# Numerical simulation and comparison of two ventilation methods for a restaurant – displacement vs mixed flow ventilation

*George* Chitaru[1,*], *Charles*Berville[1] and *Angel* Dogeanu[1]

[1]UTCB Faculty of Building Services, 021414 PacheProtopopescu Blvd., Bucharest, Romania

**Abstract.**This paper presents a comparison between a displacement ventilation method and a mixed flow ventilation method using computational fluid dynamics (CFD) approach. The paper analyses different aspects of the two systems, like the draft effect in certain areas, the air temperatureand velocity distribution in the occupied zone. The results highlighted that the displacement ventilation system presents an advantage for the current scenario, due to the increased buoyancy driven flows caused by the interior heat sources. For the displacement ventilation case the draft effect was less prone to appear in the occupied zone but the high heat emissions from the interior sources have increased the temperature gradient in the occupied zone. Both systems have been studied in similar conditions, concentrating only on the flow patterns for each case.

## 1 INTRODUCTION

Since the energy crisis in the 1970s the criteria for designing the HVAC systems has constantly changed[1].Due to the petroleum shortage, certain countries that were affectedhave implemented quite drastic measures to reduce the energy consumption of the buildings[2]. These actions have had a negative impact on the comfort of the occupants due to the reduced functionality and size of the heating and ventilation systems[3].Nowadays, there is a high interest in properly designing the HVAC systems so that a desired quality of indoor air is achieved with minimal energy consumption[4]. Most often, this balance is hard to be obtained as a higher comfort level usually demands higher energy costs[5], but certain systems are more suitable than others for specific situations[6]. Therefore, an engineer's choice can have agreat impact not only on the energy consumption of the building but also on the wellbeing of the occupants.

Taking into consideration the difficulties that appear when choosing the type of the HVAC system, the current paper comprises a numerical only study that focuses on the comparison between two common ventilation systems: a displacement ventilation system and a mixing ventilation system. Both these systems will be used to provide the necessary cooling load and fresh air during a summer scenario for a restaurant type building. The performance of each of these systems will be evaluated in the same environmental conditions (interior volume, number of occupants, cooling loads, occupant placement, etc.)based on the ventilation efficiency and the comfort level of the occupants. The comfort level is determined by certain factors like the temperature distribution/gradient in the occupancy zone, the draft rate near the tables zoneand by the velocity fields near the occupant area. The inclusion of these parameters allows to achieve a holistic view of the efficiency of each ventilation procedure which, correlated with the costs of each system, can offer important information regarding the viability of the solution before putting into practice.
Since the study is focused on determining the velocity and the temperature distribution in an open space area, with a non-uniform geometry, it was suitable to use a computational fluid dynamics program along with the finite volume method approach in order tocorrectly evaluate each case. The numerical simulations have been carried out with the academic version of the ANSYS AIM 18 software;therefore,certain simplifications to the model were necessary due to resource limitations.

## 2 PHYSICAL MODEL

### 2.1 Building's presentation

The current study concerns a self-service restaurant with the following characteristics:

*Corresponding author: chitarumadalin@gmail.com

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- Activity: Self-service restaurant
- Location: Constanta, Romania
- Occupancy: 302 [pers.] (75 [pers.] *)
- Area: 1253 m² (262 m² *)
- Ceiling height: H=3.2m

For the building in question the heating, cooling and humidity loads have been previously determined using the Romanian Standard I5-2010 [7]for the corresponding climate region:

- Cooling loads (summer): 83 [kW] (20 [kW]*)
- Heating loads (winter): 18 [kW] (4.5 [kW]*)
- Humidity loads during summer: 14 [g/s] (3.5 [g/s] *)
- Humidity loads during winter: 5 [g/s] (1.2[g/s] *)

**The values presented between ( ) are the values used for the virtual model, which accounts for only a quarter of the real building.*

**Fig. 1**Model of the real building

### 2.2 Air parameters

In order to properly size the ventilationunits for the virtual model, it was necessary to determine the external and internal air parameters and to also calculate therequired supply airflow needed to obtain an adequateair change rate.
The calculation procedure has been taken from the Romanian Standard I5-2010, as it follows:
The set point for the internal temperature has been considered at$\theta_c$ =26 [°C] with a relative humidity of 50%.
To calculate de required airflow rate of fresh air we used the relations below:

