Analysis of the influence of the return position in the vertical temperature gradient in displacement ventilation systems for large halls

Tomas Gil-Lopez^{a,*}, Miguel A. Galvez-Huerta^b, Paul G. O'Donohoe^c,
Juan Castejon-Navas^a, Pedro M. Dieguez-Elizondo^d

^a Madrid Polytechnic University, Avda. Juan de Herrera, 6, 28040 Madrid, Spain

^b Federico Santa María Technical University, Avenida España 1680, Valparaíso, Chile

^c D.I.T. Faculty of Building Services Engineering, Botton Street, Dublin, Ireland

^d Public University of Navarre; Campus de Arrosadía, 31006 Pamplona, Spain

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A B S T R A C T

Displacement ventilation systems are widely used in spaces of small and medium heights (up to 7.5 m), taking advantage of their high efficiency in removing contaminants by means of the air stratification. This outcome usually hides major drawbacks compared to admixture systems, such as inadequate temperature gradient in the occupied zone and low cooling capacity. On the contrary, its use in large spaces with high ceilings displays the best features of the system, provided the return is placed near the ceiling. When, due to architectural reasons, this option is not feasible, alternatives for the return position are assessed using simulation tools that result in excessive time consumption during the design stage. In this article a simple Mundt's equation based iterative process is proposed to quickly evaluate the influence of the return height in the vertical temperature gradient and the additional cooling load due to overheated ceiling. It has been applied to the case of Madrid airport new terminal, and their results have been compared to a CFD simulation, with a reasonable degree of accuracy for an initial stage of a design process. The results show that the higher the return is placed, the lesser airflow rate is needed to match the additional cooling load due to radiation from the ceiling. The calculation process also shows that Mundt's radiant heat exchange coefficient is far from being constant, for it is affected by the height of the return point, and takes values much greater than the usually accepted.

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1. Introduction

Mixing-type diffusion systems have traditionally dominated air conditioning of large spaces [1]. The major cause for this prevalence is the capacity to offset high cooling loads while maintaining uniform hygrothermal conditions in the occupied area with relatively low air flow rates [2]. Its main disadvantage, however, is the low air quality obtained [3].

Displacement ventilation systems, whose origins are related to the high demanding ventilation needs in industrial uses [4], are the most efficient way to achieve good air quality due to its high performance when eliminating sources of pollution in the occupied area. By means of a high thermal stratification, a vertical gradient

of concentration of contaminants is produced, which allows the system to easily remove pollutants with a return outlet near the ceiling [5].

Nevertheless, when a displacement ventilation system is used in rooms with great ceiling heights, overheating of the roof and stagnation of warm air masses near the ceiling occurs. This situation is worsened when, due to architectural decisions, air cannot be exhausted near the ceiling. In this case, if a direct discharge of stagnated air through skylights is forced, air currents due to infiltration through doors and openings are generated, and a mismatch of environmental conditions in the occupied zone occurs, which prevents the displacement diffusers to function properly.

If the return outlets are placed at low height, the main consequences are:

- Vertical temperature gradient in the occupied zone is greater than the acceptable [6].

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* Corresponding author.

E-mail address: tomas.gill@upm.es (T. Gil-Lopez).

Nomenclature

Φ_s	Specific sensible cooling load (W/m ²)
T_s	Supply air temperature (°C)
T_r	Return air temperature (°C)
T_i	Indoor air temperature (°C)
Q_s	Specific supply air flow rate (m ³ /h m ²)
$\alpha_{c,f}$	Convective heat exchange between air and floor (–)
$\alpha_{r,c}$	Radiant heat exchange between air and ceiling (–)
h_i	Surface heat transfer coefficient between floor and air (–)
H_R	Return point height (m)
H_c	Ceiling height (m)
TVG	Temperature vertical gradient (°C/m)
PPD	Predicted percentage of dissatisfied (%)
Θ	Dimensionless temperature

- The air is returned with lower temperature, which implies a reduction of the cooling capacity of the system.

This drawback can be solved by increasing the air flow rate, which leads to excessive discharge velocities and disturbing currents in the occupied zone. In contrast, when using lower supply air temperatures, in addition to the increase in energy consumption, the temperature of the occupied area exceeds the comfort conditions limits.

