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Laboratory testing of a displacement ventilation diffuser for underfloor air distribution systems

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ABSTRACT

Underfloor air distribution (UFAD) systems use the underfloor plenum beneath a raised floor to provide conditioned air through floor-mounted diffusers, which typically discharge cool air with both horizontal and vertical momentum components. These systems usually create a vertical temperature stratification when in cooling mode and this has an impact on energy, indoor air quality and thermal comfort. The purpose of this study was to characterize the stratification performance of a previously unstudied type of floor diffuser that discharges air horizontally, with almost no vertical velocity component, and that aims to combine the benefits of both UFAD and displacement ventilation (DV) strategies.

We performed 19 full scale laboratory experiments in which we varied the number of diffusers and the internal loads over a range of values typically found in office spaces. We quantified the amount of thermal stratification by measuring the dimensionless temperature at ankle height and found a degree of stratification that is typical of DV systems – higher than is typical in UFAD systems. We developed a model based on these results that can be used to simulate these systems in whole building energy simulation tools, such as EnergyPlus, and simplified UFAD design tools.

Keywords: Underfloor Air Distribution (UFAD); Displacement ventilation; Air temperature stratification; Diffuser; Thermal comfort; Laboratory full-scale testing, Thermal decay.

Symbol	Description
T	Temperature [°C]
Φ	Dimensionless temperature [-]
Γ	Gamma, a non-dimensional parameter representing the ratio of buoyancy to vertical momentum forces in the zone [-]
Ar	Archimedes number, a non-dimensional parameter representing the ratio of buoyancy to momentum forces in the zone [-]
A_d	Experimentally determined effective area [m^2]
A_c	Floor area of the chamber [m^2]
β	Volume expansion coefficient [K^{-1}]
H_c	Height of the chamber [m]
Θ	Diffuser discharge angle, measured from vertical [°]
N	Number of open diffusers [-]
M	Number of thermal plumes (i.e. heat sources) [-]
W	Zone cooling load [W]
Q	Airflow rate [l/s]
Subscripts	
P	Measurement at the exit of the duct as the air enters the supply plenum (supply air)
S	Measurement at the discharge of the diffuser (the room supply), averaged across all open diffusers.
F	Measurement at the upper surface of the concrete raised floor
0.1, 0.4, …, 2.2, 2.95	Measurement in the room at the height noted by the subscript, in meters.
OZ	Represents the occupied zone, an average of measurements at 0.1 m, 0.6 m, 1.1 m & 1.7 m
R	Measurement at the entrance to the ceiling-level exhaust duct as the air leaves the room (return air)

1   INTRODUCTION

Underfloor air distribution (UFAD) is an innovative method of providing space conditioning and ventilation to

buildings. UFAD systems use an underfloor supply plenum located between the structural concrete slab and a

raised access floor system to supply conditioned air through floor diffusers directly into the occupied zone [1].

UFAD and displacement ventilation (DV) systems are based on many of the same principles in cooling operation as they both deliver cool air into the room at or near floor level and return it at or near ceiling level. Thermal plumes that develop over heat sources in the room play a major role in driving the overall floor-to-ceiling air motion by entraining air from the surrounding space and drawing it upward. Properly controlled UFAD and DV systems in cooling mode produce temperature stratification in the conditioned space resulting in higher temperatures at ceiling level and cooler conditions in the occupied zone.

The primary difference between UFAD and DV systems is in the manner by which they supply air to the space. In the classic definition of a DV system, which is applied only for cooling purposes, air is supplied at very low velocity, thereby limiting the amount of mixing, through larger area diffusers often located in low side-wall positions. With this arrangement, finding enough available wall space for diffusers can be a challenge, particularly in open plan office settings. For UFAD systems, (1) air is supplied at higher velocity through smaller-sized floor diffusers, creating greater mixing, and (2) local air supply conditions are under the control of the nearby occupant by adjusting the UFAD floor diffuser. By using a raised access floor system to serve as the supply plenum, the entire floor surface area is available for placing supply diffusers, allowing great flexibility.

Stratified air distribution systems (DV and UFAD) are known to provide improved ventilation efficiency (increased fresh air in the breathing zone) when operated as 100% outdoor air systems, and also to provide improved contaminant removal efficiencies when contaminants are associated with heat sources. ASHRAE Research Project RP-1373 found air distribution effectiveness values for DV systems and UFAD systems using low throw height diffusers that were higher than those for conventional mixing systems and UFAD systems using diffusers with higher throw heights [2,3]. These research findings have been incorporated into ASHRAE Standard 62.1-2013 [1][1], which assigns a value of 1.2 for air distribution effectiveness for DV and UFAD with low throw diffusers in cooling mode, thereby allowing a potential reduction in the required minimum outside air rates to the space.

Further evidence of the ventilation efficiency benefits of stratified systems is provided by a laboratory experiment reported by Jung and Zeller [4] that compared the stratification and air change effectiveness (ACE) performance of

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[1] ASHRAE 62.1 - 2013 defines low throw slightly differently from those studies – it is defined as where the air velocity from the supply jet decays to less than 0.25 m/s (50 fpm) at a height of 1.4 m above the floor [1].

DV, UFAD, and overhead mixing systems. For the 100% outside air conditions of these tests, local ACE for the UFAD

system ranged from 1.2 – 2.0 with average ACE values of 1.2 – 1.3. Some of the local ACE values for the UFAD

system were even higher than the corresponding local ACE values for the DV system, a surprising finding. These

findings are relevant to the current study because the smaller-sized UFAD floor diffusers delivered only about 9.4

l/s with a vertical throw height of about 1.1 m, a value that is lower than typical throw heights (1.2-1.8 m) for most

UFAD swirl diffusers being installed today. This resulted in a rather dense diffuser layout with just about any point

in the test room being quite close to a nearby diffuser. In fact, it is this distribution of supply diffusers across the

floor that proves to be an advantage for UFAD compared to DV, which has its supply outlets located along the base

of one end wall of the test room. These findings suggest that UFAD systems can be configured to achieve

ventilation performance comparable to pure DV systems using a larger number of diffusers, diffusers that deliver

air with less mixing (lower throw height), or both. The DV diffusers of the current study match these criteria. The

most similar previous study that we identified

2   FUNDAMENTALS OF STRATIFICATION IN DV AND UFAD SYSTEMS

We describe thermal stratification in the zone using a dimensionless temperature, Φ_H (Phi), at a height 'H' in the

room, defined by:

$$\Phi_H = \frac{T_H - T_S}{T_R - T_S}$$ **Equation 1**

Where T_H, is the air temperature measured at a particular height 'H' in the room; T_S is the air temperature

measured at the diffuser discharge (the supply air); and T_R, is the air temperature measured at the room exhaust

(the return air temperature, typically located at ceiling level. Thus, lower values of Φ in the occupied zone (from

floor to 1.7 m height) indicate increasing stratification – that the air temperature at that height is lower relative to

the temperature at the exhaust. For example, a value of $\Phi_{0.1}$ = 0.2 indicates that a space is highly stratified. Higher

values of Φ in the occupied zone indicate decreasing stratification. For example a value of $\Phi_{0.1}$ = 1 indicates that a

space is not at all stratified, and that the air is fully mixed.

