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Air diffusion system design in large assembly halls. Case study of the Congress of Deputies parliament building, Madrid, Spain

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ABSTRACT

The paper describes and analyses the air distribution solution adopted in the refurbishment of the Spanish Congress of Deputies assembly hall. A new approach is proposed for the design of displacement ventilation systems. The system performance is methodically validated both with laboratory tests, computational fluid dynamics analysis and experimental field measurements related to the environmental performance of the built environment. In assembly halls, microclimate and displacement ventilation systems, which supply air by means of floor mounted outlets, have been shown to perform adequately. In this case study floor air supply is not feasible. The solution adapted uses parapet mounted diffusers, using a confluent jet flow parallel to the floor plane above the occupied area. All the tools used to analyse the performance of this system showed that the admixing overhead supply, with ceiling return, and flowing parallel to and above the occupied zone is drawn back into the occupied zone, creating a plume effect similar to that of floor mounted displacement outlets. Thus, this confirms that the displacement effect can be accomplished using a high induction turbulent overhead supply with ceiling return. Therefore supplying air directly into the occupied zone, at or near to floor level, is not a prerequisite to achieve the desired displacement effect. This report concludes that, using the correct design parameters, as set out in this paper, the displacement ventilation effect is independent of the plane of air supply. The conclusions suggest that accepted criteria for the definition and design of displacement systems should be redefined.

1. Introduction

Primary energy use has increased twofold since the 1980′s [1]. The building sector accounts for 40% of primary energy use [2] where more than 25% of buildings are classified as historic [3]. As most existing buildings have inefficient HVAC systems their refurbishment requires the use of high performance solutions which improve the indoor built environment and seek to reduce the carbon footprint of this building type. Air diffusion systems form an essential part of any HVAC building system. The adequate design and performance of the air diffusers will greatly influence the overall building system energy performance and the perceived indoor ambient quality, I.A.Q [4,5]. So, the question of how to make compatible thermal comfort, air quality and energy savings is central to the following research.

In this paper, it is addressed the HVAC refurbishment of an historical building like the Spanish Congress of Deputies, in Madrid. In this case, besides the functional requirements characteristic of large rooms, strong architectural constrains affect the performance of the air diffusion system, making canonical solutions unfeasible. This type of projects reveals as an interesting topic for the advancement of science, for it forces the designer to adapt existing solutions under unusual and demanding operation conditions which have to be tested prior to their installation. It also leads to implement new design methodologies, as the usual design rules are not fully operational. And sometimes, as is the case, the final results are also a source of unexpected findings that triggers new research challenges. All these topics are developed in the following paragraphs, where the main research objective is to establish a design procedure for air diffusion systems in assembly halls that guarantees air quality and energy efficiency in spite of architectural and functional limitations.

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When ventilating large rooms with high ceilings, such as in conference halls, theatres, and airport terminals, there are two principal issues a designer faces: the extreme temporal variability of the cooling thermal loads, which can vary from 30 W/m^2 up to 140 W/m^2 or even more [6], and their irregular spatial distribution [7]. Control systems can adapt the supply air temperature and air flow rate required to match the thermal load variations [8]. Nevertheless, preservation of acceptable comfort conditions in the occupied zone relies almost exclusively on the performance of the air diffusion system.

In tall building spaces, such as assembly halls, it has been commonly accepted that floor level displacement ventilation systems (or high induction microclimate diffusion, that leads to similar effects) are the most appropriate air diffusion system choice for this type of application [9,10]. In the cooling mode and when using this floor supply air system, the supply air introduced into the space at the floor level, is slightly colder than the room air in the occupied zone ($\Delta T \leq 5K$). The supply air spreads over the floor space and is induced upwards by heat sources in the occupied zone, creating heating plumes, taking advantage of the air stratification to enhance the warmer and more contaminated exhausted air up towards the ceiling extract [11,12]. However, one likely disadvantage with this type of air supply can occur when it is used in a very high density occupation case scenario (1 person ≤ 1.5 m^2), as is the case in the congress assembly hall, resulting in the possible risk of localized overcooling when the occupancy levels are below the maximum design capacity. During part load occupation and given the impossibility to predict where the occupants will actually be in the occupied space at any given moment the spatial specific heat load, W/m^2, will vary over the assembly hall floor area while the specific airflow rate, m^3/(h·m^2), will stay constant as it is necessary to ensure that the correct air quantity is supplied to each individual space, both for cooling and ventilation purposes. Normally, when using floor level air supply, the air introduced tends to spread evenly over the treated space, and for this to happen it requires that the air is not restricted in its movement by furniture, partitions, etc.

If, and to avoid overcooling, the supply air temperature is increased then the specific cooling capacity, will decrease in each individual supply outlet. Consequently, the performance of individual floor outlets, dedicated to each occupant space, may not be able to accomplish adequately the required thermal comfort due to the reduced specific cooling capacity of the outlet as a result of the higher supply air temperature [13]. This reduced cooling capacity cannot be compensated by the entrainment of supply air from adjacent floor outlets due to the zoning effect caused by the closed desks and benching. In the case of the congress assembly hall and given the compartmented seating areas with its closed desks and benching an unrestricted movement of the air at the floor level is not possible.

Since temperature stratification associated to displacement ventilation systems allows greater temperature difference between supply and return (when compared to admixture wall or ceiling air supply, with perfect dilution effect and little or no stratification) the air flow rate is determined by thermal loads must also guarantee the required minimum ventilation rate [14,15]. Likewise, and unless the design parameters are properly selected, the thermal environment and air quality in a room with stratified air distribution systems may not be optimal [16]. Given that the thermal gradient is a function of the specific heat load, and when the supply air flow rate cannot be lowered due to the required minimum ventilation flow rate, the advantages of displacement ventilation can be limited [17]. Furthermore, its ventilation efficiency (μ_v) is very much dependent on the movement of the heat sources (people, etc.), which tend to distort the vertical air pollutant concentration, by creating turbulent eddy currents where the people move within the occupied zone, arising in concentrations levels of contaminates in the breathing zone similar to those in an admixture system, $\mu_v \leq 1$ [18,19]. Ventilation efficiency is determined by $\mu_v = c_o/c_{rz}$, where μ_v is the ventilation efficiency, c_o is the pollution concentration in the outlet air (kg/m^3) and c_{rz} is the pollution concentration in the air in the residence zone, mean value (kg/m^3).