Optimum return position is a design parameter for which no references are found neither in design handbooks nor in scientific literature, which usually refer to lower ceiling heights [7] or assume that return is placed at ceiling level. Though other important items are also developed in this article, this circumstance alone would justify the following study.

In this work it is analyzed the influence of the return height of this element on the thermal parameters of the system, paying special attention to the vertical temperature gradient, as well as, due to its influence in the energy performance of the system, the airflow rate needed in each case to match the cooling loads [8,9]. To carry out the research, a simplified iterative calculation process has been used, and the results compared to the obtained with a CFD simulation. This method has been applied to a representative

new building such as the Barajas airport new terminal, by Estudio Lamela and Richard Rogers Partnership [10].

2. Material and methods

The satellite building of the new Barajas airport terminal is a rectangular building several hundred meters long with a constant cross section of 37 m width [11]. The ground floor level is occupied by technical systems. The roof has a characteristic wave like shape in section, and is placed at an average height of 18 m, ranging from 15 to 23 m (Fig. 1).

Air conditioning of lounge areas is provided by the displacement ventilation system that is analyzed throughout this work, supplemented with jet nozzles diffusers all along the East and West glass facades. This solution allows to offset external heating and cooling loads, as well as to eliminate cold drafts down the facade and the risk of condensation in glazed surfaces and thermal bridges. It also creates an effective climate separation area that makes internal zone independent from the perimeter areas.

As architectural decisions prevented return air ducts to be visible along the ceiling, and in order to avoid the aforementioned disadvantages of so high a ceiling, mainly radiant asymmetry and warm air stagnation near the ceiling, an “ad-hoc” solution of a vertical totem was proposed, as it is shown in Figs. 1 and 2.

This element enabled to place the return outlets at its upper point keeping the ducts from being visible at all, and to cool the return air at this point by means of a set of five air nozzles placed in the upper part of the element (Fig. 3). With this solution, warm air stagnation near the ceiling and roof overheating was conveniently avoided. Due to geometry problems, a return point height of 13 m was finally adopted.

2.1. Usual design criteria for displacement ventilation systems

When the emission of pollutants is uniformly distributed, as in rooms with high occupancy, the system design is based on the control of vertical temperature gradient, in order to maintain the comfort design parameters in the occupied area [12], namely air temperature at 1.80 m height [13], maximum air velocity [14], radiant asymmetry with the walls [15] and temperature difference between head and feet [16], within specified limits.

Fig. 1. Cross section showing air diffusion systems.

Fig. 2. View of a displacement diffuser, with a totem in the background.

2.1.1. Design method

The process, according to Skistad [17], can be resumed as follows:

a) The variables described next are initially known:

- Specific sensible cooling load, Φ_s (W/m²). Usual values for interior zones range from 40 to 60 W/m² [18,19].
- Supply air temperature, T_s . Due to the fact that air is thrown directly to the occupied zone, air is only cooled to a temperature 4 °C lower than indoor air conditions. This leads to usual supply air temperatures of 19 °C.
- Specific supply air flow rate, Q_s (m³/h m²). It is related to the difference of temperatures between the exhaust and the supply air, ΔT , by means of the specific thermal load. Whether this parameter is restricted to 4 °C or is increased to meet greater thermal loads, the resulting flow can lead to excessive diffuser sizes, problems of noise and inappropriate air velocities. In any case, it always has to be higher than the air flow needed for sanitary ventilation [20]

In most cases, a fixed $\Delta T = 12$ °C is considered. Accordingly, return air temperature is usually limited to 31 °C. Theoretically, due to the stratification, the higher the return outlets are placed, the greater the difference of temperatures achieved, and the lower the air flow needed to match the thermal loads. This is an option worthy of attention for ceiling heights over 15 m. But overheating of the ceiling prevents it from succeeding.

