation. This increase results in a higher mean air temperature (˜0.6°C). Nevertheless, also for the double grid case the heat transfer coefficient is still underestimated when compared to the experimentally derived heat transfer coefficient at the heat sources. In order to approximate the experimentally derived heat transfer coefficient, the grid distribution near the boundary must be refined and preferably a Low-Reynolds number k-epsilon model should be used (Loomans 1994). This reduced heat transfer at the walls influences other flow field variables.

4.3.2. Simulations & Results

In view of the scope of this paper, here we only want to report results regarding some flow field parameters: modelling of the inlet, the flow direction at the inlet, the turbulence intensity, position of the heat source, etc. Starting from the reference case, Table 4 indicates some of the considered variants. Table 5 summarizes the predicted velocities for these variants.

To compensate for the extra radiation heat loss, an extra heat flux is added to the heat sources placed in the room for variant 1. The amount of extra heat flux was determined from the convection/radiation heat transfer ratio as derived from the measurements. However, the under prediction of the convective heat transfer coefficient results in an increased radiant heat loss which leads to small differences in the overall flow field.

Table 4 Simulation variants

variant	based on	changed parameter
1	reference	radiation heat transfer included (extra radiation heat flux accounted for)
2	reference	modelling of the inlet
3	variant 2	vertical inlet angle: 45° upward
4	variant 2	horizontal inlet angle: 0°, parallel to the facade
5	variant 2	inlet turbulence intensity: 10%

Table 5 Predicted air velocity for the different variants at five different points in the room

location	height	ref.	variant				
			1	2	3	4	5
	m	*m/s*	*m/s*	*m/s*	*m/s*	*m/s*	*m/s*
SouthWest	0.1	0.167	0.176	0.144	0.067	0.079	0.144
SouthWest	0.6	0.098	0.077	0.122	0.201	0.056	0.122
Middle	1.1	0.062	0.064	0.062	0.069	0.163	0.062
Middle	1.7	0.023	0.007	0.011	0.016	0.113	0.010
NorthEast	0.1	0.068	0.067	0.068	0.070	0.058	0.068

Changing the inlet model causes differences near the inlet of the flow, but the flow field in the rest of the room remains nearly unchanged. Similar conclusions follow from the results when the vertical inlet angle is altered to 45° upward. Differences near the inlet are far more distinct and result in an increased thermally uncomfortable area near the desk. The flow field is totally altered when the horizontal inlet angle is changed. It results amongst others in a reduced temperature effectivity near the heat sources. Finally, a decreased turbulence intensity at the inlet has no influence on the flow field. This might partly be due to the numerical diffusion mentioned earlier.

As implied above, CFD requires considerable experience regarding modelling of the problem and the solution strategy. The simulation time is long: in the current case it ranged from more than one day for the reference situation (50,000 nodes) to about ten days for the (nearly) double grid (370,000 nodes) case. Due to these practical constraints, CFD will only be able to predict the conditions at specific points in time.

The above results show that, due to the limitations of the turbulence and numerical model, the boundary conditions, the mesh size and the grid distribution, a careful evaluation is necessary. This is further complicated by the amount of data which is generated by CFD. Although, in the current case, only a limited verification was possible due to lack of measurement results, it is felt that verification and validation should be an essential and integral part of CFD modelling and simulation. In our experience, starting from a reference case, an extended parameter study can be performed in a relatively short time. Compared to the laboriousness of indoor air flow measurements, this is thought to be the power of CFD.

Given the above, CFD does not allow predictions on cooling energy and fan electricity consumption, and it is restricted to predictions for specific points in time. In terms of whole body thermal comfort, it has the same possibilities as the previous mentioned approaches. Some authors seem to suggest that CFD would enable prediction of indices such as 'predicted mean vote' and 'predicted percentage of dissatisfied' for any location in a room, however this is highly questionable given the way these indices were derived in the first place. CFD enables prediction of local thermal discomfort due to temperature gradients and/or turbulence intensity. Although in the latter case it is not straight forward to express the turbulence intensity as used in CFD in the turbulence intensity as used in comfort analysis. Obviously, CFD is very interesting in terms of predicting indoor air quality parameters such as ventilation efficiency (perhaps expressed as temperature effectiveness as above) and contaminant concentration distribution. The latter especially for conditions within a room, but in the future also for whole building evaluations, when linkage with building energy simulation programs will be enabled. Such linkage will also provide more realistic thermal boundary conditions for the CFD analysis.

5. CONCLUSIONS

This paper briefly indicated HVAC performance evaluation criteria as commonly used by environmental engineers. Evaluating displacement vs mixing ventilation in offices was used as a case study for demonstrating merits and drawbacks of three levels of computer modelling approaches for HVAC design and performance prediction. The main conclusions in term of prediction potential of these modelling approaches are summarized in Table 6.

Table 6 Summary of prediction potential (-- = none, ++ = very good) for air flow modelling levels: A (fully mixed zones), B (intermediate), and C (CFD)

aspect	**A**	**B**	**C**
cooling energy	--	++	--
fan electricity	++	++	--
whole body thermal comfort	+	++	+
local discomfort, gradient	--	+	++
local discomfort, turbulence intensity	--	--	++
ventilation efficiency	--	0	++
contaminant distribution	-	-	++
whole building integration	++	++	--
integration over time	++	++	--

It is evident that each approach has its own (dis)advantages and different approaches should/could be used depending on the question to be answered.

In terms of future work, the following 'outstanding issues' can be identified:
- enabling incorporation of vertical temperature gradient in approach A
- modelling of 'fictitious surfaces' in approach B
- integration of CFD in general building energy simulation for approach C, and
- prediction of thermal comfort as affected by the flow and temperature field within a room.

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