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RESEARCH ARTICLE - MECHANICAL ENGINEERING

The Study on Thermal Environment and Airflow Pattern in an UFAD System Under a Cooling Mode

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Abstract

In the present research, an investigation of the UFAD system for thermal comfort has been conducted in a high-rise building located in the tropics. The indoor air conditions including temperature, relative humidity, air velocity, and mean radiant temperature have been obtained by conducting the fieldwork while the clothing insulation value and the metabolic rate of the occupants have been obtained by observing the occupants, where these data were used to obtain the predicted mean vote (PMV) and the predicted percentage dissatisfied (PPD) of the examined areas. In addition, the effects of the airflow pattern in the indoor thermal comfort have been investigated, where two different types of diffusers have been compared in order to find out which diffuser can provide a better thermal comfort to the occupants. The FloEFD simulation software is used to simulate the airflow pattern of these diffusers and to analyze the indoor air conditions of the UFAD system and also to examine the local mean age value. Based on the results obtained, the average PMV is approximately − 1.5 for each examined area, where a proper design of heating, ventilation, and air-conditioning system in a hot and humid country, the PMV result should be approximately equal to − 1. As for the PPD, the range of the PPD obtained falls in between 27.4 and 67.5%, in which it indicates that about more than half of the occupants have dissatisfied with the indoor conditions in the examined building.

Keywords UFAD · Thermal comfort · PMV · PPD · Airflow pattern · FloEFD

Abbreviations

AHU	Air handling unit
FVM	Finite volume method
HVAC	Heating, ventilating and air-conditioning
IAQ	Indoor air quality
LMA	Local mean age
MRT	Mean radiant temperature
OHAD	Overhead air distribution
PMV	Predicted mean vote
PPD	Predicted percentages dissatisfied
UFAD	Underfloor air distribution

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1 Introduction

People nowadays spend most of their time in the indoor, and hence, the indoor thermal comfort has become important. This is because thermal discomfort (occupants feel too cold or too hot) can lead the occupant to lose their focus, which, in turn, can reduce their productivity and performance of their work [1, 2]. The four major environmental components which can affect the indoor thermal comfort are the air temperature, relative humidity, air velocity (airflow rate), and the mean radiant temperature [2]. For example, in the conditioned indoor, a low relative humidity can cause overcooling while a high humidity can encourage the growth of mold, fungi, and the procreation of mites and bacteria, which can eventually lead to health problems [3–5]. Besides, while the low air speed is defined as airless, in high speed, it is felt as windy and uncomfortable [2].

The indoor thermal comfort can be improved by the use of the ventilation in the building. An underfloor air distribution (UFAD) system as shown in Fig. 1 is a mechanical ventilation system that uses the space in between the floor slab and the raised access floor to deliver conditioned air to the occupied zone directly from the underfloor by using

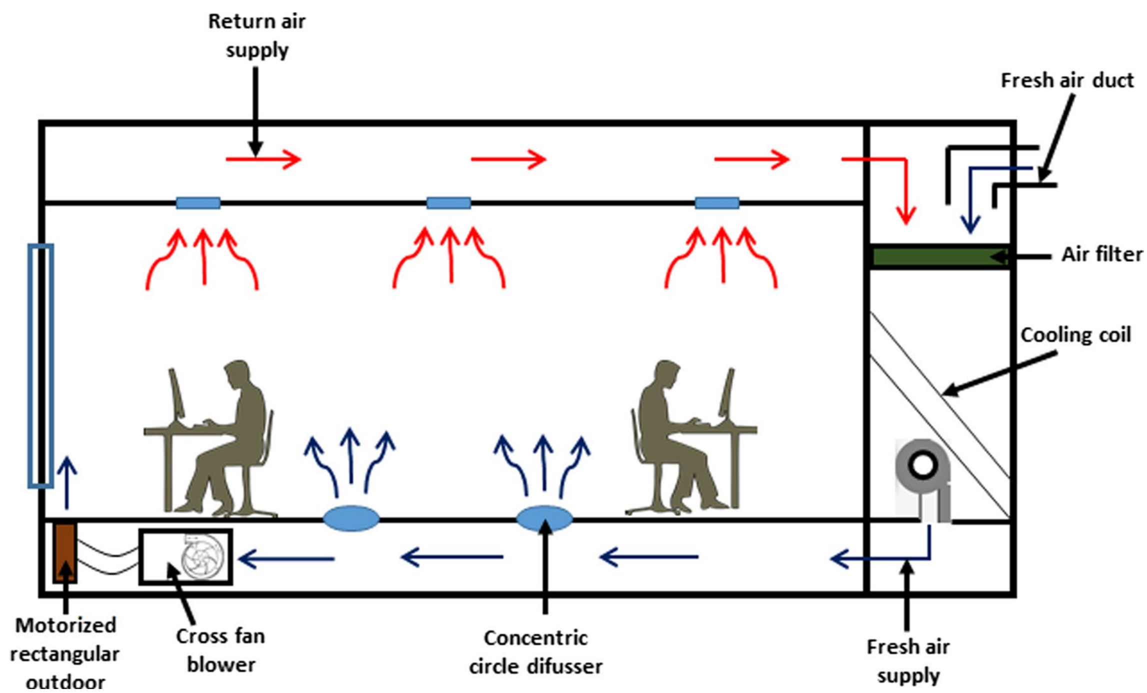

Fig. 1 Schematic diagram of UFAD systems at the internal office spaces redrawn and modified based on Ref. [26]

the floor diffusers [6–8]. This system is designed to provide a thermally comfortable environment as well as air free of contaminants. The concept of air distribution in the UFAD system is different from the other conventional ventilation systems, where this system reduces the supply airflow rate under the floor and causes the air in the upper zone (near the ceiling) to have a higher temperature compared to the supply air outlet at the underfloor [9]. In the conditioned indoor, thermal plumes generated by the heat sources make the conditioned air absorb the heat and humidity. With the ability of this system to lower the supply airflow rate under the floor, the heated air and also air contaminants will move to the upper zone, causing the air in the occupied zone to become cool and fresh [9, 10].

Compared to overhead air distribution (OHAD) system, UFAD system has its own unique features such as a stratified environment and a connection to personal environment control [8, 11]. Customers during the past decades have shifted their focus to this innovative design due to its advantages over an OHAD system from optimized thermal comfort and architectural flexibility to improve indoor air quality (IAQ) and energy efficiency [8, 12, 13]. The advantage of the UFAD system is its suitability for the development of intelligent buildings, which require a more fresh, effective and comfortable indoor environment [13]. However, the application of the UFAD system is obstructed by the gap in the fundamental understanding of these technologies, where an improper design of the UFAD system can lead to a hazardous situation, which causes various degrees of discomforts or illnesses. For example, Alajmi et al. [9] highlighted that unnecessary supply air may degrade the performance of a UFAD system, resulting in a poor predicted mean vote (PMV) rating. Thus, it is important to study or analyze the effectiveness of the UFAD system in terms of thermal comfort before using this system in the building.