In this paper, we also calculate the average temperature in the occupied zone, T_{OZ}, according to:

$$T_{OZ} = \frac{T_{0.1} + T_{0.6} + T_{1.1} + T_{1.7}}{4}$$
Equation 2

Which is then used to calculate the dimensionless stratification in the occupied zone, Φ_{OZ}, according to:

$$\Phi_{OZ} = \frac{T_{OZ} - T_S}{T_R - T_S}$$
Equation 3

For displacement ventilation systems according to Chen and Glicksman [5] $\Phi_{0.1}$ varies between 0.2 and 0.7 and according to Skistad et.al between 0.3 and 0.7 [6]. Mundt [7] developed a model for the prediction of $\Phi_{0.1}$ for displacement ventilation systems that is a function of the airflow rate and is based on a heat transfer model between the ceiling and the floor. Mundt's equation is used in a cooling airflow design modeling tool developed by Chen and Glicksman [3].Most UFAD diffusers (e.g. swirl, linear bar grille, VAV directional, etc.) create less stratification than a DV system due to the vertical velocity component of the supply air as it leaves the floor diffuser, which causes increased mixing. Thus, $\Phi_{0.1}$ for UFAD systems is typically higher than for DV systems. Underfloor displacement ventilation diffusers differ from other UFAD diffusers in that they discharge air almost horizontally. It is expected, but yet not proven, that underfloor displacement ventilation diffusers may generate stratification similar to that generated by typical wall displacement diffusers.

Lin and Linden [8] and Liu and Linden [9] theoretically developed and experimentally tested (in a small-scale salt-tank model) a prediction of Φ for underfloor air distribution systems as a function of the non-dimensional parameter, Γ (Gamma) (see Equation 4 below). Webster et. al. [10] then used this model to develop stratification predictions based on full-scale experiments. The obtained Γ-Φ_{OZ} equations obtained by Webster et al. were then implemented in the EnergyPlus UFAD module [11,12] . The Γ-$\Phi_{0.1}$ and Γ-$\Phi_{1.7}$ equations have been implemented in the online CBE UFAD design tool [13,14].

Gamma, Γ, describes the ratio of buoyancy forces to vertical momentum forces and is commonly used in the analysis of UFAD systems. For a typical UFAD diffuser, a large value indicates that mixing dominates in the zone, and a small value indicates that stratification dominates in the zone. In the case of an interior zone, with multiple diffusers and multiple thermal plumes (i.e. heat sources) we calculate Γ using:

$$\Gamma = \frac{(Q \cdot \cos\theta)^{\frac{3}{2}}}{m \cdot \left(\frac{n}{m} A_d\right)^{\frac{5}{4}} \cdot (0.0281 \cdot W)^{\frac{1}{2}}}$$

Equation 4

Where: Q is the total zone airflow [m³/s] [2]; θ is the discharge angle from vertical [°]; n number of diffusers [-]; A_d is the effective area of single diffuser [m²]; and W is the cooling load [kW]. The effective diffuser area is always smaller than the geometric, or free, area of the diffuser and should be determined experimentally. Schiavon et. al. [15] describes Γ for perimeter zones, which may be useful to the reader but was not required for this experiment as we evaluated interior zones only.

The general shape of a diffuser and its associated fluid-dynamic characteristics affect temperature stratification in the room. Thus, the relationship between Γ and Φ varies by diffuser and must be determined experimentally through full scale laboratory testing. The primary aim of this experiment was to determine this relationship for displacement ventilation UFAD diffusers, also known as horizontal discharge diffusers. To our knowledge, only one such diffuser is currently available in the market and that is the diffuser we tested. Prior research [16] has shown that the Γ-Φ relationship is similar for diffusers of similar styles, and so the relationship we have determined should be applicable to other displacement ventilation UFAD diffusers assuming they have a similar discharge flow profile. Once determined, practitioners can use the Γ-Φ relationship to predict thermal stratification in a zone under a range of conditions, leading to more effectively designed systems.

Archimedes number is often used in the analysis of displacement ventilation systems [6]. Gamma and Archimedes number are conceptually similar, in that they both describe a ratio of buoyancy and momentum forces. The significant difference between the two systems is that in a displacement ventilation system the air typically exits the diffuser with no vertical momentum, whereas in a UFAD system the air leaving the diffuser typically has a distinct vertical momentum component which encourages mixing (i.e. a 'fountain' effect [8]). Gamma and Archimedes are separately formulated to be more suited to analyzing each case. To facilitate comparison of this

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[2] Note that Q is in m³/s in this equation, but is represented using l/s elsewhere in the paper for legibility and to match units commonly used within the industry.

experiment to previous experiments that investigated displacement ventilation systems, we analyze the results

using the Archimedes number, *Ar*, using the following equation, as defined in [17]:

$$Ar = \frac{\beta g H_c (T_R - T_S)}{\left(\frac{Q}{A_c}\right)^2}$$

Equation 5

where: β is the volumetric expansion coefficient [K^{-1}]; g is gravitational acceleration; H_c is the overall height of the

chamber [m]; A_c is the chamber floor area [m^2]; and Q is the total airflow [m^3/s].

3 METHOD

3.1 EXPERIMENTAL FACILITIES

We performed full scale experiments in a climatic chamber at Price Industries in Winnipeg, Canada in May 2014.

Figures 1 and 2 show a schematic image and photograph of the chamber, which is representative of a typical

ground floor office space in the U.S. conditioned by a UFAD system. It is located indoors inside a large laboratory

facility maintained to thermal comfort conditions, typically between 21 and 23 °C as measured during the

experiment. The occupied room of the climatic chamber was 6.10 m long by 4.88 m wide and 3.05 m high. It has a

total floor area and volume of the room of 29 m^2 and 88.3 m^3 respectively (due to a small cut out for the duct to

the supply plenum that is visible in the lower right-hand corner of Figure 1). We have included the major

dimensions in Figure 1, however it was intractable to include dimensions for every diffuser and heat source. The

image is to scale and the location of each object can be identified easily against the grid of uniform 0.61 m (2 ft)
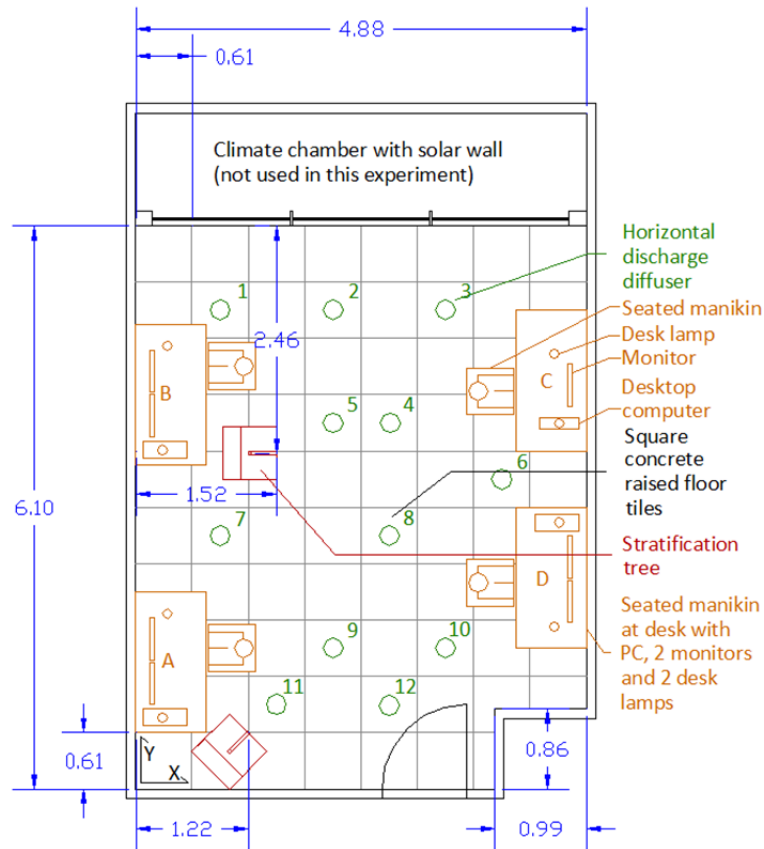
square floor tiles underneath.