Due to afore-mentioned reasons, the advantages of displacement ventilation efficiency tend to diminish with the increase of occupancy density and movement within the occupied space. However, the stratification effect associated with displacement ventilation can be an advantage from the point of view of energy savings, contributing to the reduction of the carbon footprint of the building and its indoor environment [20]. All relevant technical standards, guidelines and publications indicate that achieving a stratification effect is only possible with displacement ventilation systems and not admixture air diffusion systems.

Effectively, by means of the air stratification, displacement ventilation systems create a bigger temperature difference between supply and return air than that usually associated with high turbulent admixture system flow. The greater the temperature difference between the supply and return air the less the specific airflow rate, m^3/(h·m^2), required. This increases system efficiency in air handling unit (AHU) and chiller equipment, reducing fan power consumption and chiller capacity as well as the required space for ductwork, etc.

With regards to the choice of angle and plane of supply of air the refurbishment of historical buildings can complicate matters more, as strict space constraints, coupled with decorative surfaces, paintings, etc. limit points of entry of the supply air into the treated space. Such spatial restrictions can compel the designer to choose an air diffusion system layout which cannot achieve optimal performance. It also reveals additional problems, e.g., appropriate air outlet sizing and positioning, which influences the selection of the spatial layout of the supply air diffusers.

The management of all these contradictory requirements (thermal comfort, energy savings and architectural limitations) demands an effective method for the design and start-up of the air distribution system in complex buildings refurbishment. The methodology proposed in the article makes use of full-scale prototype laboratory tests and CFD simulation of the room. Field measurements of room temperatures and velocities were also carried out, along with thermography imaging to determine temperature gradients and the asymmetric radiant room temperature.

All these tools, combined or on its own, are usual in recent research works in the field of ventilation design. For example, research on temperature stratification for displacement assessment is addressed by means of a combination of full-scale measurements, reduced-scale measurements, analytical methods and CFD models [21]. Interesting research has been carried out [22] on the use of CFD simulation to determine the size, locations, and shape of air supply inlets and the air supply parameters (i.e., velocity, temperature, and angle), restricted to small spaces. In other recent works [23] full scale test are used to set the proper values of the design parameters. It also serves to validate the computational analysis and to confirm the accuracy and convergence of iterative calculations and their tendencies [24]. Experiences concerning the fusion of several jets have been carried out by Ref. [25], with the outcome that the resulting jet is homogenized under swirling influence.

In the case of study, the CFD model and field measurements have shown that the multiple high induction swirl diffusers supplying air in a turbulent pattern from the horse shoe shape layout mid height parapet create a displacement ventilation effect in the occupied zone. In fact, this solution confirms the similarity of performance between wall-mounted admixing outlets and displacement diffusers, as set out by Nielsen [26] for cooling mode and by Ref. [27] for heating mode. It also suggests that the practical calculation rules set out for by Mundt for displacement systems cannot be applied [28]. The existence of the phenomena of turbulence associated with the proper functioning of the projected air outlet diffuser makes it advisable, and eventually, usual to work with mixing mode criteria for system design. It finally shows that, contrary to what is commonly affirmed, displacement effect does not entirely depends on the position of the supply outlets.

In this system design, the air diffusion equipment has been selected using the laboratory validated manufacturers data based and installation guidelines. On the other hand, this paper points out other options to consider in order to address the problem. Previous research of the authors has demonstrated that the use of CFD simulation is a reliable tool to deal with air diffusion options in HVAC refurbishment of large halls provided the results are validated by means of experimental measurements [29–31]. Thus, it can be said that this methodology makes it possible to address the design of an air distribution system in the refurbishment of a large room with satisfactory results when energy performance, thermal comfort and air quality are mandatory.

In the article, the characteristics of the building are firstly described, as well as the air diffusion outlets characteristics and position. Next, the four steps methodology used for the design of the system is explained, comprising the full scale laboratory tests, the substitution model, the room CFD simulation and the field tests. The results are finally shown and discussed, also disaggregated step by step to facilitate the understanding of the sequence of the obtained conclusions.

2. Material and method

2.1. Building and air distribution system

2.1.1. Building characteristics

The hall of the Congress of Deputies of Spain has a horseshoe shape. As can be seen in Fig. 1 the very high density occupation and furniture distribution impedes the use of under floor air supply without disturbing existing historic architectural finishes.

Three zones can be distinguished in the ground floor area (Fig. 1). The deputies' seats zone area, zone 1, has an area of 409 m^2 and occupancy of 380 persons. The seats occupy a semi-circular band of 7.8 m wide, where they are tiered in steps; the highest last row reaches a total height of about 4.2 m above the lowest first row. The specific internal sensible heat load in zone 1 is 136.25 W/m^2. The central lower part of the room, zone 2, with an area of 27.28 m^2, is occupied by various typing clerks and has a specific internal sensible heat load of 42.70 W/m^2. The presidency bench's, zone 3, occupies the diameter of the semicircle, has a total area of 129.09 m^2 and occupancy of up to 12 persons and has a specific internal sensible heat load of 47.25 W/m^2.

A semi-circular second upper floor tribune, zone 4, provides accommodation to the invited public, which can reach up to 480 people, resulting in a total occupancy in the congress hall of 875 people. The public will sit in the upper part of the hall and this again favours a stratification effect within the deputy space, when using the highest point in the ceiling as the return point. The specific internal sensible heat load of zone 4 is 115 W/m^2.

These thermal sensible heat loads include lighting (31 W/m^2), equipment (29 W per electronic device) and people (65 W of sensible heat load per person), all under conditions of maximum occupancy.

2.1.2. Characteristics of the air diffusion system

In the case of the Spanish Congress of Deputies and before the refurbishment, the supply air was originally delivered via perimeter mounted tangential airflow grilles in the gallery parapet. Its malfunction (inappropriate draughts, dumping of cold air, and excessive energy consumption) made it desirable to change the existing HVAC system. The existing tangential flow air supply grilles were replaced by the new diffuser system, maintaining the existing supply air position of the supply due to the architectural and decorative restrictions. the existing return air point used with the existing tengential airflow outlets, at near floor level, was changed to a return point in the highest part of the ceiling, with 100% return air, in order to enhance the stratification effect.