However, literature review reveals that studies on thermal comfort and air contaminant levels using local mean age (LMA) value of an UFAD system in a high-rise building located in the tropics are rare, and therefore, the aim of the current work is to investigate the thermal comfort and air contaminant levels using LMA value in an office space located in a high-rise building through objective field measurements and computational fluid dynamics (CFD) approaches.

2 Background Theory Relevant to the Current Work

In the present work, a mathematical model developed and published by Ho et al. [14] is applied to investigate the thermal comfort and air contaminant levels in the examined areas. These governing equations obtained from Ref. [14] can be used to describe the fluid flow, heat and mass transfer phenomena for a given space. In order to evaluate thermal comfort and the removal of contaminants, the air velocity, temperature, airflow pattern and relative humidity are needed to be identified. These can be determined by solving the system of governing equations for the conservation of mass, momentum and energy. In the present research, the steady-state incompressible flows of air have been taken into consideration. The general governing equations applicable in the present work can be found in the Ho et al.'s study [14].

2.1 Numerical Solution Procedures

In the present research, the governing equations and the boundary conditions have been solved by using the finite volume method (FVM). Note that FVM is a numerical analysis approach for obtaining solutions to a wide variety of engineering problems. Because the engineering problems have become more complex compared to the last decades, it is necessary to find approximate numerical solutions to solve the problems rather than using the exact solutions. In the current research paper, the indoor air temperature may vary in the examined areas so that it possesses an infinite number of unknowns. By using the FVM, the computational domain is discretized into many small elements, and by expressing the unknown variables in the terms of assumed approximating functions within each element. At this stage, a computer software FloEFD is used to model the UFAD system and to determine the behavior of the air contaminants using FVM [14].

2.2 Comfort Metrics Relevant to the Present Study

One of the methods of assessing the thermal comfort is to use the PMV and PPD [15]. The PMV is a parameter for evaluating the average thermal sensation of a large group of occupants based on the combination of four physical variables (relative humidity, air temperature, mean radiant temperature, and air velocity) and two personal variables (metabolic rate and clothing insulation). However, PPD is the mathematical function of the PMV used to describe the expected percentage of people dissatisfied in a given thermal environment [16]. In other words, the PMV and PPD are measures of the likely responses of people regarding indoor thermal comfort (whether it is too hot or too cold). For the proper HVAC design in the seasonal countries, the PMV result should be close to zero [15]. However, since Malaysia is a hot and humid country, the PMV result should be close to − 1 [17]. Besides, the PPD of less than 20% is good since the consideration of satisfying 80% of occupants is at a satisfactory level [15]. The details of the PMV and PPD can be found in references [15, 18, 19].

The LMA represents how fast the fresh supply air can be delivered to a specific point within the space [20, 21]. Generally, LMA can be defined as the average lifetime for fluid to flow from the selected inlet to the selected reference point by taking both the velocity and the diffusion into account. The LMA values at the exhaust normally are expressed with respect to the nominal time constant. In fact, the individual LMAs (i.e., local mean age in second) at the exhaust indicate the interval of the supply air residing in the space, where the higher the value of the individual LMAs, the longer time the supply air takes to reside in the space and in other words, the longer time it takes to remove the air contaminants in the indoor space. Besides that, the local mean age is decreased as the total volume flow rate is increased [22].

Note that the gas concentration history at a location of a space is recorded as a function of time and the LMA of air is calculated from the quotient of the integral of the concentration of the tracer gas and its initial concentration at time t_o. Hence, the LMA, τ_p is given by Eq. (1) [21]:

$$\tau_p = \frac{\int_{t_o}^{\infty} C_t \mathrm{d}t}{C_{t=t_o}} \quad (1)$$

where t is the time and $C_t = t_o$ is the initial tracer gas concentration at time $t = t_o$ (in cm^3 m^{-3} of tracer gas). If the space has a total volume of V_R, the tracer gas volume V_I to be released into the room is given as Eq. (2) [21]:

$$V_I = C_{t=t_o} \cdot V_R. \quad (2)$$

3 Research Methodology

In the present research, the study is conducted on the 33rd, 34th and 35th floors of the high-rise building (name to be kept anonymous) located in Kuala Lumpur, Malaysia. As shown in Table 1, the examined areas for each floor are divided into several zones, where these zones include meeting room, office, and pantry.

Table 1 Examined areas

Floor	Zone	Area
All examined floors	1	Office
	2	Office
	3	Office
	4	Office
	5	Office
	P	Pantry
33rd floor (south side)	M1	Meeting room
33rd floor (north side)	M3	Meeting room
	M4	Meeting room
34th floor (north side)	M6	Meeting room
34th floor (south side)	M7	Meeting room
	M8	Meeting room
35th floor (south side)	M9	Meeting room
	M10	Meeting room

All of the floors in the examined building are conditioned by chilled water system, and each floor is subjected by five air handling units (AHUs), where these AHUs are handling fresh air and distributing it into the indoor space with the help of the UFAD system. Note that the technical information includes the specifications of the UFAD system and as-built drawings of the examined floors are given by the building's operation and maintenance team.

The inspection of the target building is done to obtain the data of the number of occupants and their activities, the quantity of the equipment and their heat gain, and the quantity and energy usage of the lighting. These data can be used as the input (boundary condition) in the simulation, which will be mentioned in Sect. 3.2. Note that the simulation is only done in areas on the 34th north floor, where the simulation result will be compared with the fieldwork result.

3.1 Fieldwork

In order to obtain the data of the PMV and PPD on each examined area, the indoor air conditions including temperature, relative humidity, velocity, and MRT as well as clothing insulation value and metabolic rate of the occupants must be known. By putting these data in the relevant PMV and PPD formulas in Excel spreadsheet, the PMV and PPD values will be calculated automatically.

The indoor air temperature, relative humidity, air velocity, and MRT for the examined areas are measured and recorded by using appropriate measuring devices that meet the ANSI/ASHRAE Standard 55-2013 [18]. The clothing insulation and the metabolic rate of the occupants are determined based on the observation made in the examined areas, in which occupants have worn light clothes and are sedentary.

A vapor test was conducted on 34th north floor in order to analyze the effect of the airflow pattern of the UFAD system on the indoor thermal comfort. In this test, two different types of diffusers which were the normal circular diffuser and the circular swirl diffuser were compared in order to find out which diffuser could provide better thermal comfort to the occupants. This test was done by observing the smoke trails of these diffusers and then compared them with each other. Note that the smoke was produced by the vapor meter.