Fig. 3. Plan and front view of the air nozzles and return outlet in the upper part of the totem.

b) Resolution of air temperature value at 0.1 m height

$$T_{0.1} = T_s + \theta_{0.1} (T_R - T_s) \quad (1)$$

where $\theta_{0.1}$, dimensionless temperature, is obtained with the Mundt equation [21]:

$$\theta_{0.1} = \frac{1}{Q_s c \rho \left(\frac{1}{\alpha_{c,f}} + \frac{1}{\alpha_{r,c}} \right) + 1} \quad (2)$$

in which $\alpha_{c,f}$ is the convective heat exchange between air and floor and $\alpha_{r,c}$ the radiant heat exchange between air and ceiling. They usually adopt values of 4.5 W/m² and 5 W/m², respectively [22–24].

c) Calculation of floor temperature, T_f

$$T_f = T_{0.1} + \alpha_{c,f} / h_i \quad (3)$$

with h_i , surface heat transfer coefficient between floor and air. For an ascendant heat flow with air at low velocity, it can be considered $1/h_i = 0.10 \text{ m}^2 \text{ K/W}$

d) Assuming that the temperature gradient in the room is constant, at least until the position where the return outlet is placed, air temperature at 1.8 m height, $T_{1.8}$, is obtained:

$$T_{1.8} = T_{0.1} + 1.7 \cdot \frac{T_R - T_{0.1}}{H_R - 0.1} \quad (4)$$

as well as the corresponding temperature gradient:

$$\text{grad } T = \frac{T_{1.8} - T_{0.1}}{1.7} \quad (5)$$

With the obtained values, accomplishment of normative comfort conditions is tested, paying special attention to the temperature difference between head and feet [25].

e) As it has been mentioned before, this vertical gradient applies for the lower zone of the room, under the return grilles. Over this height, and up to the ceiling, natural stratification occurs due to heat exchange with walls, ceiling and lighting devices. In this case, thermal gradient depends on many variables, namely outdoors ambient temperature, roof insulation level and air diffusion mode [26], and no general law to describe the phenomenon has been found in the state of the art review. Authors like Obersby [27,28] have addressed this problem for low ceiling

heights and heating mode. Wang et al. [29] have studied stratification when cooling underfloor distribution systems are used. Recently, Awad et al. [30] have dealt with the vertical temperature gradient with displacement ventilation, but for pollutant removal purposes only. Porras et al. [31] have measured, under summer conditions, the thermal gradient in an air-conditioned warehouse with similar dimensions (32 m long, 26 m wide and 8 m high) to the building studied in the article. So, in the absence of references, in this work a fixed vertical temperature gradient of 1.8 °C/m has been adopted for the upper zone of the room. This value corresponds to the occupied zone gradient obtained when the return is placed 4 m high, which is the most unfavourable solution. As it is stated later, it has been confirmed with the results obtained with the CFD simulation.

In summary, for high ceiling spaces with displacement or under-floor air distribution systems, two zones can be distinguished: the occupied zone, from 0.1 m up to 2.00 m high, governed by the Mundt's gradient produced by the plume effect of the thermal loads; and the unoccupied zone which, in turn, is divided in two subzones by the horizontal plane where the return is placed. At this point, the design temperature that matches the thermal loads is set constant to 31 °C. From here and up to the ceiling, the HVAC system is supposed to have no effect, and the air stratification is governed by the heat exchange with the roof. The division of the room into zones is shown in Fig. 4.

2.1.2. Objections to the method

With respect to the radiant heat exchange between air and ceiling, the value of $\alpha_{r,c}$ is far from being constant. It depends on the ceiling height and the position of thermal loads such as lighting devices or skylights. In this case, its value is much greater than those usually accepted.

In order to determine it, they have to be considered the thermal effect on the floor due to:

- The long wave radiation from the roof.
- The short wave radiation through the skylights [32].
- The short wave radiation from the lighting devices [33].

It is obtained, from the initially unknown floor and ceiling temperatures, by means of an iterative process.

Furthermore, when the outside part of the ceiling is exposed to direct radiation, warm air stagnation in the upper part of the space results in an additional thermal load not considered in the initial

Fig. 4. Division into zones of room height for displacement ventilation systems in spaces with high ceilings.

Table 1
Initial data.

Supply air temperature, T_s	19 °C
Unitary cooling load	77.90 W/m ²
Return air temperature, T_r	31 °C
Indoor air temperature, T_i	23 °C
Ceiling height, H_c	17.50 m
Return point height, H_r	Variable (ranging from 4 up to 13 m)
Supply air flow rate, Q_s	19.19 m ³ /h m ²

calculations that can finally end in a mismatch of the thermal load by the projected system. This makes in some ways necessary to limit the ceiling temperature in such HVAC applications. In order to do it, a possible solution is to cool down the return air temperature to the expected 31 °C.