Figure 1: Plan view of the climate chamber.

Figure 2: Photograph of a manikin seated at a desk in the climate chamber

The climatic chamber walls are made of 2 layers of 16 mm gypsum wallboard insulated with 150 mm of mineral wool (U factor of 0.26 W/m^2K including air films). There are also three windows to the laboratory space outside of

the test chamber. We did not require these for the purposes of this experiment so we applied rigid Styrofoam

insulation (50 mm) to the windows (visible in Figure 2) and sealed around the edges with tape (U factor of 0.37

W/m^{2}K including air films).

Standard 12.5 mm ceiling tiles formed the drop ceiling of the climate chamber. We insulated this with 0.1 m batt

fiberglass insulation to ensure that there was limited heat transfer in this direction (U factor of 0.34 W/m^{2}K

including air films). Concrete 25 mm raised floor panels, 0.46 m above a concrete slab-on-grade, separate the room

from the ductless supply plenum (U factor of 2.5 W/m^{2}K including air films). An air handling unit supplied

conditioned air to this underfloor supply plenum which we used to control the supply air to a set-point

temperature of 17.8 °C, fixed in all experiments. This supply air temperature set-point is commonly used in

underfloor air distribution systems in the US [18]. The air exhausted from the room through a duct in the ceiling in

the corner of the room (the lower left hand side of the plan Figure 1).

There is also a sealed and insulated solar simulator chamber adjacent to the climatic chamber (shown in Figure 1)

but separated by a clear glass double-pane window. The solar simulator was not used in this experiment.

3.2 INSTRUMENTATION

3.2.1   Temperature

We measured all temperatures using PT100 platinum resistance temperature detectors (RTDs) (Class 1/10 DIN)

accurate to ± 0.1 °C (or better) within the temperatures ranges in the experiment. We calibrated these prior to the

experiment using a Fluke 1524 Reference Thermometer in a Fluke 9171 Metrology dry well calibrator (± 0.029 °C).

We shielded air temperature sensors from radiant heat transfer using a highly reflective aluminum cylinder. The

only exception was the temperature sensor at 2.95 m (0.1 m from the ceiling). The shield was not present due to

technical difficulties. We attached temperature sensors to surfaces using reflective aluminum tape for surface

temperature measurements. Figure 1 shows the location of the two 'stratification trees' that we used to measure

room air temperature stratification. Each of these measured air temperatures at 9 heights: 0.1, 0.3, 0.6, 1.1, 1.4,

1.7, 2.2 and 2.95 m. We measured supply and exhaust air temperature at the opening of the duct as the air

entered or exited the test chamber. We measured diffuser discharge air temperature in open diffusers (up to a limit of 6 diffusers due to a limited number of data acquisition channels) by placing a shielded temperature sensor at a depth of 0.05 m inside each open diffuser.

We used the temperatures measured by the stratification tree near the center of the chamber (adjacent to Desk B in Figure 1) when calculating results using Equations 1-3, as it is far from both thermal plumes and open diffusers in all tests. Excluding locations directly influenced by the thermal plume from a heat source, or by the airflow from an open diffuser, we assumed there to be little significant horizontal variation in temperature within the space – i.e. that the temperatures measured near the center of the chamber are representative of the room as a whole, as have been discussed or measured in previous papers on the topic (e.g. [6,18–20]).

We calculated the propagated uncertainty for Φ according to JGCM guidelines [21] with a level of confidence of 95% (coverage factor of 2). The uncertainty depends partially on the value of Φ, but is between ± 0.02 and ± 0.03 for all values reported in this paper.

3.2.2   Flow and pressure

We measured air flow rate at both the supply and exhaust using a Gill Propeller Anemometer (Model 27106). These devices have a minimum accuracy of ± 2% of the reading, or ± 0.5 l/s at the low end of the measurement range (± 16 l/s). We calibrated both the supply and return devices in-situ using a subsonic venturi, recently calibrated and accurate to ± 0.13 l/s within the flow ranges used in the experiment.

We measured air pressure in the supply air plenum using a Setra very low differential pressure transducer (Model 265) with an accuracy of ± 0.6 Pa within the pressure ranges used in the experiment.

3.2.3   Other measurements

We measured the diffuser effective area, A_d, using a Duct Blaster [22], which has an uncertainty of ±3% in the measured range. We visually assessed the diffuser discharge angle using video footage of repeated smoke tests and estimate the uncertainty at ±5 °. We measured the electrical power consumed by each internal load

component described in section 3.3 using a power meter (Fluke 41 Power Harmonics Analyzer, Fluke Corp.) which was calibrated prior to the experiment.

3.3   INTERNAL LOADS

We simulated internal loads using actual equipment used in office spaces – computers, monitors, and lamps – an approach that has been used in other similar studies [20]. This ensures that the loads are realistically represented and their simulation is not a factor in this experiment, as it has been shown have an effect in other studies [17,19,23]. For example, many studies use a standard cylinder design to simulate all internal loads, such as that described in EN 14240:2004 [24]. This design draws air through the bottom of the cylinder at floor height due to a miniature 'stack effect' and would likely have a significant effect on the thermal plume when compared to typical office heat gains. In turn, this may have an effect on temperature distribution in a space, particularly one that investigates a system supplying cold air at floor level. While actual equipment can be used to for lighting, monitor, and computer loads, it was not feasible to use human subjects in this experiment. Thus, we simulated occupants using manikins wrapped in electrically heated tape which generate 75 W of internal load at realistic skin temperatures and over a realistic geometry. We will present the details of the design and testing of these low-cost manikins in a future paper, making it publicly available.

The primary disadvantages of using actual equipment and manikins to simulate thermal loads is that it adds complexity to the experiment and makes it more difficult to replicate. However, the authors believe that the benefits outweigh this disadvantage and have provided the necessary detail to replicate this experiment in Figure 1 and Table 1, which describe the internal loads used in the experiment. In addition to these loads, the overhead lighting consumed 110 W per fixture and the data acquisition equipment in the room consumed a constant 6 W. We varied the number of occupied desks, the number of monitors and lamps per desk, and the overhead lights in order to obtain the wide range of internal load test conditions examined in these experiments. The bulb in the lamp at desk-height on Desk D burnt out during the testing and was replaced with a 60 W bulb. This is taken into account in the relevant calculations in this paper.

Table 1. Internal heat source summary (as measured using Fluke 41 Power Harmonics Analyzer).