3.2 Theory Relevant to the Uncertainty Analysis

A complete set of realistic approximated field data to represent the performance characteristics of the UFAD system is needed in this research. For this purpose, the uncertainty analysis of field data for each examined floor is performed in order to examine whether or not the errors present in the data obtained by the fieldwork are at a satisfactory level. The bias uncertainty for each examined floor can be obtained by using the following formula [23, 24]:

$$\text{Bias Uncertainty}_{(n)} = \frac{R_{\text{zone}(n)}}{2} \tag{3}$$

where R_{zone} is the range value (difference between the maximum value and minimum value) of the field data for the examined zones for each floor and (n) is the floor number. Note that the uncertainty obtained can be converted into the percentage uncertainty [25]:

$$\text{Percentage Uncertainty}_{(n)} = \left(\frac{\text{Bias Uncertainty}_{(n)}}{\bar{R}_{\text{zone}(n)}}\right) \times 100 \tag{4}$$

where $\bar{R}_{\text{zone}}$ is the average value of the field data for the examined zones for each floor. By using the same concept and equations as mentioned above, the uncertainty analysis between the simulation results and fieldwork results also can be done.

3.3 Simulation

In the present research, the examined areas in 34th north floor including the meeting room, pantry and office are modeled using the SolidWorks, where the dimensions of these areas as shown in Table 2 are based on the floor blueprint from the previous study [26]. After modeling using the SolidWorks, these models are imported into FloEFD in order to simulate the UFAD system with the use of the circular swirl diffuser in terms of indoor air temperature, relative humidity, air velocity, airflow pattern, and LMA of the air. It is pertinent to mention that the simulation result will be compared with the fieldwork results by using uncertainty analysis.

In order to simulate the examined areas in terms of different airflow patterns, two different types of diffusers which are normal circular diffuser and circular swirl diffuser are used in the present research. It is pertinent to mention that some assumptions have been made during the simulation, which are:

- All diffusers have the same ambient condition and the volumetric flow rate.
- All windows are covered and the solar heat energy is absent.
- The geometry of the floor is modeled as a cuboid.

Table 2 Dimensions of the examined areas

Area	Width (m)	Length (m)	Height (m)
Meeting room	2	5	3
Pantry	7	13	3
Office	7	13	3

- The circular swirl diffuser is represented by a circular fan.

This simulation can be done after setting the geometry and the boundary conditions. The relevant data are set as the boundary conditions as shown in Table 3. Note that the information obtained regarding the number of occupants and their heat gain, the quantity of the equipment and its heat gain, and the quantity of light and its power usage in each area is extracted from the design library of FloEFD and directly inserted into the model. The details of heat sources in the examined areas are shown in Table 4.

Table 4 Surface heat sources in the examined areas

Parameter	Heat load per unit or person (W)	Meeting room	Pantry room	Office
Total number of occupants	85 (Seated occupant)	5	3	6
Total number of computers	100	2	3	20
Total number of projectors	50	1	2	0
Total number of ceiling lights	120	2	3	3

3.4 Limitations

The present research is bounded by some limitations, which could probably compromise its precision. The accuracy of the data depends on the measurement skill of the researchers and the accuracy of the relevant equipment. For example, the equipment used for measuring the air velocity in the present research might also not highly sensitive to the changes in the indoor conditions, where this equipment loses its sensitivity at air velocities less than 0.5 m/s, causing inaccurate results [27]. Note that the hot wire anemometer was used to measure the air velocity in the present research.

For the simulation, the airflow rate of each diffuser is obtained by dividing the average total volume flow rate by the total number of diffusers. However, in a real-life condition, the airflow rates for the diffusers are not constant, depending on the performance of the diffuser itself. Therefore, the simulation results might be inadequate. Besides, in order to simplify the simulation, a series of assumptions have been made, namely the geometry of the floor is modeled as a cuboid, all windows are covered and the solar heat energy is absent. Thus, the simulation results may have a slight difference with the fieldwork results.

4 Results and Discussion

The fieldwork results and their uncertainty analyses as well as the vapor test in terms of the airflow pattern inspection will be discussed first in Sect. 4.1, while the simulation results will be elaborated later in Sect. 4.2. Since the simulation is only done for the 34th north floor, the uncertainty analysis between the simulation result and fieldwork result

Table 3 Boundary conditions in the simulation

Parameter		Meeting room	Pantry room	Office
Opening (doors)		Pressure driven	Pressure driven	Pressure driven
Supply air temperature (°C)		22	22	22
Supply air relative humidity (%)		70	70	70
**Diffuser*				
Quantity		3	6	12
Dimension: diameter (m)		0.34	0.34	0.34
Volume flow rate (cfm) or (m^3/s)		395.00 [0.19]	395.00 [0.19]	395.00 [0.19]
**Square exhaust*				
Quantity		2	5	5
Dimension: length × width (m)		0.4 × 0.4	0.4 × 0.4	0.4 × 0.4
Velocity (m/s)		0.5	0.5	0.5
**Pressure exhaust (square exhaust and front wall)*				
Quantity	Square exhaust	1	1	1
	Front wall	1	1	0
Static pressure: front wall (pa)		101,325	101,325	0
Environmental pressure: square exhaust (pa)		101,325	101,325	101,325

*For both normal circular diffuser and circular swirl diffuser is performed for the 34th north floor only. This will be discussed in Sect. 4.3.

4.1 Fieldwork

In the present research, the fieldwork was conducted on the 33rd, 34th and 35th floors. However, as for the vapor test, it was conducted only on the 34th north floor, where the test was done by inspecting the smoke trails produced by the diffuser of the UFAD system.

4.1.1 Fieldwork Results

Based on Table 5, the PMV and PPD results have been computed; they indicated that the range of PPD falls in between 27.4 and 67.5%, while the average PMV is approximately − 1.5 for each examined area as shown in Tables 5. The results obtained clearly show that the occupants perceived the examined areas to be too cool. For a proper design of the HVAC system in a seasonal country, the PMV result should be approximately equal to 0. However, since Malaysia is a hot and humid country, the PMV result should be close to − 1. The obtained PPD result indicates that about more than half of the occupants have dissatisfied with the indoor conditions in the examined building. This is because the obtained average indoor air temperature, relative humidity, and air velocity have not met the requirements of the Malaysia Standard [28].