As it was explained before, underfloor air distribution is not feasible in this case due to the high degree of heritage protection of the building and also because of the difficulty to combat the spatial use and load distribution. Instead, air supply is carried out by means of 17 swirl type diffusers placed on the parapet of the semi-circular second floor grandstand, (position 2 in Fig. 1), spaced a distance of 2.65 m. Thus, supply air is delivered in cooling mode between 12 and 20 °C (variable depending on the system load) and is directed toward the centre of the room, with a jet throw of 14 m.

Supplementary to this system, linear diffusers, (position 2 in Fig. 2) in the upper stand, the public tribune, create a microclimate and separate this zone from the rising plume of the stratifying air in the hemicycle area. High induction ceiling mounted radial flow diffusers (position 3 in Fig. 2) for the perimeter gallery zone, placed under the tribune, have also been considered.

The Fig. 2 shows the type and position of the outlets as well as the return point in the upper part of the assembly hall. The layout of each of the 3 diffusers types indicated is symmetrical in the entire hall circumference.

As can be seen in Fig. 3 the air supplied from the parapet wall by the high induction swirl diffusers, is projected to achieve a spatial separation of the thermal load in height, divided by the drive plane at a 5.4 m height over the deputy's seats.

2.2. Methodology of design, analysis and validation

For the design and validation of performance of the air diffusion system, a four steps methodology approach was used;

Step 1. The discharge pattern of the air diffusion is measured in a

Fig. 1. Image of the deputies' hall during a parliamentary plenary session.

Fig. 2. Position of the supply air outlets and return.

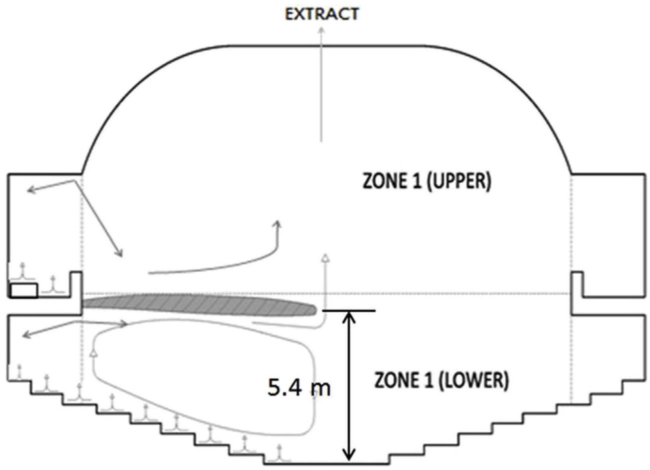

Fig. 3. Projected airflow pattern in the assembly hall.

laboratory test using a sectional full scale model of a section of the assembly hall [25].

Step 2. A CFD computer simulation of the air diffusion system performance is then carried out.

Step 3. The results of the laboratory test and the computational simulation are compared in order to validate the CFD output results.

Step 4. A field test is carried out to validate the performance of the air diffusion system.

In step 4 the following field analysis's were carried out: Smoke visualization tests to determine flow patterns which are represented in Figs. 3 and 10; thermography analysis; and measurements of air temperature, and velocity profiles.

A summary description of each step taken is shown below.

2.2.1. Laboratory test

In order to validate concluding design recommendations of the performance of the diffusers, a 1/1 scale test was made. The tests were carried out for an 1100 mm nominal length diffuser with air flows ranging from 347 m^3/h to 868 m^3/h, and temperature differentials between supply and return from −10K to + 3K. The tested diffuser contains 7 discharge chambers, 16 mm × 85 mm in section, laid out in a linear row to a length of 1100 mm occupying an equivalent spacing of approximately 2200 mm between each outlet.

Measurements were taken at the vertical axis of the combined confluent jet of the various discharge chambers, as well as in two parallel planes spaced ± 1 m [25]. In order to characterize the drop of the jet, measurements were taken at four different heights (0.00, 1.80, 2.97 and 4.14 m)

The instruments used in the experimental laboratory test model to measure temperature, are the Pt 100 Probes, from Herth GmbH Type: S (4Ltr.)-1-Pt100-A-3, 0-“NL” Tolerance: 1/5 DIN. To measure velocity and turbulence the TSI Anemometers Type 8465 with a similar tolerance where used, to 1/5 DIN.

As shown in Fig. 4A horizontal measurements have been taken at 2.6, 5.2, 7.8, 10.4, 12.0 and 14.3 m from the outlet, being 14.8 m the maximum jet throw for the maximum airflow rate considered. Although the project considers 17 outlets arranged in a semi-circular shape with the centre at a radial distance of 14 m, it was not necessary to arrange this layout in the laboratory as the tests results of 1 outlet will be the similar for all the units, also the confluent effect of the 7 chambers of each diffusor will be similar to that of all diffusors combined. The confluent jet effect of the various discharge elements measured in the laboratory is used in the computational analysis as a partial section simulation repeated for all outlets.

The above figure, Fig. 4B, shows the laboratory test facility layout where the whirl outlet is installed at a height of 5.4 m. The heat loads due to the occupants and equipment were simulated by means of heating plates distributed over the floor area. Thus, the heat output from the occupants and their electronic equipment was achieved with 16 heating plates with a heat output of 200 W each per unit, arranged in a length of 2–8 m and a width of 3.2 m between the measurement points.

2.2.2. Laboratory test model

Effects of high turbulent systems have to be simulated by substitution models [32,33]. Here, the laboratory test is used to calibrate the CFD substitution model. Fig. 5A and B show the air temperature and velocity fields (respectively) of the side wall mounted whirl type diffuser outlet laboratory measurements.

In Fig. 5A and B (these images do not represent a computational simulated model but real measurements taken in the laboratory and are colour scheme represented) the measurements shown correspond to a supply air temperature of 19 °C (a –6K temperature difference with respect to room ambient temperature) and an airflow rate of 848 m3/h. Note that in both Fig. 5A and B the yellow point shown indicates the reference point of the average velocity and temperature used to compare the experimental output values with the CFD simulation post process output.