- Recommended specific flow per person: $q_{pers} = 25\ [m^3/(h*pers)]$
- Recommended specific flow per unit area: $q_{surf} = 2.52\ [m^3/(h*m^2)]$
- Total floor area: $S_{floor} = 262\ [m^2]$
- Fresh air flow required:

$$D_{FA} = 75[pers]*q_{pers} + q_{surf} * S_{floor} \quad (1)$$

$D_{AF} = 2535\ [m^3/h]$

### 2.3 Heat gains

Due to the activity inside the building, it was considered that,a large amount of the heat and humidity gains will come from the people and the food that is served. Moreover, significant heat gains would come from the exterior walls due to thedirect and diffuse solar radiation that enters through the glazing on the sides.
For the heat gains per person, for a restaurant, the Standard I5-2010 recommends a higher value that includes the heat gains from the warm food that is served:

- Heat gains from people and food: $Q_{(pers+nourr)}=12\ [kW]$
- Heat gains from glazed walls: $Q_{(par\ vitres)} =7.5\ [kW]$
- Heat gains from the inner walls: $Q_{(par\ int)} =0.5\ [kW]$

### 2.4 Ventilation systems

As it was said in the introduction, in this current study two common ventilation systems will be compared: a displacement ventilation system and a mixing ventilation system. In order to have a better understanding of these two systems and the choices concerning the sizing, in the following section the calculations that were used will be presented. Considering that the conditions for the two systems are the same, the same air flow calculation will apply for both cases.

#### 2.4.1 Sizing and calculation

First, by knowing the cooling and humidity loads for the summer season, the indoor air parameters and the introduced air parameters have been determined using the Mollier diagram.

**Table 1**Air parameters determined with theMollier diagram

| Parameters | t | x | φ | h |
|---|---|---|---|---|
| unit | [°C] | [g/kg] | [%] | [kj/kg] |
| $I_v$ | 26 | 11.5 | 50 | 53.5 |
| $C_v$ | 20 | 9 | 60 | 42.5 |

Where:$C_v$ is the blowing temperature and $I_v$ is the indoor comfort temperature.

With these parameters, the total supply airflow required to maintain the indoor temperature at 26 [°C] can be calculated. For this calculation, two formulas are proposed,from which the most important value will be chosen for the design. The first one is based on the enthalpy of the air and the other one on the humidity of the air:

$$D_{intake} = Q_v / \Delta h \quad [kg/s] \quad (2)$$

$$D_{intake} = G_v / \Delta x \quad [kg/s] \quad (3)$$

Where: $Q_v$ is the total heating loads [kW], Δhis the enthalpy difference [kj/kg], $G_v$ is the humidity gain [g/s] and Δx [g/kg] humidity content difference.

The maximum of the two airflow results has been chosen, which in our case corresponds to the necessary value to eliminate the excess humidity content. With the calculated air flow, the air change rate can be determined and then compared to the recommended values of the Standard I5-2010 to verify the choice.

In this case, the standard I5-2010 recommends a value between 5-10 [$h^{-1}$] for non-smoking rooms, and our result was 8.3 [$h^{-1}$]. Next, the value for the supply air flow rate has been calculated for each of the ventilation units.

### 2.4.2 Mixing ventilation

The mixing ventilation is the most widespread ventilation system technology and it is characterized by supplying air in such a mannerthat the entire room volume is fully mixed.
In the mixing ventilation case, the fresh air is introducedand extracted through the ceiling after it has been mixed with the interior air[8].
The dimensions used for the inlets and outlets are the following:

- Inlets: 0.5*0.43 [m]
- Outlets: 0.5 [m²]

Since, for this case, the outlets and inlets are both placed on the ceiling it was taken into consideration that a sufficient gap between the intake and extraction grilles is necessary to avoid a potential negative interaction.