The average indoor air temperature and relative humidity are in the ranges of 21.0–23.4 °C and 61.5–72.3%, respectively. However, Malaysia Standard MS 1625:2014 [28] has recommended that the limit of indoor relative humidity is between 50 and 70%, while the indoor air temperature is between 23 and 26 °C. Thus, the result of indoor air temperature is considered reasonable, although the minimum indoor temperature is slightly lower than the recommended limit, while the maximum indoor temperature is in the range of the recommended limit. As for the indoor relative humidity, the maximum value is above the recommended limit. Furthermore, most of the indoor air velocities, for each examined area, are below the recommended limit required by Malaysia Standard MS 1625:2014 [28]. The recommended limit is in the range of 0.15–0.5 m/s for buildings in the tropics, while the obtained indoor air velocity is in the range of 0.01–0.12 m/s. The result indicates that the ventilation rate is clearly inadequate and the occupants might feel uncomfortable in the workplace.

The MRT for all examined areas is generally at 23.3 °C, where it clearly shows that the MRT is constant at all examined areas and is in good agreement with the average air temperature. Note that the MRT is assumed to be equal to the average air temperature based on the assumption that surrounding indoor surfaces have uniform temperatures and radiation heat fluxes [29].

4.1.2 Uncertainty Analysis of Fieldwork Results

As shown in Table 6, the percentage uncertainties for the average air velocity, PMV, and PPD are too high, which clearly show that some errors occur in these field data. The percentage uncertainties for the PMV and PPD are in the ranges of 3.97–25.00% and 6.68–39.76%, respectively; these results suggested that nonuniform indoor temperature and relative humidity distributions are present during the fieldwork.

In reality, the outdoor temperature varies for every hour, where this variation may have the impact either directly or indirectly on the indoor relative humidity distribution. Furthermore, the poor performance of the existing diffusers used in the UFAD system in the aspect of carrying out or removing the latent heat load created by the occupants and the sensible heat load generated by the equipment in the examined areas may cause the uneven cooling and the uneven air distribution.

In addition, the percentage uncertainty of the average air velocity is in the range of 34–100% due to the use of different numbers of diffusers for every examined area. Besides, the equipment used to measure the air velocity may not be highly sensitive to the changes in the indoor conditions as mentioned in Sect. 3.4, causing an inaccurate measurement.

However, the percentage uncertainties for the PMV and PPD in the 33rd Floor (north side) and 34th floor (north side) are much lower compared to the other zones due to the better mixing and an even air distribution in these zones. Besides, the percentage uncertainty for MRT is zero due to the reason that the MRT is almost constant at 23.3 °C.

4.1.3 Vapor Test Result (The Airflow Pattern Inspection)

Table 7 shows the result of the vapor test between the circular swirl diffuser and the normal circular diffuser. As observed, the circular swirl diffuser allows a better mixing of air, as the smoke trails form a vortex-type structure in the air. On the other hand, the smoke trails of the normal circular diffuser directly go up without much distribution. As the normal circular diffuser does not distribute the air evenly into the occupied space, there may be a risk of air stagnancy and infection [30].

4.2 Simulation Results

In the present research, the simulation process was done in the examined areas on 34th north floor in terms of airflow pattern as well as the indoor air temperature, relative humidity, air velocity, and LMA of the air. The details of the results will be elaborated in the next few subsections.

Table 5 Fieldwork results (thermal sensation data)

Zone	Average air temperature (°C)	Mean radiant temperature (°C)	Average air velocity (m/s)	Average relative humidity (%)	Clothing (clo)	Metabolic rate (met)	PMV	PPD
33rd floor (north side)								
1	21.7	23.3	0.04	62.5	0.45	1	− 1.45	48.3
2	22.1	23.3	0.04	62.5	0.45	1	− 1.34	42.8
3	22.0	23.3	0.06	62.6	0.45	1	− 1.37	44.0
4	22.0	23.3	0.04	63.0	0.45	1	− 1.36	43.8
5	22.1	23.3	0.03	63.2	0.45	1	− 1.36	43.4
P	21.8	23.3	0.09	62.7	0.45	1	− 1.42	46.9
M3	21.8	23.3	0.02	61.5	0.45	1	− 1.43	47.4
M4	22.1	23.3	0.01	65.0	0.45	1	− 1.34	42.3
33rd floor (south side)								
1	21.9	23.3	0.02	65.5	0.45	1	− 1.38	44.6
2	21.8	23.3	0.03	65.0	0.45	1	− 1.40	45.6
3	21.8	23.3	0.04	64.1	0.45	1	− 1.41	46.0
4	21.9	23.3	0.03	64.0	0.45	1	− 1.37	44.3
5	22.3	23.3	0.07	65.2	0.45	1	− 1.29	40.1
P	22.4	23.3	0.05	65.4	0.45	1	− 1.26	38.4
M1	22.7	23.3	0.02	65.8	0.45	1	− 1.19	34.7
34th floor (north side)								
1	21.7	23.3	0.01	67.7	0.45	1	− 1.40	45.9
2	21.9	23.3	0.05	66.1	0.45	1	− 1.38	44.8
3	21.7	23.3	0.04	65.8	0.45	1	− 1.43	47.1
4	21.5	23.3	0.04	65.6	0.45	1	− 1.48	50.2
5	21.3	23.3	0.03	65.9	0.45	1	− 1.52	52.2
P	21.4	23.3	0.09	65.2	0.45	1	− 1.49	50.8
M6	22.1	23.3	0.01	69.7	0.45	1	− 1.30	40.6
34th floor (south side)								
1	22.4	23.3	0.05	68.0	0.45	1	− 1.25	37.9
2	21.8	23.3	0.06	68.3	0.45	1	− 1.38	44.9
3	22.2	23.3	0.05	66.7	0.45	1	− 1.31	40.8
4	21.5	23.3	0.03	67.1	0.45	1	− 1.47	49.2
5	21.0	23.3	0.05	66.4	0.45	1	− 1.59	56.0
P	20.1	23.3	0.07	68.3	0.45	1	− 1.81	67.5
M7	20.7	23.3	0.12	72.3	0.45	1	− 1.66	60.0
M8	21.4	23.3	0.01	71.2	0.45	1	− 1.46	49.0
35th floor (south side)								
1	23.4	23.3	0.04	65.0	0.45	1	− 1.03	27.4
2	23.0	23.3	0.04	64.1	0.45	1	− 1.14	32.3
3	22.3	23.3	0.06	63.8	0.45	1	− 1.29	39.8
4	21.5	23.3	0.07	65.6	0.45	1	− 1.48	49.8
5	21.0	23.3	0.05	66.3	0.45	1	− 1.58	55.6
P	20.4	23.3	0.08	68.8	0.45	1	− 1.72	63.1
M9	21.4	23.3	0.05	68.1	0.45	1	− 1.48	50.2
M10	22.1	23.3	0.08	69.9	0.45	1	− 1.31	40.9