This comparative analysis serves as a basis for validating the CFD output values.

2.2.3. CFD simulation

A high resolution mesh is used with 2.7 millions of hexagonal cells. The mesh resolution is variable depending on the geometry. To address this problem, PHOENICS simulation software was used for this work. A hybrid approach called parSol (partial solid treatment) is used [34]. This is a structured Cartesian mesh, modified on cell level to fit the body contour, a technique for improving the accuracy of flow simulations for situations in which a fluid/solid boundary intersects obliquely some of the cells of the Cartesian grid.

Within the domain, the mass conservation, movement quantity and energy equations must be solved simultaneously. This mesh guarantees robust enough results. The standard k-ε turbulence model has been used. The computational code simultaneously solves the Navier-Stokes equations and the energy equation, after 10,000 iterations.

The following Table 1 shows the values measured by the Laboratory test and by the final computational simulation. A comparison at one selected point (x = 10.3 m, H = 3 m) shows the following values, where there is less than a 2% deviation between the laboratory test measured values and the computational simulated values. This difference is considered more than sufficient to give an accurate representation of the experimental data when compared to the CFD results.

2.2.4. On site tests

2.2.4.1. Temperature and velocity measurements. The instruments used in the field test to measure temperature and velocity were the TESTO 435-4, using also a TESTO temperature-velocity probe type 0635-1553 with a temperature tolerance of ± 0.03 °C and a velocity tolerance of ± 0.03 m/s, or 4% of measured value.

As shown in Fig. 6 air temperature and velocity measurements were taken in 52 points uniformly distributed throughout the room, at a height of 1.2 m, with an air flow variable supply from between 30% and 100% of the nominal value. The supply air temperatures used during the tests were 28, 25, 23, 19, 15 and 12 °C respectively.

2.2.4.2. Thermography analysis. A Thermography analysis was carried out with a FLIR T335 infrared camera and analysed with the FLIR software.

In Fig. 7A and B two thermography images are taken, the first during a plenary parliament meeting. It can be seen in Fig. 7A that at the floor level there is an average room temperature of 22 °C which is

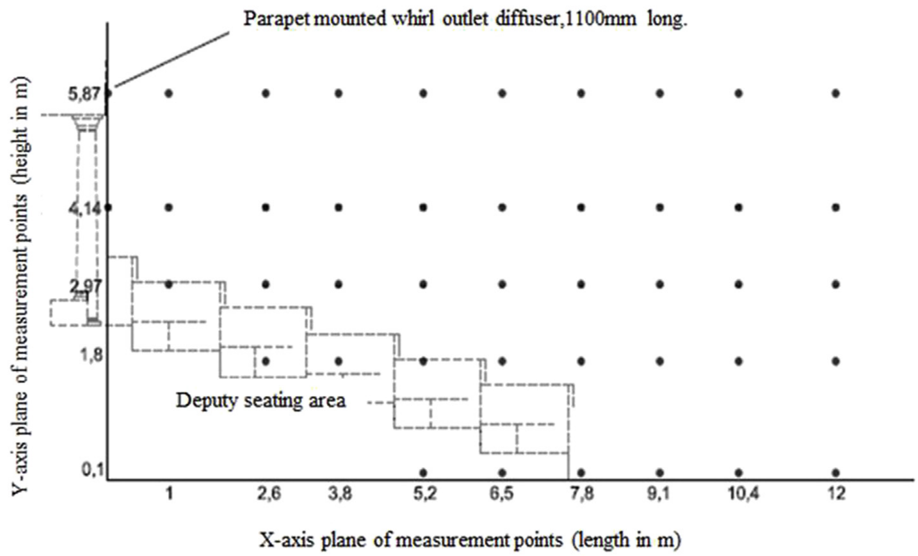

Fig. 4A. Measurement points of laboratory test on the X axis.

Fig. 4B. Image of laboratory test facility layout.

similar to that of the floor temperature in displacement ventilation systems. The thermographic image also captures the heat loads dissipated from the occupants and their electronic equipment. In Fig. 7B a thermography image visuals the plume affected created by an occupant in the seated deputy area and their electronic equipment.

In the photographic image of Fig. 7D the black painted plywood sheets, 1.8 m high and 0.6 m wide, where used to capture the occupied space temperatures when using the thermography imagining as shown in Fig. 7C. This analysis was carried out in steady state conditions.

The above thermography analysis only seeks to validate and confirm aspects of local comfort by visualizing the asymmetric mean radiant temperature of the assembly hall and especially within the seated area. This allows calculating the radiant temperature asymmetry, RTA, as stipulated by the UNE-EN-ISO-7730 standard and allows determining the predicted percentage of dissatisfied people, as reflected in the Table 3.

2.2.4.3. Smoke tests. The smoke test was carried out with a supply air temperature of 19 °C and an average room temperature of 23 °C. The maximum projected airflow rate, 868 m^3/h, per outlet was used. A fog making machine, type Fansteck, model FM-02 was connected upstream into the ductwork supplying air to the turbulent flow air outlets installed in the parapet wall of the assembly room. Smoke was then introduced into the air outlets which supplied the air with smoke into the room so as to visualize the airflow pattern of the supply over the deputies seating area. The smoke was delivered to the hall via 5 of the 17 outlets installed, so as to permit a better visualization of the air jet

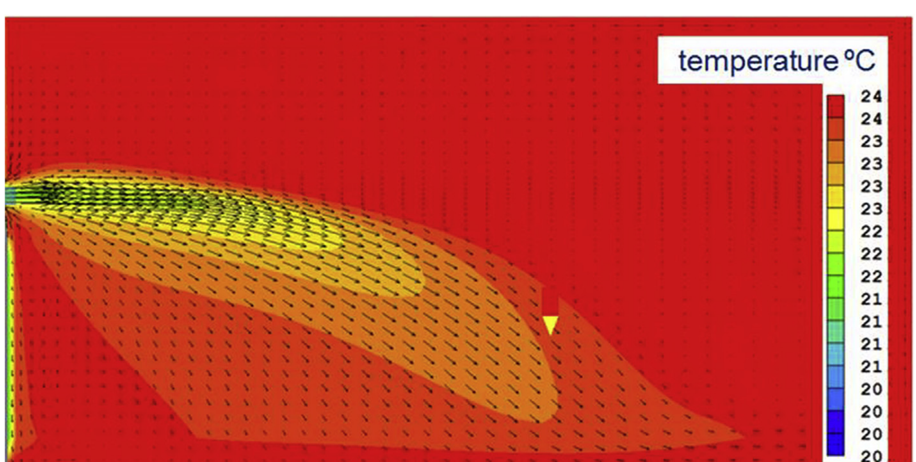

Fig. 5A. Whirl type diffuser outlet substitution model, measured air temperature field.