### 2.4.3 Displacement ventilation

Displacement ventilation systems are characterized by an introduction of the air at low velocity, which causes minimal induction and mixing. Displacement inlets introduce the air through the lower part of the room and outlets are located inthe higher part of the room. Displacement ventilation systems use the buoyancy forces generated by the interior heat sources like the occupants. This way, polluted air and internal heats gains are removed more efficiently [9].
Considering the dimensions of the room, it was decided that four displacement unitsshould be installed (one in each corner of the room) as it presented in Fig.2.
To properly choose the dimensions of the units, the inlet air flow on each unit must first be calculated, taking into account the thermal loads, the parameters of the air introduced and the parameters of the indoor air. As previously specified, the total airflow will remain the same as for the mixing ventilation.In addition, for displacement ventilation, the velocity of the introduced air must be at very low valuesin order to avoid areas of discomfort for the occupants. Therefore, a proper dimensioning of the units was needed.

Each inlet unit has a height of 2.1m, and the surface depends on the placement in the room:

- Inlet unit 1: $S_{(unit\ 1)} = 2.87$ [m² ]
- Inlet unit 2: $S_{(unit\ 2)} = 2.87$ [m² ]
- Inlet unit 3: $S_{(unit\ 3)} = 3.5$ [m² ]
- Inlet unit 4: $S_{(unit\ 4)} = 1.97$ [m²]

The outlets are the same as for the mixing ventilation case: 0.5 [m²].

## 3 MATHEMATICAL MODEL

The geometry of the virtual model has been realized with the integrated ANSYS AIM module of Space Claim. In order to decrease the simulation time, and the complexity of the model, it was considered best to use a simplified version of the real building, therefore the model only contains a quarter of the open restaurant area.

The virtual building model presents the following characteristics:

- Two exterior walls: 20m + 13m
- Two adjacent walls: 26.5m + 11.25m
- Ceiling height: 3.2m
- A ground floor: 262 $m^2$
- A roof top: 262 $m^2$

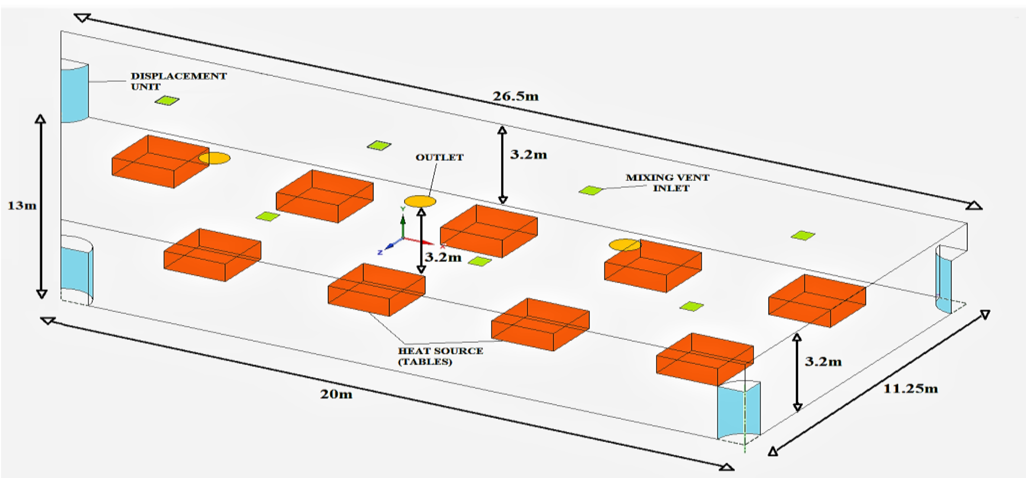


**Fig. 2**Virtual model of the restaurant (1/4 of the real building)

The meshof the virtual model has been created using an automated tetrahedral mesh generation with element size constraints imposed on important boundary surfaces: inlets, outlets and heat sources (face element size set to 0.05m). The mesh resolution has been set on its highest setting, resulting in aminimum element size of 0.003 [m], a maximum face size of 0.29 [m] and a maximum element size of 0.58[m]. No inflation has been added near the walls as there is no interest in studying the boundary-layer flow which would have increased the complexity of the model. The resulting mesh, and its characteristics, for each of the two cases can be seen in Fig. 3 and Fig.4 below:

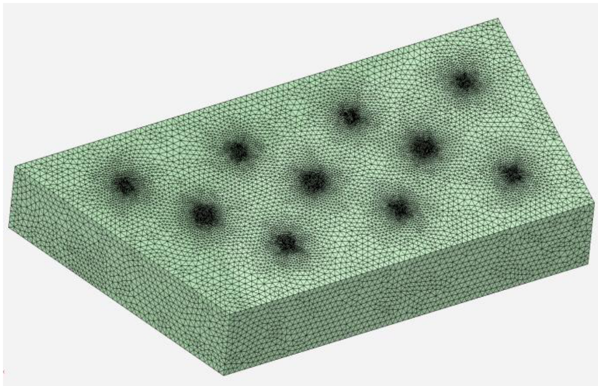

**Fig. 3**Mixing ventilation mesh (2.686.615 Cells, 480086 Nodes, Average Element Quality 0.84, Max Aspect Ratio 13.2, Average Skewness 0.21)

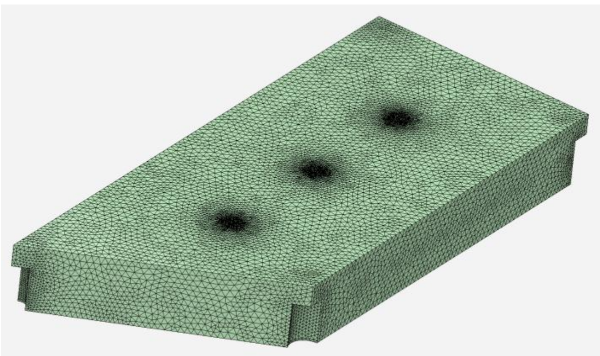

**Fig. 4**Displacement ventilation mesh (2.698.948 Cells, 612979 Nodes, Average Element Quality 0.77, Max Aspect Ratio 32, Average Skewness 0.22)

Considering the assumptions made in the previous paragraphs, for this study, a steady-state numerical simulation was used to determine the velocity and temperature distribution for a given situation. The Reynolds-Averaged Navier Stokes (RANS) equations were used, along with the Standard k-εturbulence model. To account for the natural convection, the Boussinesq approximation has been used to determine the buoyancy driven flows inside the room (especially near and above the tables). Next, the boundary conditions have been imposed for each of themodel's elements. The building envelope has been considered as adiabatic since the approximation of the heat-flux through each element could affect the behavior of the model.

The configuration for the numerical solver is presented in Table 2 below:

**Table 2.** Mathematical Solver Configuration

| ***Physics/Setting*** | ***Model/ Value*** |
|---|---|
| Calculation type | Static/Steady |
| Turbulence | RANS |
| | k-ε Standard model |
| Energy | Heat Transfer Model |
| Flow (Buoyancy) | Boussinesq approximation |
| Gas properties | Default air properties ($\rho$=1.183 [kg/m$^3$] ; $c_p$= 1006.30813404 [J/kg* K]; $\mu$=1.84480821367e-05 [Pa*s] |
| Numerical controls (Discretization and Solution controls) | Green Gauss Cell Based Gradient Method |
| | Second Order Upwind Advection |
| | PRESTO Pressure Scheme |
| | Coupled Pressure Velocity Coupling |
| | 0.5 High Order Term Relaxation Factor |
| Initial conditions | T=20 [°C] |
| | P=101325 [Pa] |

Therefore, all the heat gains that are supposed to come from the exterior walls (roof and floor included), and from the adjacent interior walls, have been introduced as an internal heat source that is uniformly distributed throughout the whole interior volume ($P_i$=8000 W).
On the other hand, the interior heat gains from the occupants and the food that is served (heat gains from electric/electronic devices, household appliances, etc. are not taken into consideration) are equally divided between 9 parallelepipeds with dimensions of 1.8mx1.8mx0.7m (LxWxH), which results in an approximate heat flux of:

$$\Phi_{(food+occ)}=12000\ W/54.36\ m^2=220\ [W/m^2] \quad (3)$$

Each parallelepiped is considered as a table with 8 persons placed around it and for each person an additional heat flux due to warm food is taken into account.

The boundary conditions for each of the surfaces are presented in Table 3.