4.2.1 Simulation of the Airflow Pattern

Figures 2 and 3 show the 3D streamlines for the circular swirl diffuser and normal circular diffuser, respectively, in the meeting room. Note that the simulation of the airflow pattern as shown in these figures also acts as the representative of the simulation airflow pattern from other areas. As shown in these figures, the streamlines begin at nine

Table 6 Uncertainty analysis of fieldwork results for different areas

Area	Average air temperature	MRT	Average air velocity	Average relative humidity	PMV	PPD
Bias uncertainty						
33rd floor (north side)	0.2	0	0.04	1.75	0.05	3
33rd floor (south side)	0.45	0	0.03	0.90	0.11	5.65
34th floor (north side)	0.40	0	0.04	2.25	0.11	5.80
34th floor (south side)	1.15	0	0.06	2.95	0.28	14.80
35th floor (south side)	1.50	0	0.02	3.05	0.35	17.85
Average value	0.74	0	0.04	2.18	0.18	9.42
Percentage uncertainty (%)						
33rd floor (north side)	0.91	0	96.97	2.78	3.97	6.68
33rd floor (south side)	2.03	0	67.31	1.38	8.28	13.47
34th floor (north side)	1.85	0	103.70	3.38	7.70	12.24
34th floor (south side)	5.38	0	100.00	4.30	18.78	29.21
35th floor (south side)	6.85	0	34.04	4.59	25.02	39.77
Average value	3.40	0	80.40	3.28	12.75	20.27

different starting points on the surface of the diffuser. The streamlines' colors represent the air speeds.

It can be observed from Fig. 2 that the equitable distribution of air can be found in most parts of the meeting room, and the air distribution is in the vortex-type airflow pattern. Based on the result, it has been shown that the air discharged from this circular swirl floor diffuser allows rapid mixing of the indoor air in the meeting room, and the air goes up smoothly. While going up to a certain level, the supply air will lose speed and spread wider. As the driving force depletes, it will easily be influenced by the pressure at the door region and the exhaust region. Because of the environmental pressure at the door region, the airflows tend to flow toward the door at an increasing speed. However, when the door is in the closed condition, the air normally will tend to flow toward the exhaust region. In addition, a small part of the airflow is affected by the buoyancy effect in the warm region closer to the occupants and office components.

Figure 3 shows the airflow pattern of the normal circular diffuser, which indicates a poor performance in distributing the air evenly to the indoor space. In fact, the air distribution through this diffuser is directed into the meeting room like a jet without proper infiltration and distribution. While the air is going up to a certain height, the supply air will lose speed and spread wider. As the driving force is exhausted, it will easily be influenced by the higher pressure at the door region. It is the same for the circular swirl diffuser, and a small part of the airflow of the normal circular diffuser is affected by the buoyancy effect in the warm region, which is closer to the occupants or office components. However, when compared with the circular swirl diffuser, the normal circular diffuser will cause air stagnancy created in the higher region (near the ceiling), and therefore, it causes the occupants to feel stuffy after long hours of working.

Based on the simulation result, it clearly shows that the circular swirl-type diffuser is the most suitable one to be used with the UFAD system due to the high efficiency in the removal of air contaminants.

4.2.2 Simulation of the Meeting Room

Figures 4, 5, 6 and 7 show the simulation results in the meeting room. Based on the indoor air temperature result in Fig. 4, when the conditioned air is close to the heat loads, the color of the temperature area varies from 22 °C to about 24 °C. On top of that, the temperature contour has shown that a layer of stratification is formed in the meeting room with the cool air below and the warm air above, which is closer to the ceiling in the occupied zone.

As shown in Fig. 5, the relative humidity is relatively high and almost exceeds 70% at maximum level. On average, it falls between 67 and 69%, while it drops to 63% near the heat loads. This clearly shows that the existence of heat loads will affect the relative humidity result. From Fig. 6, a higher velocity is found at the door region because of the airflow pattern influenced by the environment pressure at the door part. In addition, it has shown that the underfloor diffusers are able to distribute the air evenly to the meeting room and have successfully removed the latent heat load, which is generated by the occupants and the sensible heat load emitted by the equipment used in this area. It is noted that regions of low air velocities are found below the chairs and the conference table because of the obstruction.

Figure 7 shows that LMA of air is higher around the occupants in the range of 133–144 s in the meeting room. A high LMA value indicates that the ventilation effectiveness around the occupants at the meeting room is low and not fresh, and also the concentration of the air contaminants is

Table 7 Airflow patterns of the circular swirl and normal circular diffusers

Circular Swirl Diffuser	Normal Circular Diffuser
	
	
	
	

Table 7 (continued)

Circular Swirl Diffuser	Normal Circular Diffuser
	
	