Occupant station	Desk A [W]	Desk B [W]	Desk C [W]	Desk D [W]
Manikin	75	75	75	75
Lamp (at desk height)	42	51	51	41
Monitor 1 (left)	21	28	28	29
Monitor 2 (right)	33	28	N/A	24
Vertical 'tower' desktop computer (at desk height)	52	54	69	112
Lamp (at computer height)	60	60	87	59

3.4 LEAK TESTING

It is essential to assess the amount of air leakage from the supply plenum before performing experiments that use

underfloor air distribution systems. High leakage rates will have a negative impact on experiment accuracy –

particularly for air that leaves the chamber into the outer laboratory space. In order to assess leakage we sealed all

of the diffusers in the raised floor and pressurized the supply plenum within the range of operating pressures used

in the experiment, as measured between the supply plenum and the laboratory space. We then measured the

resultant leakage flow and present those results in Figure 3. The support staff performed a thorough inspection

and sealed a number of leakage paths. Figure 3 shows the very low leakage rates achieved before starting the

experiment. The typical supply plenum pressure range for this diffuser is between 7 Pa and 30 Pa in practice.
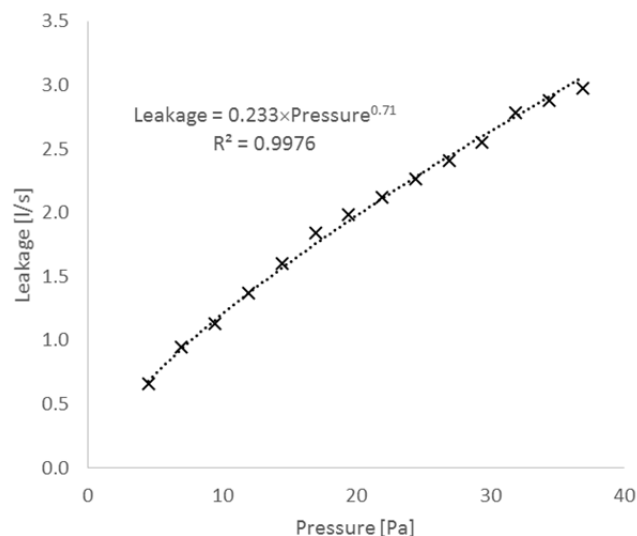

$$Leakage = 0.233 \times Pressure^{0.71}$$
$$R^2 = 0.9976$$

3.5   DIFFUSER CHARACTERISTICS

We tested an underfloor air displacement ventilation diffuser, described in detail in the manufacturer's specification [25]. We determined that the diffuser had an effective area (A_d) of $2.1 \cdot 10^{-3}$ m^2 and a 75 ° discharge angle measured from the vertical. These diffusers do not modulate flow, and thus operate at a fixed airflow for a given plenum pressure. The design operating range is between 12 l/s and 27 l/s at 7 Pa and 30 Pa respectively, and we saw little observable change between the discharge angle at the high and low end of this range. We varied both the number of diffusers and the supply plenum pressure to obtain a range of conditions in this experiment.

Figure 4: Photograph of diffuser installed in raised floor with discharge air temperature sensor in place

3.6   EXPERIMENT PROCEDURE

We conducted a total of 19 tests from May 14th to 28th, 2014. Table 2 summarizes the parameters that we varied in each test, which are we numbered sequentially in the order that they were performed. We performed several repeated tests both initially (within the first internal load block) and again at the end of the experiment (repeating tests in the medium-high internal load block).

Table 2 gives an overview of the major parameters used in each test. After selecting the internal load, we estimated the airflow required at that load to maintain acceptable thermal comfort conditions for a typical office space according to ASHRAE Standard 55-2013 [26]. We then fixed the airflow set-point for each internal load level and varied the number of open diffusers in order to obtain a large range of Γ values while still operating within the diffuser design specifications. We performed three replicated tests at the medium load case (27.6 W/m^2), highlighted by the superscript 'r1' after the test number. We also duplicated two of the medium-high load cases (41.3 W/m^2), highlighted by the superscript 'r2' and 'r3' after the test number.

Table 2: Experiment input parameters (overview)

Test No.	No. open diffusers	Powered workstations	Occupants [W]	Monitors [W]	PCs [W]	Desk lights [W]	Overhead Lights [W]	Total internal load [W/m^2]	Supply airflow [l/s]
1^{r1}	3	A, B & C	225	77	175	93	220	27.6	48.5
2	2	A, B & C	225	77	175	93	220	27.6	48.6
3^{r1}	3	A, B & C	225	77	175	93	220	27.6	48.6
4	4	A, B & C	225	77	175	93	220	27.6	48.7
5^{r1}	3	A, B & C	225	77	175	93	220	27.6	48.7
6^{r2}	4	A, B, C & D	300	191	287	185	220	41.3	89.7
7^{r3}	6	A, B, C & D	300	191	287	185	220	41.3	89.7
8	3	A, B, C & D	300	191	287	185	220	41.3	89.7
9	5	A, B, C & D	300	191	287	470	440	58.8	132.1
10	9	A, B, C & D	300	191	287	470	440	58.8	132.2
11	3	A, B, C & D	300	191	287	470	440	58.8	132.1
12	4	A, B, C & D	300	191	287	470	440	58.8	132.0
13	6	A, B, C & D	300	191	287	470	440	58.8	132.2
14	4	A & C	150	49	121	42	220	20.4	40.6
15	3	A & C	150	49	121	42	220	20.4	35.7
16	2	A & C	150	49	121	42	220	20.4	28.5

17[r3]	6	A, B, C & D	300	191	287	204	220	41.9	91.6
18	7	A, B, C & D	300	191	287	204	220	41.9	89.9
19[r2]	4	A, B, C & D	300	191	287	204	220	41.9	89.5

While performing the tests, we modified the return fan airflow set-point to maintain a slight positive pressure (> 0 Pa and < 2 Pa) between the climatic chamber and the laboratory surrounding the chamber. This differential pressure ensured that any leakage was from the chamber out into to the laboratory, and not a confounding variable in our analysis. We also performed numerous smoke tests, in which we injected smoke into the supply air duct and observed the resultant flow pattern within the room.

Table 3 shows the specific diffusers (as numbered in Figure 1) that we opened for each level of diffusers (i.e. total number). For example, regardless of the internal load level if we had three open diffusers, these were always #2, #8, and #10. We selected the locations of diffusers in an effort to maintain (a) an even distribution of airflow across the room, (b) a realistic distance from the diffusers to the manikins (i.e. the simulated occupants) and (c) a reasonable distance from the diffusers to the stratification tree. Finding an 'undisturbed' location to place the stratification tree can be an issue, as we have found in previous studies with typical UFAD diffusers. However, we did not encounter this issue in this experiment, presumably due to the low flow rates and almost horizontal discharge angle characteristics of this displacement ventilation diffuser. For each diffuser case, at the highest internal load case (and correspondingly the highest diffuser discharge velocity), we measured the air velocity at the lower three measurement locations on the stratification tree to verify that the airflow from the diffusers was not directly impinging on the temperature sensors.

Table 3: Experiment input parameters (diffuser details)

Number of diffusers	Open diffuser locations
2	#2, #10
3	#2, #8, #10
4	#2, #6, #8, #10

5	#1, #3, #8, #9, #10
6	#1, #3, #6, #8, #9, #10
7	#1, #3, #6, #7, #8, #9, #10
9	#1, #2, #3, #4, #6, #7, #8, #9, #10

We recorded measured data once the test had reached steady state conditions, which we defined as a difference of less than 0.05 °C between the mean of the most recent 200 samples against the mean of the 200 samples immediately prior, for every sensor used in the test. After recording the data we calculated: (1) The average temperatures in the supply plenum, at the diffuser discharge, and in the occupied zone; (2) the dimensionless temperature stratification, Φ, at each height within the room according to Equation 1; (3) Γ according to Equation 4; and (4) Ar using Equation 5.