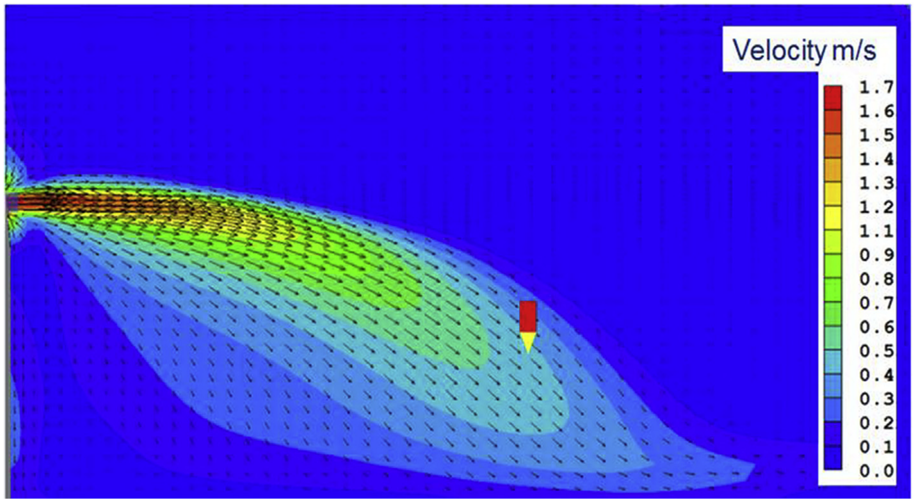

Fig. 5B. Whirl type diffuser outlet substitution model, measured air velocity field.

Table 1
Air velocity and temperature at measurement points.

Value measured	Laboratory test	Simulation model
Velocity, u (m/s)	0.47	0.48
Temperature, T (°C)	24.1	24.0

penetration and the drawback effect of the supply air returning back up the deputy seated area creating this way the displacement effect in the occupied zone.

3. Results and discussions

A summary of the results obtained in laboratory tests, substitution model, and CFD simulation and on site tests is presented below.

3.1. Laboratory test

Air temperatures and velocities at the aforementioned points were taken for each of the 16 tests, those corresponding to flow rates, V_0, of 347, 520, 694 and 868 m^3/h and temperature differentials, ΔT, of −10, −5, 0 and + 3 K.

The air leaves the diffuser in whirl shaped jets with a fixed discharge angle of approximately 12° in the upwards direction, thus creating a perfectly horizontal flow without the jet opening in the lower part. Due to the high rate of induction, admixing between the supply air and room air causes the air jet temperature to quickly approximate towards the room occupied temperature thus avoiding the dumping of cold air.

Since the mixture is better when the air flow rate is higher,as the lower the airflow rates the air jets tend to fall earlier. Thus, for a flow of 868 m^3/h and ΔT = −10 K, the air is seen to flow over the seats without falling(dumping)onto them, while with lower flow rates, the

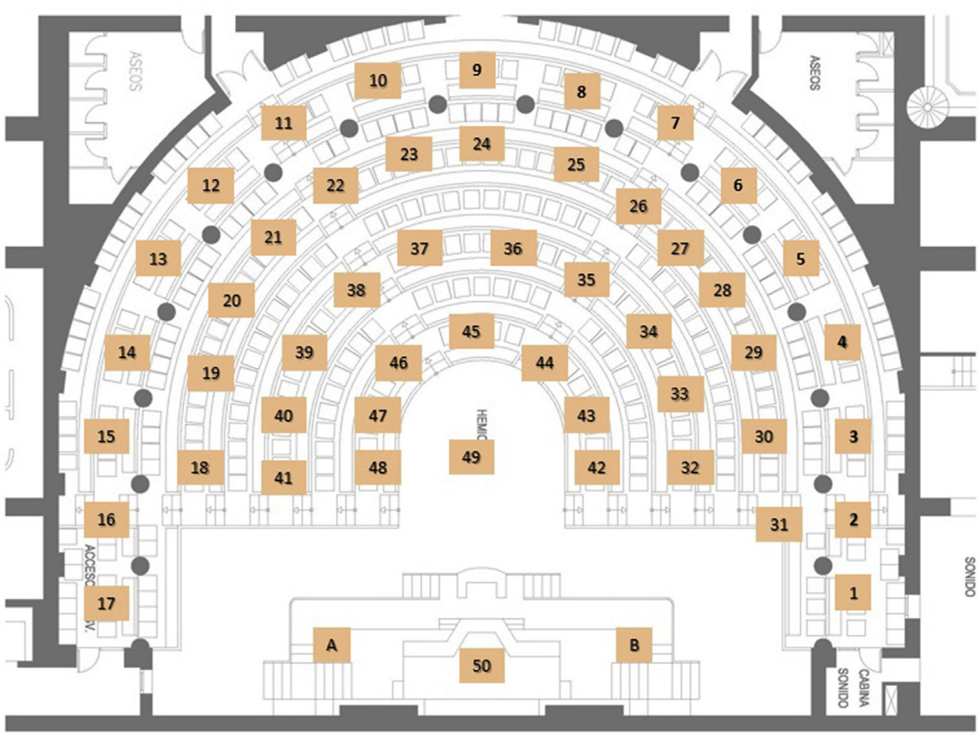

Fig. 6. Distribution of air temperature and velocity measurement points.