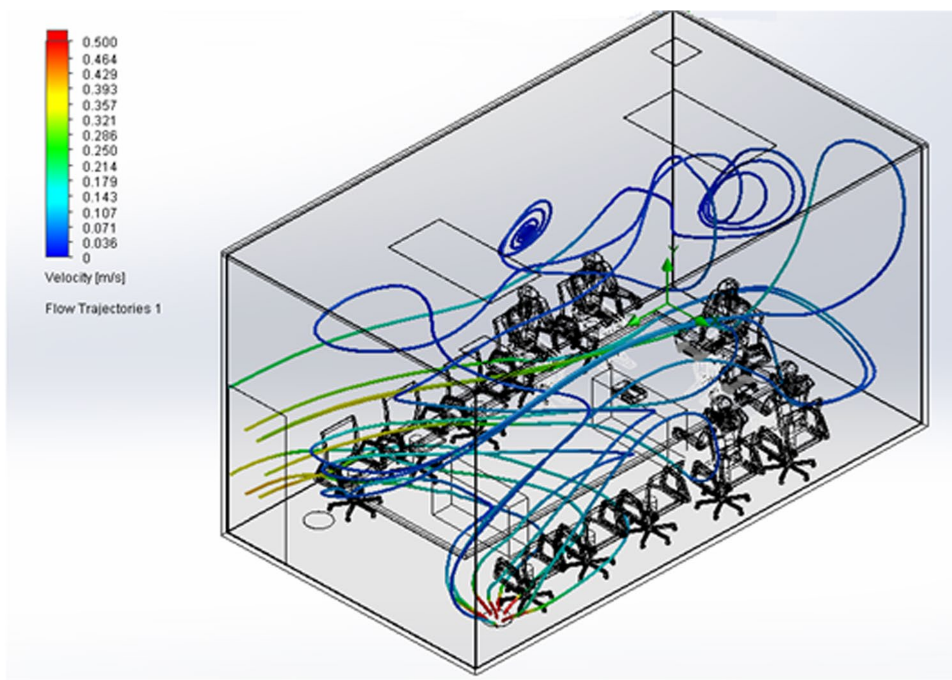

Fig. 2 Airflow pattern of the circular swirl diffuser

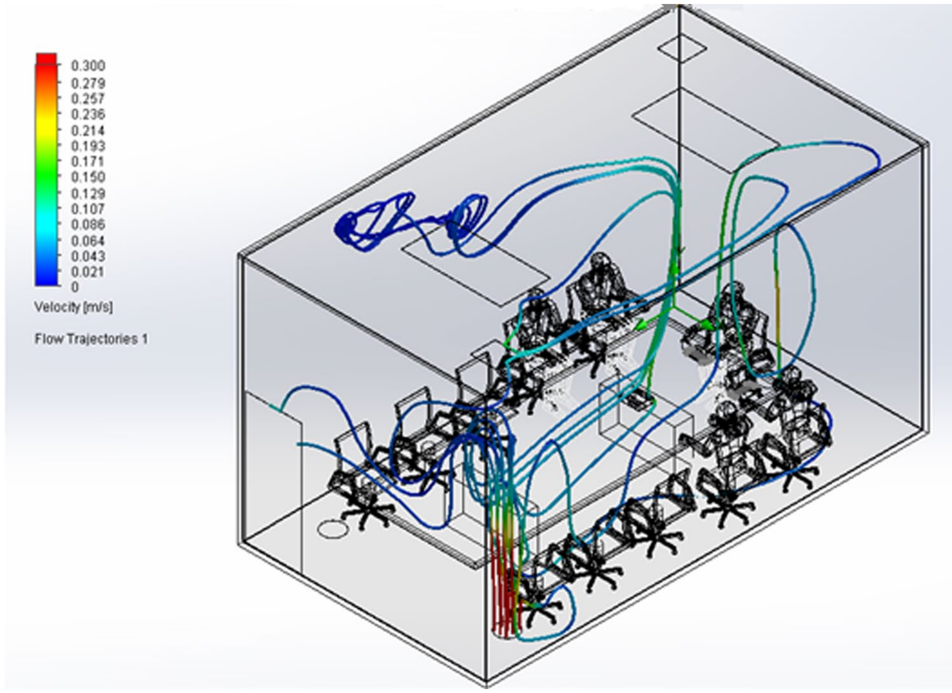

Fig. 3 Airflow pattern of the normal circular diffuser

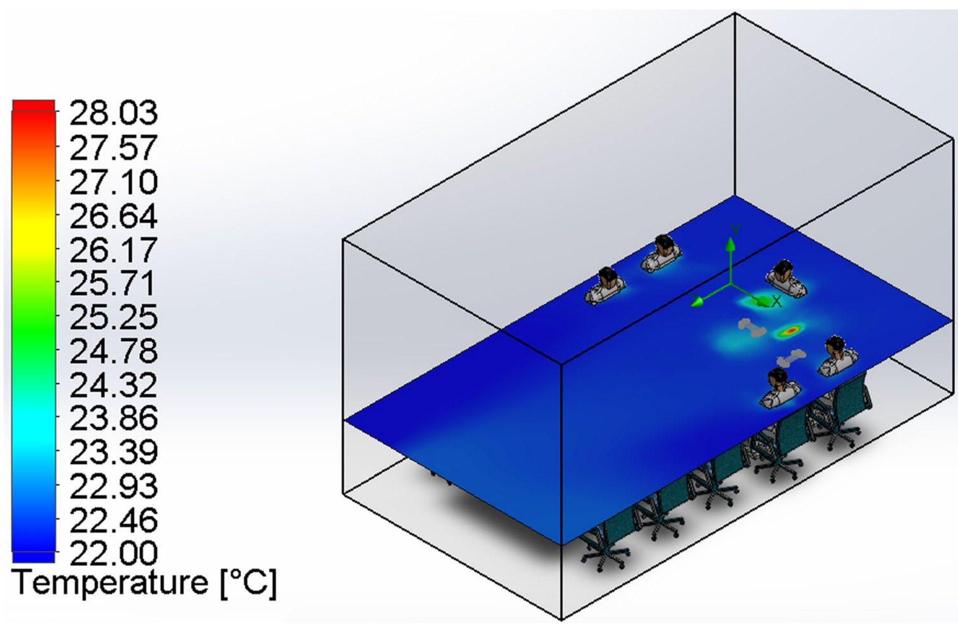

Fig. 4 Meeting room temperature contour isometric view plane at $y=1$ m

high. However, Han et al. [22] show that there is no fixed reference for LMA values because they depend on the size of the room and the number of diffusers.

4.2.3 Simulation of the Pantry

For the pantry, the temperatures in Fig. 8 are almost in the same range as the meeting room, going from 22 °C to about 23.5 °C. The indoor air temperature difference is slightly less than the meeting room because the number of diffusers used in the pantry is twice the number of diffusers used in the meeting room. It is the same for the meeting room, in which the indoor air temperature in the pantry increases near the heat loads. As shown in Fig. 9, the relative humidity in the pantry is higher than the meeting room due to the fact that the room temperatures in the pantry are lower than the meeting room. However, it is similar to the meeting room,