3.7 STATISTICAL ANALYSIS

We present the results for an individual measurement in a specific test using the mean of 200 samples at ten second intervals rounded to one decimal place, or to two significant digits, whichever is more precise. We did not report the 95% confidence interval for the mean because it was lower than the instrument accuracy for every sensor in every test, and we wanted to ensure that the reader does not infer a higher degree of precision than is valid for these measurements. We present aggregate results across multiple tests using the median of the means from each individual experiment followed by the lower and upper quartile in parentheses. For example, the median air temperature at 1.4 m high, $T_{1.4}$, across all of the tests was 24.6 (24.0, 25.0) °C.

We performed the statistical analysis using the R programming language, version 3.0.2 [27] and developed each of the following figures using Microsoft Excel 2013.

4   RESULTS AND DISCUSSION

4.1   OVERVIEW OF THE RESULTS

The supply air flow rate was 89.7 (48.6, 111.8) l/s and the supply plenum pressure was 6.6 (5.1, 19.8) Pa. The supply air temperature was 17.8 (17.8, 17.8) °C and the diffuser discharge temperature was 19.6 (19.5, 19.8) °C. The temperature increase across the plenum highlights the effect of thermal decay in the supply plenum typical of an underfloor air distribution system 1.8 (1.7, 2.0), which is at the low end of the range reported in a recent simulation study [28]. We theorize that this is due to the higher level of stratification achieved using these diffusers, which yield cooler air temperatures near the top of the raised floor, and hence, a lower temperature differential (and less heat transfer) from the room to the supply plenum than for a more typical UFAD diffuser.

The temperatures in the room were 21.5 (21.4, 22.0) °C at 0.1 m high (ankle height), 22.0 (21.9, 22.2) °C at 0.6 m high, 24.0 (23.8, 24.2) °C at 1.1m high (seated head height), and 24.9 (24.2, 25.4) °C at 1.7 m high (standing head height). These standard head heights are defined according to international, American and European standards (ISO 7730, ASHRAE 55, and CEN 15251). The average occupied zone temperature, T_{OZ} was 23.1 (22.9, 23.4) °C which is within the suggested comfort range for typical comfort conditions from ASHRAE 55-2013 [26] for each experiment. Likewise, the difference in temperature between head and ankle height for a standing occupant ($T_{1.7}$ - $T_{0.1}$) in the experiments was 3.4 (2.7, 3.5) °C, which is below the comfort recommended limit of 4 °C. Also, the difference in temperature between head and ankle height for a seated occupant ($T_{1.1}$ - $T_{0.1}$) was 2.3 (2.2, 2.6) °C which is below the comfort recommended limit of 3 °C. Lastly, the exhaust air from the room was 25.1 (25.5, 24.4) °C.

We performed tests at same internal load and same number of diffusers (with lower and higher number of diffusers levels in between) in order to assess the repeatability of the experiment. The maximum variation in $\Phi_{0.1}$, Φ_{OZ}, or $\Phi_{1.7}$ was 0.0094 between these repeated experiments, indicating a high level of repeatability.

4.2 VERTICAL TEMPERATURE PROFILES

Figure 5 shows the vertical temperature profiles within the chamber. We present the results by load level as we found this factor had a larger effect than the number of diffusers open. We also group the tests at the 41.3 W/m^2 and the 41.9W/m^2 load levels together, and represent them using 41.5 W/m^2.
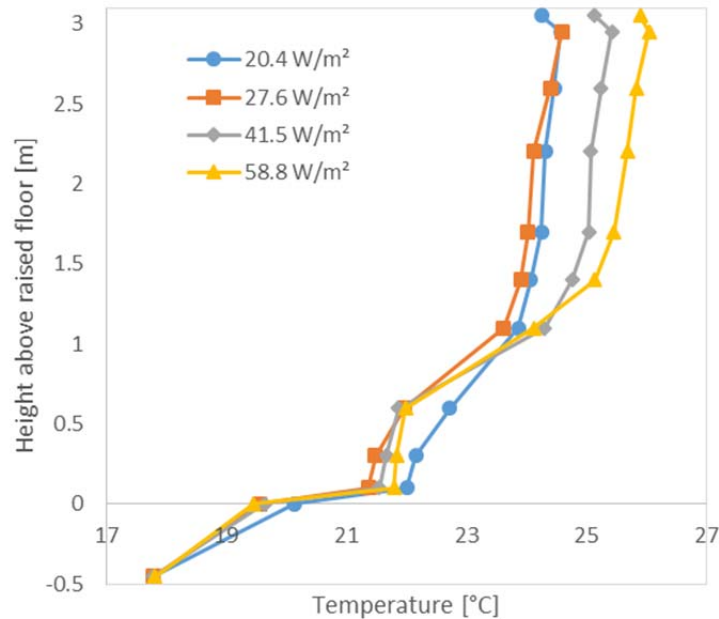

Figure 5: Vertical temperature profiles within the room (averaged at each level of internal load

Each profile in Figure 5 shows a marked increase in temperature between the temperature measurements at 0.6 m and 1.1 m, approximately the height at which the majority of heat gains occur in the chamber. We varied airflow with the level of internal load to approximate a constant zone air temperature, which is how these systems are typically controlled in buildings, and thus the difference in the temperature profiles in the occupied zone was not very large, usually less than 1°C in the occupied zone. By almost tripling the load the temperature at each height increased by less than 1.5°C, and the larger temperature differences occur in the upper mixed zone. As discussed later, there is a slight tendency for stratification to increase (i.e. for $\Phi_{0.1}$ to decrease) as the level of internal load (and airflow) increases.

As well as measuring temperature profiles in the room, we also injected smoke into the air as it enters the supply plenum and visually observed the air flow pattern in the space afterwards. These smoke tests showed a layer of smoke occurring at a height within approximately the same range as Figure 5 indicates. This is visible in Figure 6 and the video available in the electronic version of this publication. Both show that the supply air forms a thick layer at the bottom of the zone. The continuous supply airflow drives the smoke vertically quite slowly - except where it is entrained by a thermal plume. We measured the variation of temperature with height, but did not do so with fine vertical resolution. We can see from the measurements at 0.6m and 1.1m that stratification occurred somewhere between those two heights for each of the experiments, but cannot state with any more precision where it occurred.
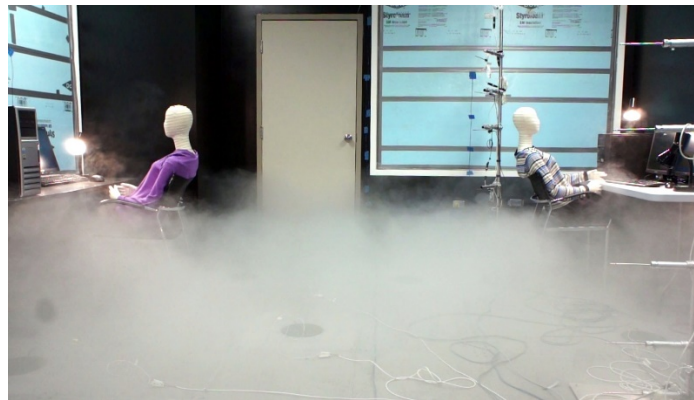
Figure 6: Still frame from a video of a diffuser smoke for test number 10. The video can be viewed in the online version of this article.

Figure 5 also shows that the temperature difference between head and ankle height at the highest internal load case (58.8 W/m^2) approaches the recommended comfort limit of 4 °C and 3 °C for a standing and seated occupants respectively [26], indicating that this load level is a reasonable design maximum for these types of diffusers. This 58.8 W/m^2 load level corresponds to an occupancy of 7.25 m^2/person, 15 W/m^2 lighting, and 32 W/m^2 plug loads, which is representative of a densely occupied office with very high plug loads per occupant, and is far higher than the loads in a typical office building.