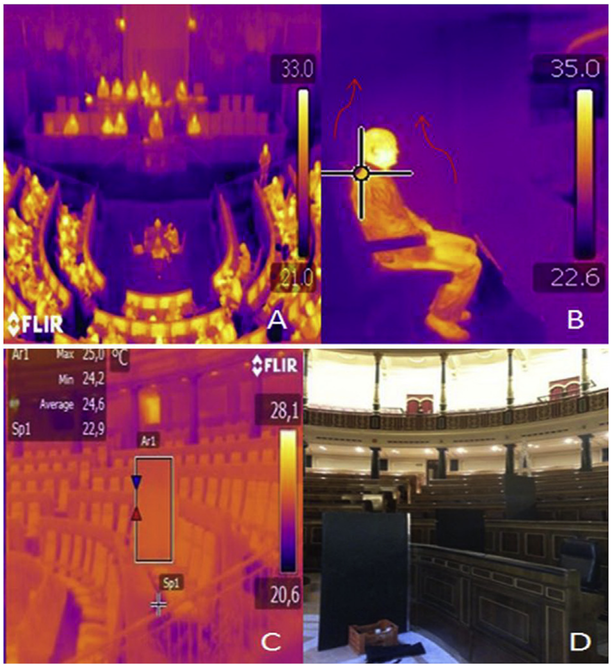

Fig. 7. A–DThermography imagining and analysis.

air jet penetrates into the occupied zone when reaching the last rows of seats. This also confirms that the existing tangential flow outlet grilles created dumping of the supply air into the occupied zone but with a much lower temperature mixture due to its tangential flow effect.

The performance of the air jet makes it clear that for $\Delta T = -10K$, higher supply flow rates are suitable. That is, when the cooling load increases, air flow should be increased to its optimum flow rate and only afterwards should the temperature differential be varied. Therefore, when the peak cooling load is achieved the flow rate gets to its maximum value and maximum temperature difference.

The results of indoor air velocities and temperatures show acceptable outcomes. Fully intermixed airflow and very low velocities in the occupied zone are achieved.

For the isothermal mode ($\Delta T = 0$ K) it was observed that the flow pattern is correct, sweeping the entire occupied zone (up to 14 m) with low velocities.

Numerical results and the jet pattern obtained for the heating mode, with $\Delta T = +3K$, show that, although air flows almost horizontally at an average height of 5 m, partial air mixing in the occupied zone is achieved.

3.2. Laboratory test substitution model

From the experimental results, the following recommendations for the CFD substitution model are derived;

When the cooling load increases, air flow should be increased to its optimum, and only afterwards should the temperature differential be varied. Therefore, only when the peak cooling load is achieved the flow rate gets to its maximum.

To determine the optimum air flow, computer simulations and or field tests have to be carried out.

When the manufacturer operational instructions are considered and when the system performance is adjusted as per computational analysis, the use of high turbulent effect whirl outlet diffusers results in uniform low velocities within the occupied area and achieves a temperature stratification in the vertical plane in the room.

3.3. CFD simulation

The CFD room airflow simulation shows the heat plumes in occupied zones. These plumes are verified in the thermography images shown in Fig. 7. Their evacuation into the vertical plane is induced by the horizontal high induction turbulent overhead airflow (Fig. 8A and B).

In the lower part of the images, in Fig. 8A and B, an effect similar to that of the displacement flow is observed. The supply air produces a displacement effect (not an admixture effect) in the occupied zone, and reaches a temperature between 20 and 22 °C at its end, a temperature similar to that used in the displacement systems with floor supply. These conditions make possible the recirculation of the air into the occupied area, ascending first slowly around the seats, neutralizing the thermal loads by the "plume" effect, and cooling again in contact with the driven air as it restarts the airflow cycle.

As it is shown in the Fig. 9, the air extraction of the whole room is done through the center of the hemicycle, out of the reach of the throw of the diffusers, mixing with the air of the upper zone as it rises up to the ceiling extract. The thermal load of the upper part of the assembly hall (mainly due to lighting) is eliminated exclusively by the return through the roof, where the air reaches its highest temperature, above 25 °C.

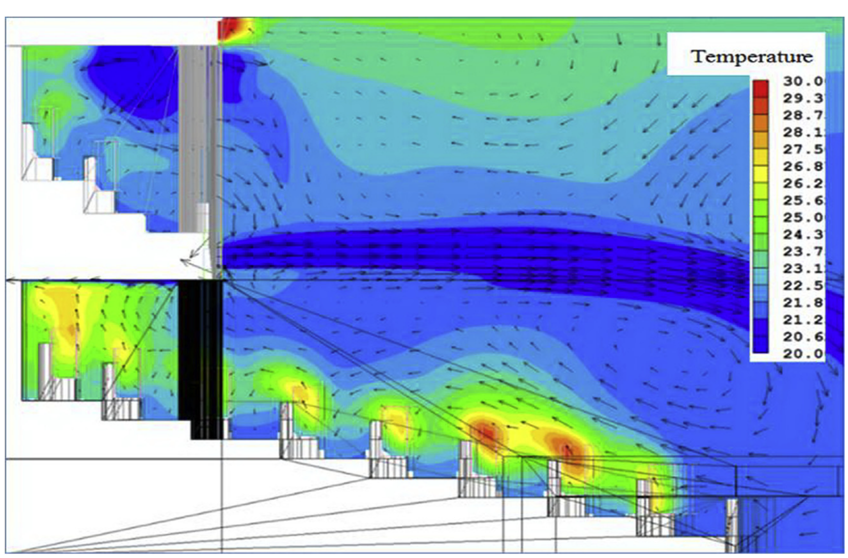

Fig. 8A. Air temperature patterns in the assembly hall.