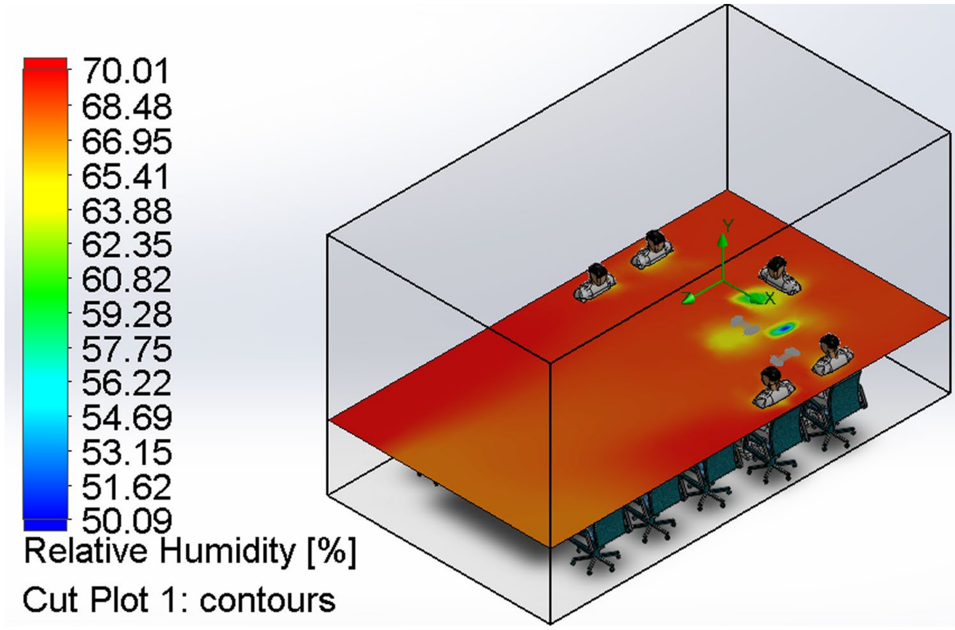

Fig. 5 Meeting room relative humidity contour isometric view plane at $y = 1$ m

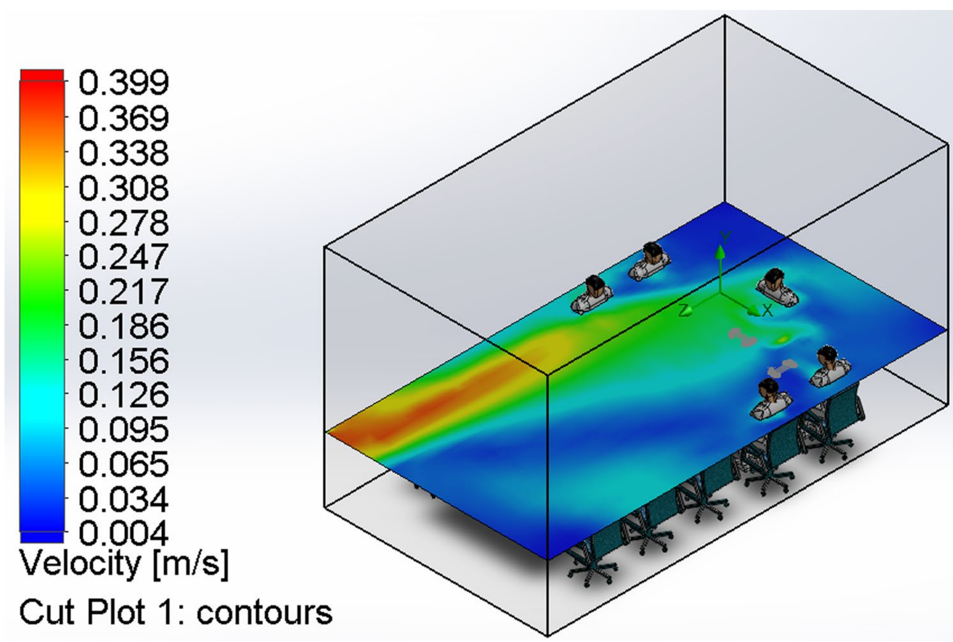

Fig. 6 Meeting room air velocity contour isometric view plane at $y = 1$ m

where the relative humidity in the pantry decreases near the heat loads, which is from 69 to 65%.

The velocity profile in Fig. 10 shows a very nice coverage of air on the left side of the pantry, which clearly shows that the air is distributed evenly to the pantry and successfully removes the latent heat load generated by the occupants and the sensible heat load emitted by the equipment used in this area. This result is also reflected in the simulation result of the LMA as shown in Fig. 11. The LMA on the left side is low, which is 13.5 s. This is because the air velocities are higher near the diffusers because the left side of the pantry is practically empty. However, the ventilation on the right side is insufficient as indicated by the LMA at 176 s, which suggests that the air's age is old.

4.2.4 Simulation of the Office Space

The indoor air temperature difference in the office space is the least compared to the other areas as shown in Fig. 12.