Finally, in the case of more than 7 m ceiling heights, special attention must be paid to the point H_R at which the return is placed, for it determines the vertical temperature gradient and, eventually, the ceiling temperature. Evidence of this influence is shown in the following section.

2.2. Proposed method for high ceiling cases

The solution used in Madrid-Barajas airport has been analyzed for different return positions. Heights of return of 4, 7, 10 and 13 m have been considered.

As an example, it is only developed a numerical application for the case of a return height of 13 m, with an air flow rate of 19.19 m³/h m² which derives from the existing building design conditions (Table 1). An extension of the procedure to greater return heights can be easily carried out.

In order to obtain the thermal load due to the ceiling temperature, an iterative procedure is used [34], which leads to α_r values different to those usually accepted. It consists in running the algorithm with an initial value equal to the usually adopted (step 1). Once the corresponding floor and ceiling temperatures are obtained with Mundt's expression, the energy balance equation between these surfaces is set out. From it, radiant flux from ceiling to floor is obtained. This flux, supplemented with the otherwise constant heat exchanges from skylights, Φ_{skylight} , and luminaires, Φ_{lighting} , is used as the α_r value within the next iterative step. After no more than 3 steps, results become stabilized.

The intermediate results of the iterative process used to obtain the characteristic thermal parameters of the problem are shown in Tables 2 and 3.

As it has been previously mentioned, α_r value of 35.53 W/m² is considerable higher than those usually adopted.

It is also worth mentioning that the radiant exchange from the ceiling, which amounts to 26.50 W/m², and means an additional load not considered in the initial calculations. It finally results in an increase of air flow of 6.53 m³/h m² to match the thermal load with the adopted $\Delta T = 12$ °C. If this air were supplied by the displacement diffusion system, it inevitably would worsen the vertical temperature gradient and would question the suitability of this system in such cases.

This means not only that α_r value, even in the best possible case, is greater than the usually accepted, and that a general process to calculate it has been established, but also that it is this coefficient which allows the designer to size the admixture ventilation

Table 2
Iterative calculations for determining Mundt's radiant coefficient.

Radiation source	Units	Data	Step 1	Step 2	Step 3
Φ_{skylight}	W/m ²				4.62
Short wave radiation	W/m ²	11.85			
View factor FF_{sklf}	(-)	0.39			
Φ_{lighting}	W/m ²				4.41
Total thermal load	W/m ²	18			
Thermal load on floor or ceiling	W/m ²	9			
$\epsilon_{\text{lighting}}$	(-)	0.72			
View factor FF_{tlf}	(-)	0.46			
$\Phi_{\text{lighting to ceiling}}$	W/m ²	2.98			
View factor FF_{tlf}	(-)	0.68			
Φ_{ceiling}	W/m ²		22.99	26.44	26.50
$\epsilon_{\text{ceiling}}$	(-)	0.60			
View factor FF_{clf}	(-)	0.39			
T_{ceiling}	°C		310.40	314.02	314.05
T_{floor}	°C		294.77	296.51	296.52
$\alpha_r = \Phi_{\text{ceiling}} + \Phi_{\text{skylights}} + \Phi_{\text{lighting}}$	W/m ²	5.50	32.01	35.47	35.53

Table 3
Iterative calculations of the characteristic thermal parameters.

Thermal parameters	Units	Step 1	Step 2	Step 3	Step 4
α_r	W/m ²	5.50	32.01	35.47	35.53
θ_{mundt}	(-)	0.270	0.371	0.375	0.376
$T_{0.1}$	°C	22.23	23.45	23.50	24.52
T_{floor}	°C	21.77	23.45	23.50	24.52
$T_{1.8}$	°C	23.39	24.45	24.49	24.50
gradT	°C/m	0.68	0.58	0.58	0.58
T_{ceiling}	°C	37.40	40.46	40.90	41.04

system needed to cool down the return air. Due to the fact that a mixing system thermally conditions the whole volume of air of the premises while only its lower part is occupied, this circumstance makes it economically unfeasible. On the other hand, it has previously been said that displacement ventilation systems by no means solve the problem, for they worsen the hygrothermal conditions in the occupied area while they entail a significant energy consumption increase. Therefore it can be concluded that, in this type of building, coexistence of air diffusion systems is essential to simultaneously achieve hygrothermal comfort, air quality and energy efficiency [35–37].