As mentioned at the beginning of this section, the level of internal load had a far higher effect on the results than the number of diffusers open. In fact, the number of open diffusers only had a noticeable effect when at the

highest internal load level, as Figure 7 shows. There was very little difference between the number of diffusers open at the three internal load levels – for brevity, we do not present these results.
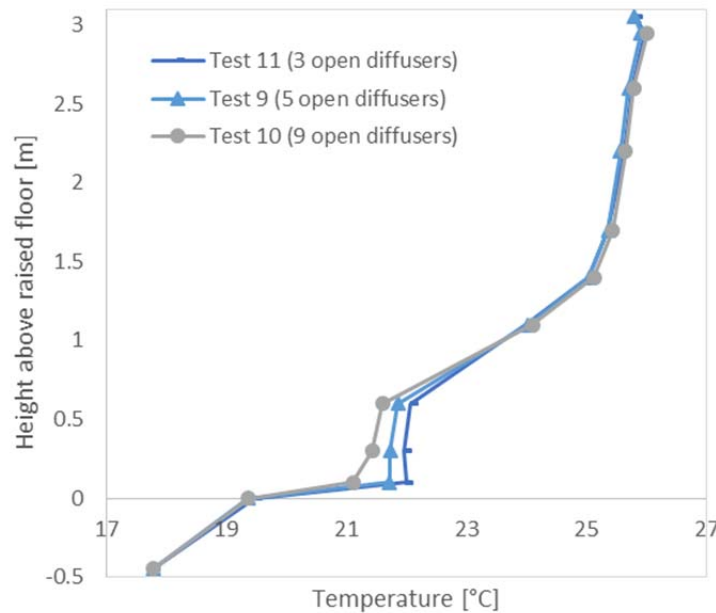

Figure 7: Effect of varying the number of open diffusers at a fixed internal load level of 58.8 W/m^2

4.3 Γ-Φ RELATIONSHIP

Gamma, as defined in Equation , was equal to 15.3 (11.4, 20.7). The Archimedes number, as defined in Equation 5, was 60 (44, 185) · 10^3. $\Phi_{0.1}$ was 0.38 (0.36, 0.40), Φ_{OZ} was 0.67 (0.62, 0.67) and $\Phi_{1.7}$ was 0.94 (0.93, 1.00). The range of stratification values at ankle height are within the ranges typically reported for DV systems ($\Phi_{0.1}$ = 0.2-0.7 [5]), and below those reported for typical UFAD systems ($\Phi_{0.1}$=0.70 (0.60-0.85)) [16]. A decrease in Φ indicates increased stratification, and thus the results show that the displacement ventilation diffusers produce more stratification in the occupied zone than a typical UFAD diffuser. The results show that these diffusers can maintain comfort conditions while causing more temperature stratification. This implies that the DV diffuser will use less energy (when operating in cooling mode) because the average room temperature will be higher than in rooms conditioned by other UFAD diffusers, or by a typical overhead system, while providing the same level of comfort.

The results also show that Φ, is almost equal to 1 at head height (1.7 m), indicating that the temperature at that height is very close to the room exhaust air temperature, and that the upper zone is fully mixed. Figure 8 shows the relationship between gamma and phi for ankle height (0.1 m), the occupied zone, and at head height (1.7 m).
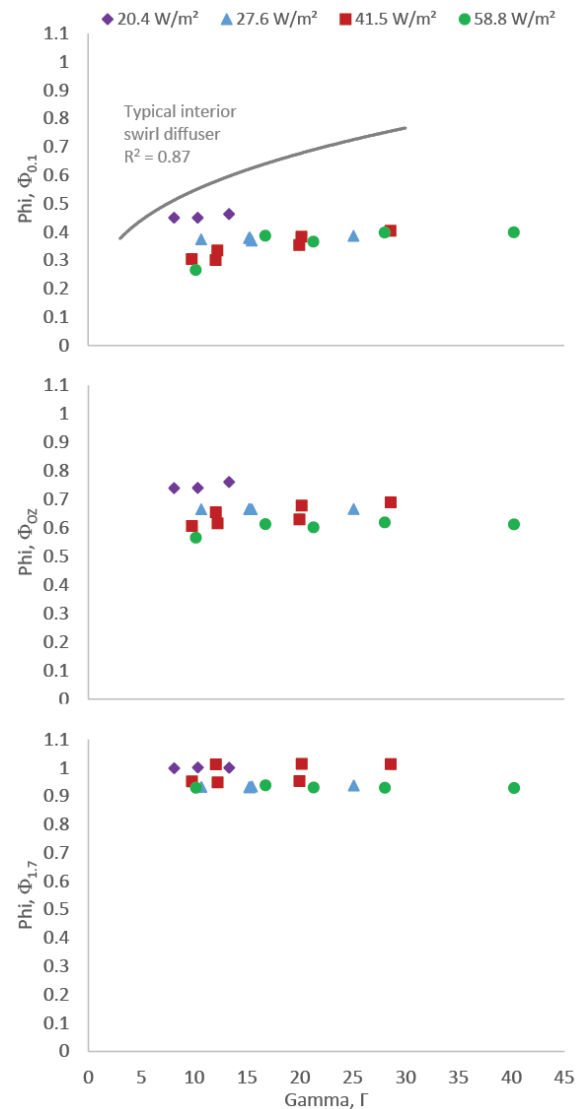

Figure 8: Γ-Φ plot showing the results from all 19 tests for $\Phi_{0.1}$, Φ_{OZ} and $\Phi_{1.7}$ and load level indicated by colored legend. The uppermost chart also includes a regression fit for a swirl diffuser for comparison purposes [16].

It is clear from Figure 8 that Γ has little effect on Φ. This means that thermal stratification is far less dependent on the level of internal load, the total airflow, the number of thermal plumes, and the number of diffusers than a typical UFAD system. As the vertical momentum of the discharge air is very low for this diffuser, a ratio of

buoyancy-driven to vertical-momentum-driven flow (i.e., Γ) should not yield a correlation. However, as this is an essential relationship for all other UFAD diffusers to date, existing models (such as those implemented in EnergyPlus or the CBE UFAD design tool) use this relationship. Thus that is how we present our results in this paper - so that they can be easily used in existing tools.

It is also useful to present the results in a way that facilitates direct comparison with a typical displacement ventilation system. This has typically been done by presenting dimensionless stratification at ankle height for a particular flow rate per floor area. Comparing the results from this experiment to other displacement ventilation experiments, we measured values for $\Phi_{0.1}$ at the lower end of the reported range of 0.2- 0.7 [5] and 0.3 – 0.7 [6]. Looking at specific experiments, Mundt [7] presents the data from 12 different full scale laboratory studies of displacement ventilation systems in this manner. It found values ranging from approximately $\Phi_{0.1}$ = 0.5 at relatively low flow rates of 0.0015 $m^3/s{\cdot}m^2$ down to approximately $\Phi_{0.1}$ = 0.2 at higher flow rates of 0.008 $m^3/s{\cdot}m^2$. To put the results from our experiment in this context, we measured $\Phi_{0.1}$ = 0.45 at the lowest flow rates (0.0012 $m^3/s{\cdot}m^2$) and $\Phi_{0.1}$ = 0.39 at the highest flow rates (0.0046 m^3/sm^2). There is less variation in our results at different flow rates, however it is important to note that our experiments varied airflow and heat gains simultaneously (as typically occurs in buildings that use these systems), and this may also have had an effect.