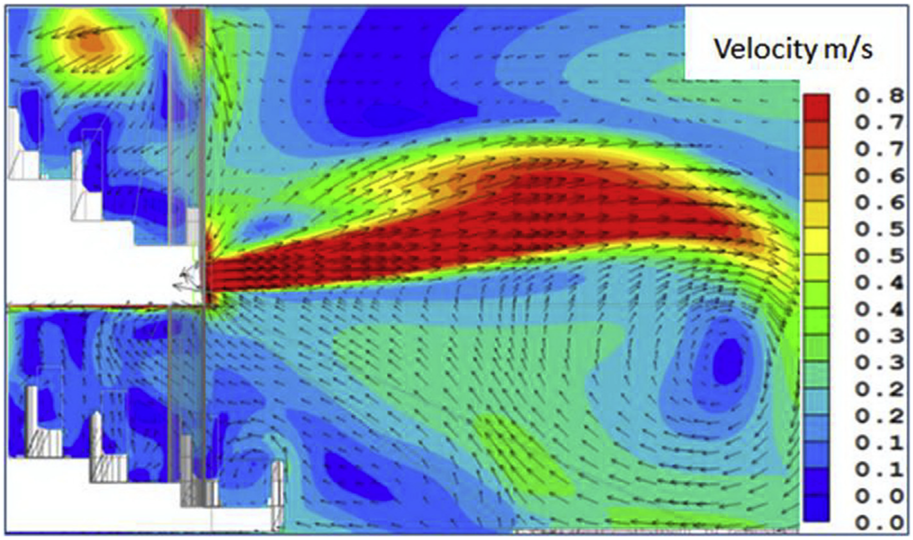

Fig. 8B. Air velocity patterns in the assembly hall.

The laboratory measured data shown in Fig. 5A and B validate the airflow pattern shown here in Fig. 9 where it can be seen, and represented as a temperature profile, that the supply air from the parapet mounted air outlets flows out over and above the occupied zone to be later and partially drawn back over the seated area in the lower part of the hall. The room air temperature shown in the lower part of the simulation in Fig. 9 does not exceed 25 °C which confirms the proper evacuation of the heat load from the seated area of the deputies.

3.4. Field tests

3.4.1. Measurement of air temperature and velocity

Table 2 shows the measurements for the critical supply air temperatures, ranged between 12 °C and 19 °C, included below. Distribution of velocities and temperatures in the range of 19 °C–28 °C were also tested, but they resulted in lower velocity values with similar spatial temperature differences and are therefore not included in this paper.

The supply air was set at 15 °C to determine the dumping effect of the colder supply air under minimum internal loads conditions. Also to determine the spatial homogeneity of the occupied zone temperature and it can be seen that a very constant average value is achieved implying that no cold air dumping occurs.

Note also that not all measurement points shown in Fig. 7 are included in the table as their values were constant with the next closest measured point. The indicated 33 points measured points shown in Table 2 are more than sufficient to determine an accurate mean average value of both temperature and velocity of the entire occupied space of the assembly hall.

3.4.2. Thermography analysis

Thermography imagining confirmed the plume effect next to the sensible heating loads of occupants and equipment, in the sitting area.

Using the thermography imagining, a uniform radiant temperature was observed throughout the entire occupied zone. This indicates also a very good mixing effect, due to the high induction ratio of the diffusers, as the air gets to sweep all the surfaces of the interior spaces and

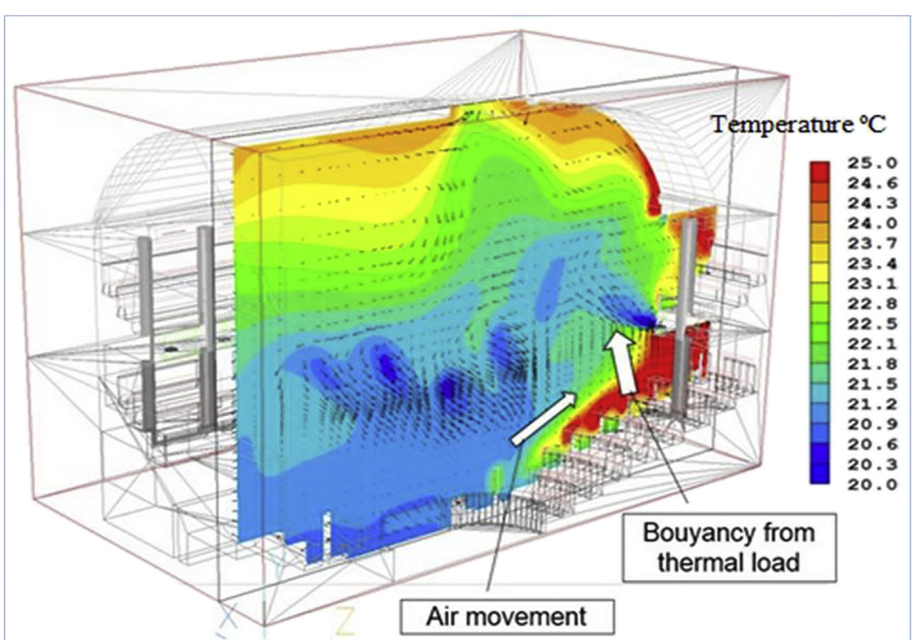

Fig. 9. CFD simulation of the room Temperature in the assembly hall.

Table 2
The air velocity and temperature values at the indicated measurement points, measured at a height of 1.2 m above the corresponding floor level.

Point	1	2	3	4	5	6	7	8	9	10	11
°C	21	19.9	20.5	20.2	19.9	20.2	20.4	20.5	20.2	19.9	19.7
m/s	0,09	0.04	0.06	0.04	0.04	0.06	0.06	0.04	0.05	0.06	0.06
Point	12	13	14	15	16	17	18	37	38	39	40
°C	19.8	19.9	19.2	18.8	18.6	18.9	19.1	18.4	18.6	18.2	18.4
m/s	0.04	0.05	0.04	0.03	0.04	0.05	0.03	0.06	0.07	0.06	0.05
Point	41	42	43	44	45	46	47	48	49	50	A
°C	18.9	18.5	18.6	18.4	18.3	17.9	17.7	17.9	17.6	18.9	20
m/s	0.04	0.05	0.04	0.05	0.07	08.8	0.07	0.08	0.13	0.12	0.14

The values sown in Table 2 correspond to when there was only lighting and external heat gains with a minimal occupation without any electronic equipment loads.

balances the heat exchange in every surface point of the occupied zone, avoiding temperature significant differences in the surfaces.

Finally, the thermographic analysis also verified the stability of the air jets and the flow patterns with supply temperatures ranging from 12 °C to 28 °C.