**Table 3.** Boundary conditions for each case

| Boundary | Displacement | Mixing |
|---|---|---|
| Inlet(Velocity) | v=0.2 [m/s] | v=1.29 [m/s] |
| Outlet (Pressure) | P= $P_{atm}$ | P= $P_{atm}$ |
| Heat source term (interior volume) | $P_i$=8000 [W] | $P_i$=8000 [W] |
| Ext + Int Walls | Insulated (adiabatic) | Insulated (adiabatic) |
| Heat source term (Food+Occupants) | $P_{occ+food}$=12000 [W] | $P_{occ+food}$=12000 [W] |
| Ground floor | Insulated (adiabatic) | Insulated (adiabatic) |

## 4 RESULTS

For both cases the temperature and velocity fields were analyzed for different height planes, each one corresponding to a human body zone which is prone to draft effects: 0.1m for the feet zone, 0.6m for the pelvic region, 1.1m for the head zone at seated position and 1.7m for the head zone at standing position[10]. The interest was to determine an approximate draft sensation rate in certain zones of the room where the combined effect of high velocity, low temperature and high turbulence could affect the occupant comfort. The relation used to calculate the draft effect[11] in these regions is:

$$DR = (34 - t_a)*(v - 0.05)*0.62)*(0.37* v* T_u+3.14) \quad (4)$$

Where: $t_a$is the average temperature of the air in the region, v is the average velocity of the air in the region and $T_u$ is the local turbulence intensity

**Table 4. Draft results for targeted areas**

| Plane height [m] | Displacement | | | Mixing | | |
|---|---|---|---|---|---|---|
| | Average velocity* m/s | Average temperature* °C | Draught sensation** % | Average velocity* m/s | Average temperature* °C | Draught sensation** % |
| Feet 0.1m | 0.25 | 21 | 5.08 | 0.35 | 24 | 5.86 |
| Body 0.6m | 0.2 | 22 | 3.51 | 0.3 | 25 | 4.39 |
| Head (Seated) 1.1m | 0.15 | 25 | 1.76 | 0.4 | 25.5 | 5.81 |
| Head (Standing) 1.7m | 0.07 | 26.5 | 0.29 | 0.45 | 26 | 6.26 |

*Values calculated beneath air diffusers for the mixing ventilation case and above or around the tables for the displacement ventilation case. The most uncomfortable zones have been taken into consideration to avoid using average values that take into accountregions of no occupancy (room corners, entrance, etc.)
** Value calculated with a Turbulent Intensity of 10%

Analyzing the above table, it can be said that the displacement ventilation system presents a considerable reduction in draft sensation compared to the mixing ventilation;therefore, itmight be more suited for buildings that require high performance in terms of indoor environmental quality. Even though the draft percentage is not very high in any of the cases, for the mixing ventilation system there are certain "hot spots" where the level of discomfort can increase dramatically.

As it can be seen in the figures presented below, for the mixing ventilation case, the high inlet velocity combined with the high turbulence of the flow might increase the draft sensation in the occupant area due to increased heat losses through body parts that are not covered by any clothes. It can be seen in the iso-surfaces in Fig. 5 that the flow arrives at the head level of seated occupants (1.1m) at a velocity of 0.3-0.4 m/s with a temperature as low as22°C, right beneath the diffuser. In comparison, the ISO 7730 recommends an average velocity of 0.1-0.15m/s in the occupant area. The situation is even worse if we analyze the differences for a person standing right beneath an air diffuser (velocity between 0.5-0.7 m/s at a temperature of 22-26°C).Of course, the direction and position of the diffusers relative to the occupants is an important factor[8]. The direction of the flow in this case has been considered normal to the boundary surface (flow in the direction of the gravity).