Another way to compare this experiment to previous displacement ventilation experiments is to use the Archimedes number. Figure 9 shows that $\Phi_{0.1}$ increases with increasing Archimedes number, but the effect is minimal. To put the results from this experiment in context, we measured values of $\Phi_{0.1}$ in the range of 0.3 to 0.4 at Ar= 60$\cdot$10^3, which is at the low end of values presented in a previous study [19] where $\Phi_{0.1}$ ranged from 0.32 to 0.65 at the same Archimedes number, depending on whether the heat source was a point, ceiling light, thermal manikins, or a heated floor.
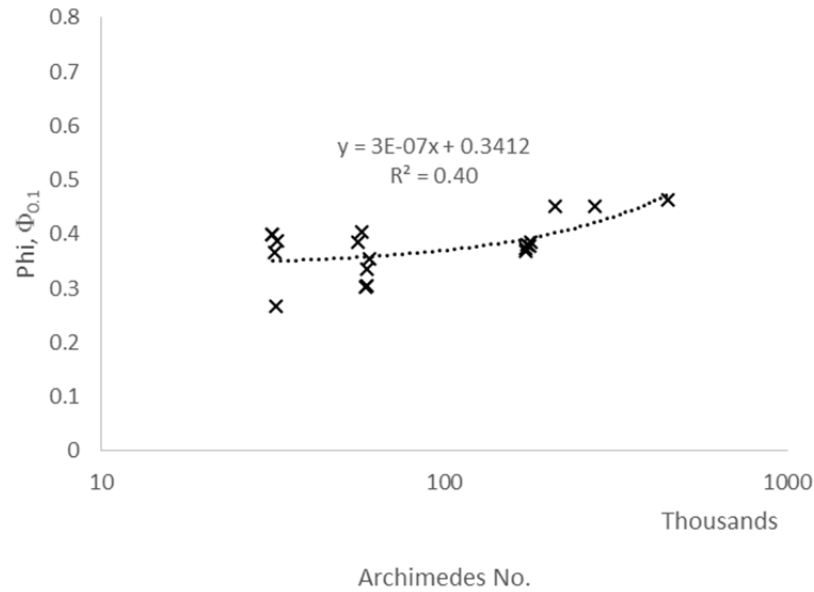
Figure 9: Plot showing variation in $\Phi_{0.1}$ with respect to _Ar_, Archimedes number for all experiments.

However, it should be noted that all of the compared studies above do not necessarily represent the distribution of room loads that occur in a UFAD system, where conditioned air passes through a ductless supply plenum and absorbs heat from the room before entering the occupied space through the supply diffuser in the raised floor. We found only one study that investigates a case that approximates a UFAD system with displacement ventilation. This study [20] tested air supplied through a floor plenum and then into the room through a highly perforated (9 mm holes on 30 mm centers), raised wooden floor – so the primary difference is a more uniform airflow pattern compared to the case presented in this paper, in which horizontal discharge diffusers were used. This study also measured values of Φ at the low end of the range typically reported for DV systems ($\Phi_{0.05}$ as low as 0.2).

The level of internal load affects the amount of temperature stratification achieved in the space. Figure 10 shows a clear correlation between occupied zone stratification and level of internal load. We theorize that this is either due to the increased overall total airflow required to cool the zone under these conditions, as discussed in Mundt [7], or the average height at which the loads occur within the zone. Previous studies on displacement ventilation systems [6] note that the height at which the loads are located in the zone affects the height at which stratification occurs. This phenomenon has been reported in experiments with displacement ventilation systems [6,29,30]. This yielded increased stratification implying that designers and operators can improve both energy and comfort

performance by increasing the height at which internal loads are located in the space. This may be difficult to do in practice, but there are several measures that may be possible depending on the particular building and use case: to locate computers on shelves instead of at floor or desk height; to mount task lights on ceilings or attached to walls/furniture above the typical desk height; or to supply occupants with sit-stand desks.
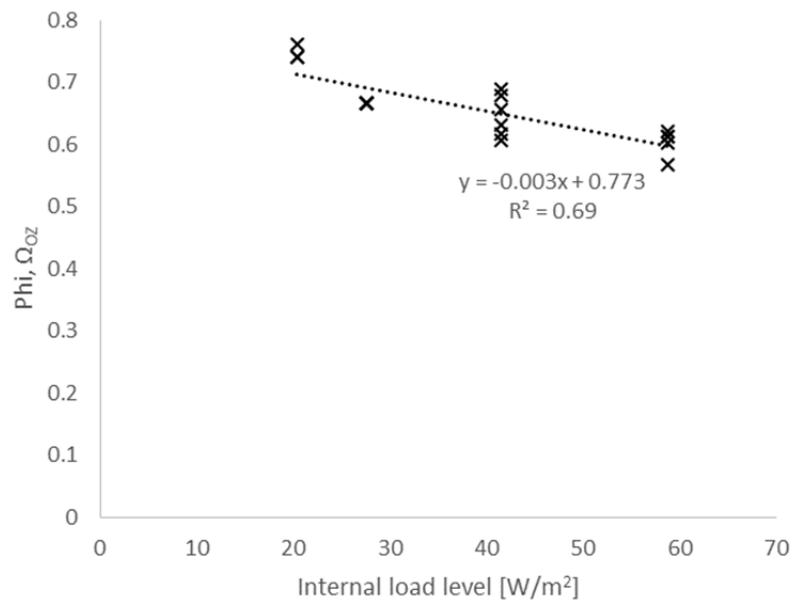

Figure 10: Relationship between load and stratification in the occupied zone.

The findings from this experiment indicate that the following fixed values $\Phi_{0.1} = 0.38$, $\Phi_{OZ} = 0.67$ and $\Phi_{1.7} = 0.94$ are reasonable approximations for the range of conditions present in a typical interior zone office that uses this system. These values are based on this experiment and the results may vary under circumstances that are not represented in the experiment as described in this paper. However, in the absence of further experimental data, these values can be used when designing or simulating buildings that use displacement diffusers (i.e., horizontal discharge diffusers) supplied with air from an underfloor plenum. We have updated the publicly available UFAD cooling load design tool developed by the Center for the Built Environment at UC Berkeley with the new Γ-Φ relationships for this diffuser (http://www.cbe.berkeley.edu/ufad-designtool/online.htm) [14].

5 CONCLUSIONS

We measured the vertical temperature distribution within a room where we varied the level of heat gain in the room and the number of open diffusers in the raised floor. The main conclusions of this study are:

- The dimensionless stratification at ankle height, $\Phi_{0.1}$, was 0.38 (0.36, 0.4) with a displacement diffuser in an underfloor air distribution system. This represents a level of thermal stratification within the range typically reported for displacement ventilation systems, and a higher level of stratification than reported for typical UFAD systems $\Phi_{0.1}$ = 0.7 (0.6, 0.85). The dimensionless stratification in the occupied zone, Φ_{OZ}, was 0.67 (0.62, 0.67). The average stratification at head height, $\Phi_{1.7}$, was 0.94 (0.93, 1.00), representing temperatures that are very close, or equal to the exhaust air temperature for the zone.

- Temperature stratification in the occupied zone increased with increasing levels of internal load in this experiment, as evaluated using the dimensionless temperature, Φ_{OZ}, although the effect was minor.