3.4.3. Smoke tests

The smoke tests confirmed that the visualized air distribution flow coincides with the air flow patterns of the laboratory tests and the computational fluid dynamics simulation (Fig. 10). In all cases, it was possible to confirm the evidence of an ascending effect of the return air that allows an immediate balance of the thermal loads. In addition, it was clearly shown a displacement effect in the floor area caused by the plume effect from the heat sources in the occupied area. In the centre height of the hemicycle and parallel to the floor an upwards confluent jet type flow is created, that forces the air in the occupied zone to be drawn (induced) back up the seating area, with the aforementioned displacement flow effect.

The red arrow in the upper part of the hall shows the heated air rising to the extract points in the ceiling. The purple and green pattern shows the outlet supply air penetrating into the occupied zone with a parallel flow over the seating area. The blue arrows represent the induced room-supply admixture ascending up the seated tribune area creating the displacement effect at the lower floor level.

3.4.4. Thermal comfort validation

Based on the measurements made during the aforementioned on-site tests, the degree of thermal comfort achieved has been determined, according to the UNE-EN-ISO-7730 procedure [35]. In this case, for all the analysed variables, a class A has been reached. A predicted

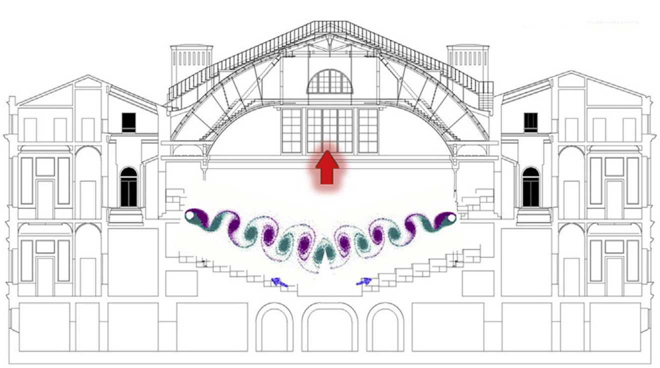

Fig. 10. Cross section of the hemicycle showing smoke patterns into and over the occupied zone.

percentage of dissatisfied (PPD) lower than 6% is achieved [36]. A summary of the results for local thermal comfort variables are shown in Table 3.

As mentioned and stipulated by the standard EN-ISO-7730, local thermal discomfort values are classified in 3 categories, A, B and C, where DR stands for draught rate; VATG for vertical air temperature gradient; FT for the floor temperature and RTA for radiant temperature asymmetry.

* Obtained with

$$DR = (34 - T_S)(v - 0.05)^{0.62}(0.37T_U v + 3.14)$$

where T_s is the average room dry bulb temperature (measured 25 °C), v is the average room air velocity (measured 0.14 m/s) and T_u is the average local turbulence intensity (measured 30%).

4. Conclusions

This paper distinguishes between what is the specific mode of neutralization of the thermal load by displacement airflow patterns and how it is achieved, arriving at the conclusion that, for the purposes of air diffusion, displacement ventilation is an effect that, in certain occasions, such as the one described in the article, is independent of the drive system used and the angle and plane of supply.

The results from this research confirm that in buildings with high ceilings, a displacement effect can be created in the occupied zone as well as in height by means of turbulent supply airflow with an overhead parallel jet flow. Consequently, it can be stated that it does not matter how the air reaches the occupied area, but what matters is the effect that occurs at ground level once it arrives.

In the design procedure, an effective method for the design and start-up of the air distribution system in complex buildings refurbishment is proposed; laboratory tests, CFD simulation with validation and verification of the computational analysis using comparative laboratory analysis to confirm the accuracy and convergence of iterative calculations and their tendencies, and finally on-site measurements where smoke tests and thermography analysis were carried out so as to complement each other in order to get a proper system solution that combines both thermal comfort and energy savings. The energy savings is achieved by creating the stratification effect similar to displacement diffusion technology, as this implies that a temperature gradient similar to displacement ventilation can be achieved between the supply air and room extract air.

The described system is not just an exclusive design for the Congress of Deputies building of Spain, its use can be extended to the refurbishment of any large assembly hall with tiered seating aisles such as conference rooms, cinemas, theatres, lecture halls, etc. The CFD models takes into account the interferences between confluent turbulent flow jets, and reveals the presence of unexpected effects that govern air movement and define how the loads are matched. The posterior field validation test confirms the results of the CFD analysis.

Given that an overhead confluent jet supply allows better for air movement above the occupant seated area the sensation of comfort will be superior as to that compared with a floor mounted supply with many physical obstructions which will impede a satisfactory air movement.

With the proposed system, draft problems caused by the original system have been solved. The system is sensitive not only to the thermal load as a whole but also to its spatial variation, which can range from 30 W/m^2 to 140 W/m^2 due to the difference of occupation density between areas and the use of electronic equipment. Under such conditions, mean air velocity and temperature vertical gradient in the occupied zone are lower than the stipulated standard limits. Thermography analysis also shows a uniform radiant asymmetry in the occupied zone. Accordingly, a percentage of predicted percentage of dissatisfied (PPD) lower than 6% is achieved, which corresponds to an A class, according to UNE-EN-ISO-7730.

More research is needed for the definition of the stratification effect

Table 3
Thermal comfort limits and measured values.

Class	PPD %	DR % *		VATG (°C/m)		RTA (°C)		FT(°C)	
		Limit	Measured	Limit	Measured	Limit	Measured	Limit	Measured
A	< 6	< 10	9.5	< 3	0.5	< 5	< 2	< 10 FT < 15	< 21 FT < 15
B	< 10	< 20		< 5		< 5		< 10 FT < 15	
C	< 15	< 30		< 10		< 10		< 15 FT < 15	

in high turbulent system flow when used in a horizontal plane. These studies should serve as a basis to redefine the ventilation system technology as is known today.

However it can be concluded that this case study shows an alternative design strategy for displacement effect in other similar type applications, both in general architectural design in tall spaces as well as similar type industrial applications.

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Appendix A. Supplementary data

Supplementary data to this article can be found online at https://doi.org/10.1016/j.buildenv.2019.106311.

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