2.3. CFD simulation and validation by means of experimental data

To confirm the results obtained with the iterative method, the departures lounge of the Madrid airport new terminal has been modelled by means of a CFD simulation, based upon the experience in the use of this methodology acquired by the authors in previous research works [12,38,39].

To carry out the simulation, ANSYS-CFX software was used, approximating the geometry of the building with a polyhedral mesh. To reduce calculation time consumption, the following simplifications to the model have been assumed:

- Due to the existence of a longitudinal axis of symmetry, only half of the lounge has been modelled.

Table 4
Supply air diffusers characterization used in CFD simulation.

Supply air diffusers	Number of units	Height (m)	Air flow (m ³ /h)	Specific air flow (m ³ /h m ²)	Supply temperature (°C)
Displacement diffusers	4	0.00	13,760	19.19	19
Totem jet nozzles	1	13.00	4445	6.19	19
Total			18,205		

Fig. 5. (a) Temperature variation along the longitudinal axis of symmetry for a return height of 13 m. (b) Temperature variation in the cross section for a return height of 13 m.

Fig. 6. Temperature profiles for CFD simulation and experimental data, for a return height of 13 m.

- As the pattern of the air supply units (displacement diffusers and jet nozzles) position is repeated along the whole building, only a module between two consecutive totems has been analysed.

With these assumptions, the geometry of the simulated volume has been discretized in a mesh of approximately 6.3×10^6 elements

(with slight variations depending on the totem height hypothesis). This implies an average size per element of nearly 1 dm^3 that guarantees robust enough results. Due the low speeds, laminar flow has been considered. The code simultaneously solves the Navier-Stokes equations and the energy equation, after 10,000 iterations.

Fig. 7. (a) Vertical temperature profiles with the iterative method for different return heights. (b) Vertical temperature profiles with the CFD simulation for different return heights.

Table 5
Summary of cooling loads used in the CFD simulation.

Item	Load (W)	Specific load (W/m ²)
Envelope (roof and floor)	11,250	15.7
Occupants	10,045	14.0
Appliances	11,520	16.1
Lighting	23,045	32.1
Total	55,860	77.9

The main characteristics of the supply air units used in the simulation are shown in Table 4.

In the same way, Table 5 summarises the cooling loads used in the CFD simulation, distributed by source. Envelope loads due to walls and fenestration have been neglected, as they are matched by the supplementary system of jet nozzles described at the beginning of Section 2.

Fig. 5a and b show the air temperature distribution within the longitudinal and cross sections of the analysed module, being 13 m the return height. They have been obtained by means of a nonlinear calculation. Similar calculations have been conducted for return heights of 4, 7 and 10 m.

In order to validate the CFD simulation, a set of experimental data have been used. A Krantz VAZK displacement diffuser (630 mm diameter, 1480 mm height) was tested in the company laboratories for the cooling mode design airflow rate. Return grille was placed at a height of 9.5 m, due to the limited laboratory ceiling high (10.5 m), which made unfeasible to test higher return position options.

This experimental test has confirmed the accordance of temperature values for the occupied zone. Additionally, measurements of return air temperature also confirmed the 31 °C design value in this point. Although no air temperature measurement have been taken from here up to the ceiling, experimental data for the ceiling temperature have been eventually obtained by means of thermography images taken in the occupied building under summer conditions.

With the purpose of validating the accuracy of the CFD simulation with the aforementioned experimental data, the vertical temperature profiles obtained with both methods have been compared in Fig. 6. This figure shows that temperature differences between experimental measurements and CDF simulation are always lower than 1.3 °C. This confirms the accuracy of the simulation tool, and supports its use for the purposes of this research.

3. Results and discussion

Using the aforementioned method, a comparison of the temperature profiles has been obtained for different return heights, H_R , namely 4, 7, 10 and 13 m (Fig. 7a and b).

From the analysis of the results it is observed an agreement of the ceiling temperatures, and that temperatures trend in the upper side of the non-occupied zone agrees as well. Nevertheless, mayor differences exist in the occupied zone, which can be due to inaccuracies in the CFD simulation of the interaction of the floor and the diffuser air flow pattern. To the purpose of this study, which is to offer a tool to take quick design decisions in the early stages of a project, it can be said that the iterative Mundt's equation based method provides a good approximation, and the calculation process is much less time consuming than numerical techniques like the finite elements method.