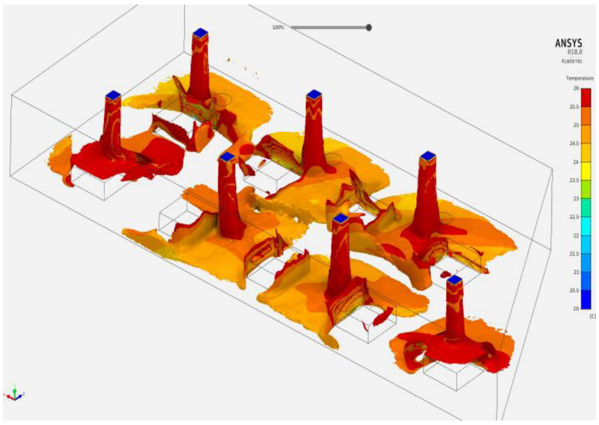


**Fig. 5**Iso-surface Velocity-Temperature coupling for mixed ventilation method (Blue = 20°C – Red =26°C)

A better representation of the cold air courant that comes from the mixed ventilation inlets can be seen in Fig. 6. Here it can also be seen that the temperature distribution on the height of the room is more uniform (gradient of only 2°C in occupant zone). But, in the end,the overall uniformity highly depends on the positioning of the inlets and on the angle of introduction. Therefore, a more exhaustive analysis of this method should be done in order to have a thorough comparison.On the other hand, for the displacement ventilation case, the draft sensation is drastically reduced due to the low inlet velocity and lower turbulence intensity near the occupant area.On the other hand, for the displacement ventilation case, the draft sensation is drastically reduced due to the low inlet velocity and lower turbulence intensity near the occupant area.

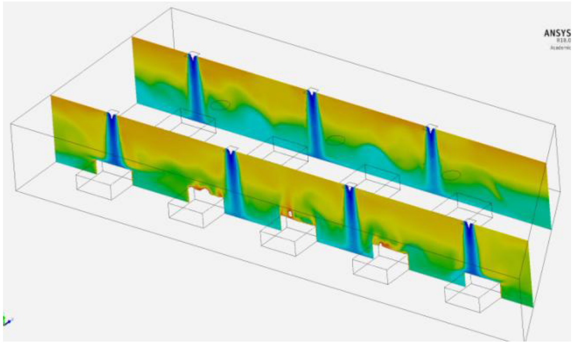

**Fig. 6**Temperature contours beneath the mixed ventilation inlets (Blue = 20 °C, Red = 30 °C)

The inlet temperatures are also usually higher, but in this case the loads where high enough to go as low as 20 [°C] for the supply air.Due to this fact, higher discomfort can only be found at feet level as the velocity of the air is higher near the floor level and the temperature of the fluid is low.As it can be seen in the figures below, the iso-surfaces show a more evenly distributed temperature across the room, while the gradient in the occupant zone increases from 24-26°C to 21-26.5°C. In Fig. 7 it can be observed how the fresh air that enters the room, and ventilate the entire area where the occupants are eating and then how the buoyancy forces caused by the people and the warm food drive the contaminated air towards the ceiling where it is extracted.

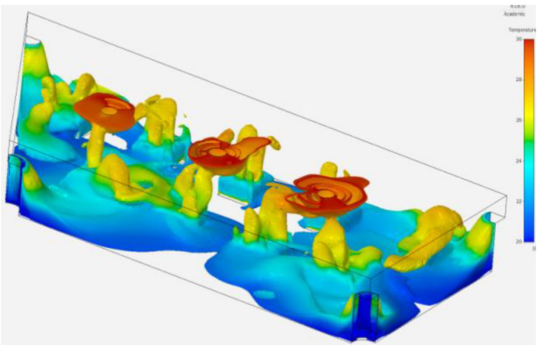

**Fig. 7**Iso-surface Velocity-Temperature coupling for displacement ventilation method (Blue = 20°C – Red =30°C)

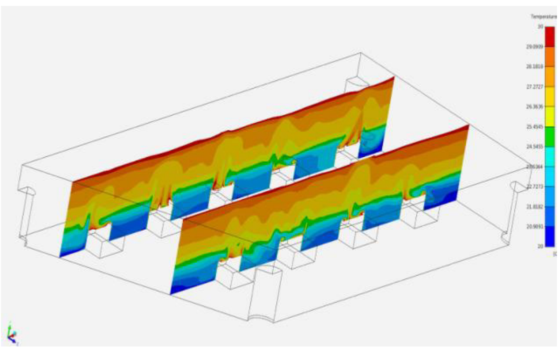

**Fig. 8**Temperature contours above the tables for displacement ventilation (Blue = 20°C – Red =30°C)

In Fig. 8, temperature distribution across the length of the room is almost uniform, and how the temperature increases gradually with the height, arriving at almost 30°C near the ceiling.

## 5 CONCLUSION

In the conditions of this paper, the displacement ventilation system has shown some advantages over the mixed flow ventilation system, concerning the comfort level in the occupant area. Taking into account these results and the complementary advantages of a displacement system (reduced noise, energy consumption), it can be said that this type of ventilation is well suited for an open-space restaurant area.

The current study has only shown a possibility for the comparison of these two systems, therefore a more thorough numerical and experimental investigation is necessary in order to fully support these claims.

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