- The number of diffusers did not have a large effect on stratification within the test conditions evaluated within this experiment

- The DV floor diffusers investigated in this study have advantages over typical low side-wall DV diffusers because (1) they can be located anywhere on the large surface area of a raised access floor (even in open plan offices which may have limited wall area in which to accommodate a typical DV diffuser), and (2) they can be controlled (open/closed) by nearby occupants. Since these DV floor diffusers demonstrated equivalent stratification performance to conventional DV systems, it is likely that their ventilation performance will also be comparable. Future research is needed to verify this performance.

6 ACKNOWLEDGMENTS

7   REFERENCES

[1]   ASHRAE, ASHRAE Standard 62.1-2013 - Ventilation for Acceptable Indoor Air Quality, (2013).

[2]   Kisup Lee, Zheng Jiang, Qingyan Chen, Air distribution effectiveness with stratified air distribution systems, ASHRAE Transactions. 115 (2009). https://engineering.purdue.edu/~yanchen/paper/2009-9.pdf.

[3]   Kisup Lee, TengFei Zhang, Zheng Jiang, Qingyan Chen, Comparison of airflow and contaminant distributions in rooms with traditional displacement ventilation and under-floor air distribution systems, ASHRAE Transactions. 115 (2009). https://engineering.purdue.edu/~yanchen/paper/2009-8.pdf.

[4]   A. Jung, M. Zeller, Analysis and testing of methods to determine indoor air quality and air change effectiveness, Rheinisch-Westfälische Technical University of Aachen, Germany. (1994). http://escholarship.org/uc/item/5kd7w8q8 (accessed January 16, 2015).

[5]   Qingyan Chen, Leon Glicksman, System Performance Evaluation and Design Guidelines for Displacement Ventilation, ASHRAE, 2003.

[6]   Håkon Skistad, Elisabeth Mundt, Peter V. Nielsen, Kim Hagström, Jorma Railio, No 1: Displacement ventilation in non-industrial premises, REHVA, 2000.

[7]   E. Mundt, Displacement ventilation systems—Convection flows and temperature gradients, Building and Environment. 30 (1995) 129–133. doi:10.1016/0360-1323(94)E0002-9.

[8]   Y.J.P. Lin, P.F. Linden, A model for an under floor air distribution system, Energy and Buildings. 37 (2005) 399–409. doi:doi: DOI: 10.1016/j.enbuild.2004.07.011.

[9]   Q.A. Liu, P.F. Linden, The fluid dynamics of an underfloor air distribution system, Journal of Fluid Mechanics. 554 (2006) 323. doi:10.1017/S0022112006009414.

[10]   F. Bauman, T. Webster, P. Linden, F. Buhl, Energy performance of underfloor air distribution systems - Final Report to CEC PIER Program. CEC Contract No. 500-01-035., Center for the Built Environment, University of California Berkeley, USA, 2007. http://www.cbe.berkeley.edu/research/briefs-ufadmodel.htm.

[11]   Tom Webster, Fred Bauman, Fred Buhl, Allan Daly, Modeling of Underfloor Air Distribution (UFAD) Systems, in: Proceedings of the Third National Conference of IBPSA-USA, 2008.

[12]   Q.A. Liu, P.F. Linden, The EnergyPlus UFAD module, in: Proceedings of the Third National Conference of IBPSA-USA, 2008: pp. 23–28.

[13]   Fred Bauman, Stefano Schiavon, Tom Webster, Kwang Ho Lee, Cooling load design tool for UFAD systems, ASHRAE. (2010) 62–71.

[14]   S. Schiavon, K.H. Lee, F. Bauman, T. Webster, Simplified calculation method for design cooling loads in underfloor air distribution (UFAD) systems, Energy and Buildings. 43 (2011) 517–528. doi:10.1016/j.enbuild.2010.10.017.

[15]   Stefano Schiavon, Kwang Ho Lee, Fred Bauman, Tom Webster, Development of a simplified cooling load design tool for underfloor air distribution (UFAD) systems, (2010). http://escholarship.org/uc/item/6278m12z.

[16] S. Schiavon, T. Webster, D. Dickerhoff, F. Bauman, Stratification prediction model for perimeter zone UFAD diffusers based on laboratory testing with solar simulator, Energy and Buildings. 82 (2014) 786–794. doi:10.1016/j.enbuild.2014.07.056.

[17] P.V. Nielsen, Temperature Distribution in a Displacement Ventilated Room, in: Proceedings of the 5th International Conference on Air Distribution in Rooms, 1996. http://vbn.aau.dk/en/publications/temperature-distribution-in-a-displacement-ventilated-room(de62ef83-f9ed-4798-8ee8-a37300765869).html (accessed July 9, 2014).

[18] F. Bauman, UFAD Guide: Design, Construction, and operation of Underfloor Air Systems., 1st ed., ASHRAE, 2013.

[19] P.V. Nielsen, Vertical Temperature Distribution in a Room with Displacement Ventilation, in: IEA Annex 26: Energy Efficient Ventilation of Large Enclosures, Rome, Italy, 1995. http://vbn.aau.dk/en/publications/vertical-temperature-distribution-in-a-room-with-displacement-ventilation(9650dca4-485e-41c7-ad03-7f6684106e9b).html (accessed July 9, 2014).

[20] T. Akimoto, T. Nobe, Y. Takebayashi, Experimental Study on the Floor-Supply Displacement Ventilation System, ASHRAE Transactions. 101 (1995) 912–925.

[21] JCGM, JCGM 100-2008: Evaluation of measurement data - Guide to the expression of uncertainty in measurements, Joint Committee for Guides in Metrology, (2008).

[22] The Energy Conservatory, Duct Blaster Systems and Accessories, Duct Blaster Systems and Accessories. (2014). http://www.energyconservatory.com/products/duct-blaster%C2%AE-systems-and-accessories.

[23] D. Zukowska, A. Melikov, Z. Popiolek, Impact of personal factors and furniture arrangement on the thermal plume above a sitting occupant, Building and Environment. 49 (2012) 104–116. doi:10.1016/j.buildenv.2011.09.015.

[24] ISO, EN 14240:2004 Ventilation for buildings, Chilled ceilings, Testing and rating, (2004).

[25] Price Industries, Round Floor Displacement Diffuser | Price Industries - The Science of Comfort, Round Floor Displacement Diffuser | Price Industries - The Science of Comfort. (2014). http://www.priceindustries.com/products/details/rfdd-arfhd-roundfloor-displacement-diffuser (accessed July 23, 2014).

[26] ASHRAE, ASHRAE Standard 55 - 2013 Thermal Environmental Conditions for Human Occupancy, (2013).

[27] R Core Development Team, The R Project for Statistical Computing, (2014). http://www.r-project.org/ (accessed April 7, 2014).

[28] K.H. Lee, S. Schiavon, F. Bauman, T. Webster, Thermal decay in underfloor air distribution (UFAD) systems: Fundamentals and influence on system performance, Applied Energy. 91 (2012) 197–207. doi:10.1016/j.apenergy.2011.09.011.

[29] S. Schiavon, F.S. Bauman, B. Tully, J. Rimmer, Chilled ceiling and displacement ventilation system: Laboratory study with high cooling load, Science and Technology for the Built Environment. 0 (2015) 00–00. doi:10.1080/23744731.2015.1034061.

[30] P.V. Nielsen, Displacement Ventilation, Dept. of Building Technology and Structural Engineering, Aalborg, 1993. http://vbn.aau.dk/ws/files/36321474/Displacement%20Ventilation.pdf (accessed July 9, 2014).