For other thermal comfort parameters such as TVG (temperature vertical gradient) and PPD (predicted percentage of dissatisfied) [40], as well as for the mixing ventilation air flow requirements, Table 6 can be consulted.

It can be observed that for a given return height, comfort parameters like air temperature at the head ($T_{1.8}$) and temperature vertical gradient (TVG) decrease as return point is placed higher. This leads to recommend a return position as high as possible in order to maintain hygrothermal comfort conditions in the occupied zone.

Table 6
Thermal parameters for different return heights.

	Return height (m)			
	4	7	10	13
Q displacement (m ³ /h m ²)	19.19			
T _{supply} (°C)	19.00			
T _{return} (°C)	31.00			
T _{floor} (°C)	23.80	23.77	23.66	23.52
T _{1.8} (°C)	26.94	25.55	24.92	24.51
T _{ceiling} (°C)	65.21	56.86	48.82	41.05
α _r (W/m ²)	79.96	63.23	48.58	35.53
α _{rcelling} (W/m ²)	70.63	54.20	39.55	26.50
Q _{mix} (m ³ /h m ²)	17.42	13.36	9.75	6.53
Q _{total} (m ³ /h m ²)	36.61	32.55	28.94	25.72
TVG (°C/m)	1.85	1.05	0.74	0.58
PPD (%)	2.52	0.86	0.65	0.56

The minimum value obtained for the radiant heat exchange coefficient of Mundt's equation is 35.53 W/m², almost seven times greater than the accepted values. As it has to be calculated independently for each return point height, it is far from being a constant, as it is usually considered.

If the return air is not cooled, air temperature close to the ceiling results in an additional cooling load that has to be neutralized by means of an increase of air flow rate, which takes greater values the lower the return is placed.

In the case of Barajas airport, with a return height of 13 m, a supplementary complementary mixing diffusion system was adopted to match the 26.50 W/m² additional cooling load. The resultant 6.53 m³/h m² air flow rate was used to size the nozzles and to adequately space the totems.

In this case, raising the return from 4 m up to 13 m represents a 62.5% reduction of the air flow rate supplied by the mixing system. This reduction becomes of 29.7% if the total air flow rate (by means of both mixing and displacement systems) is considered.

Finally, it could be thought that decisions concerning ΔT (by means of either varying T_{supply} or T_{return}) would lead to different results. Using the aforementioned model, it has been simulated the performance of the system for different air flow rates. Figs. 8 and 9 show that temperatures in the occupied zone are sensitive to these changes, as expected. Nevertheless, for ceiling temperatures (Fig. 10) differences are negligible.

4. Conclusions

This research has been confirmed the good performance of the displacement ventilation systems for large spaces air conditioning. With these systems, optimal hygrothermal conditions in the occupied zone, mainly vertical temperature gradient, are obtained when the return is carried out through the ceiling.

When, due to architectural reasons, returning through the ceiling is unfeasible, stagnation of air in the upper zone occurs, and results in an additional thermal load, increases the air volume needed and can even make inadequate the displacement ventilation system. This situation worsens if the roof is in contact with the outside air and exposed to sunlight direct radiation.

A solution to this problem, that minimizes the total volume of air moved by the system and thus can be considered as the highest energy efficient, is to cool the return air near the ceiling, with an additional mixing ventilation system.

In this type of applications, to size the HVAC system, corrections in the Mundt equation must be made. Its radiation coefficient, $\alpha_{r,c}$, has to be calculated for each specific case by an iterative procedure, and takes values much higher than usual. This coefficient is also used to size the complementary mixing diffusion system.

Fig. 8. Floor temperature dependence on supply air temperature for different return heights.

Fig. 9. 1.80 m high temperature dependence on supply air temperature for different return heights.

Fig. 10. Ceiling temperature dependence on supply air temperature for different return heights.

A numerical application to the case of Barajas airport new terminal has been made. A full scale test was made for the occasion that showed the validity of the theoretical results and led to the finally executed solution. In this case, raising the return from 4 m up to 13 m represents a 62.5% reduction of the air flow rate supplied by the mixing system. This reduction becomes of 29.7% if the total air flow rate is considered.

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