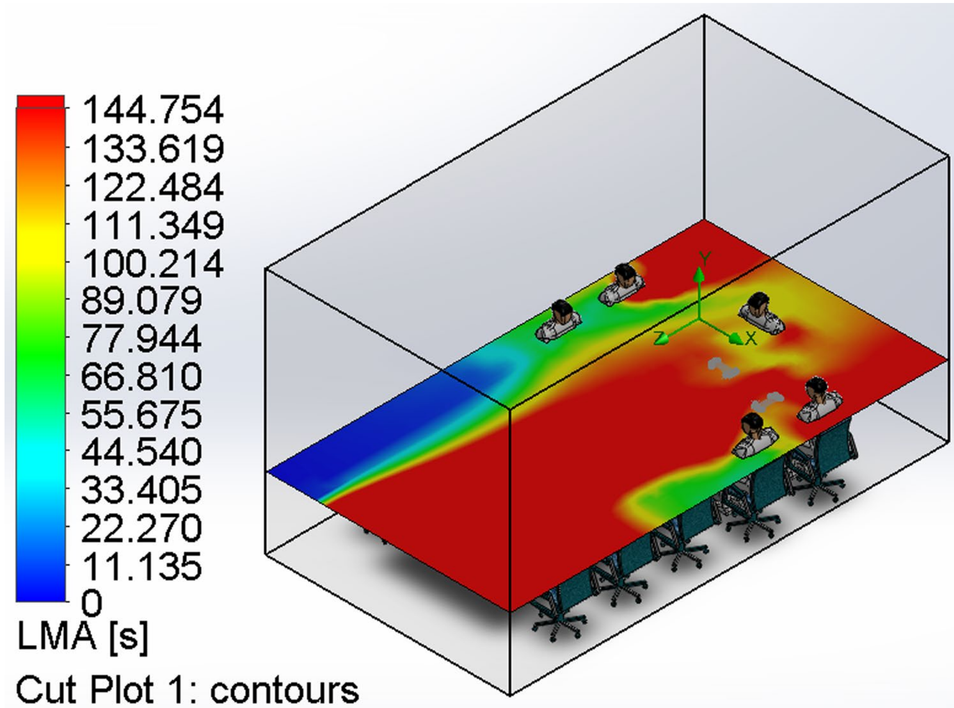

Fig. 7 Meeting room LMA contour isometric view plane at $y=1$ m

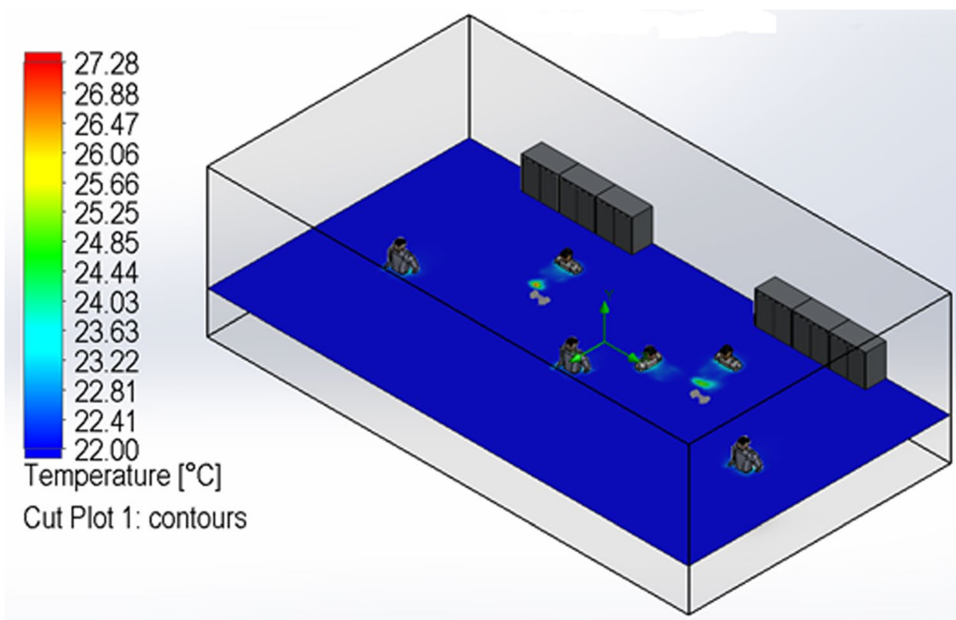

Fig. 8 Pantry temperature contour isometric view plane at $y=1$ m

This is because the number of diffusers used in the office space is more than the other areas. However, it is similar to the other areas and the indoor air temperature in the office increases near the heat loads. Based on Fig. 13, the relative humidity results are relatively high when compared with the other areas, where the result is almost exceeded the threshold of the Malaysia Standard's upper limit of 70%. This is due to the fact that the room temperatures in the office are the lowest among the other areas. Besides, during the daily operation, the cooler air is kept on supply by these 12 diffusers and makes the indoor environment cooler than the meeting room and the moisture content becomes higher. However, it is the same as other areas and the relative humidity in the office decreases near the heat loads, which is 70–60%.

As shown in Fig. 14, the air velocity in the office is quite low when compared with other areas because there are more furniture and devices in the office space. On the

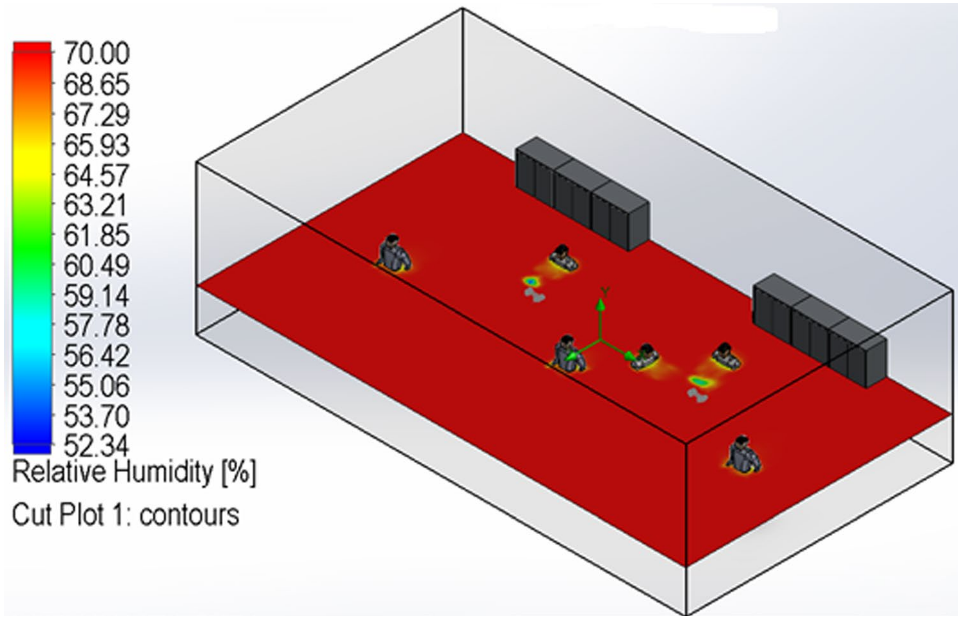

Fig. 9 Pantry relative humidity contour isometric view plane at $y = 1$ m

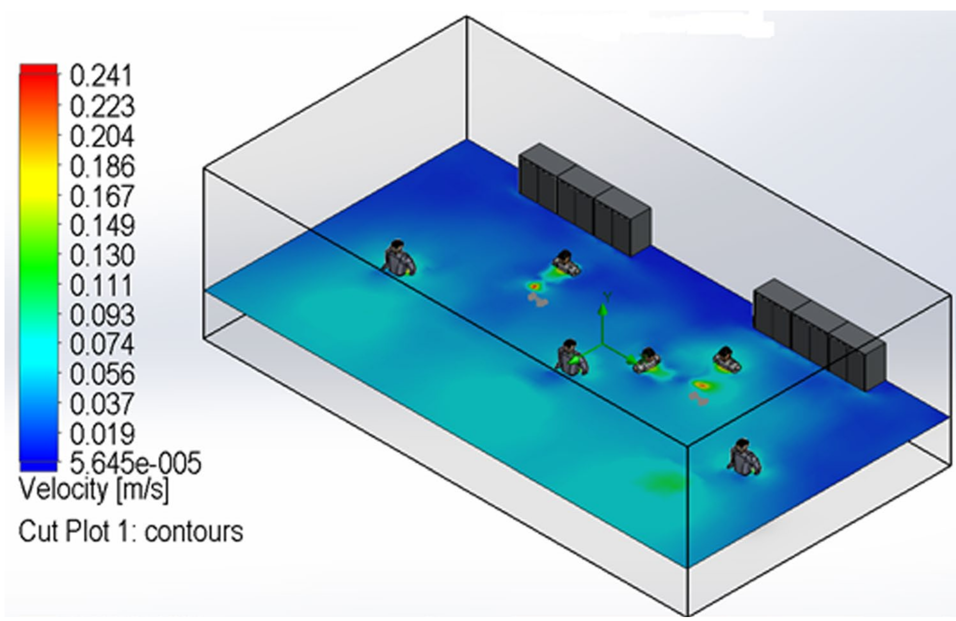

Fig. 10 Pantry air velocity contour isometric view plane at $y = 1$ m

average, the air velocity of the office falls in between 0 and 0.3 m/s. It is evident that from the LMA value at 150 s in general in Fig. 15, the result has indicated that the space has low ventilation effectiveness and the air is not fresh.

4.3 Uncertainty Analysis Between Simulation and Fieldwork Results

From Table 8, the percentage uncertainties obtained for most of the results are below 5%, which clearly show that the simulation result is fairly close to the fieldwork result. However, the percentage uncertainties of the average velocity for every zone are quite high. These errors may occur due to the limitations of the present research as mentioned in Sect. 3.4 earlier.

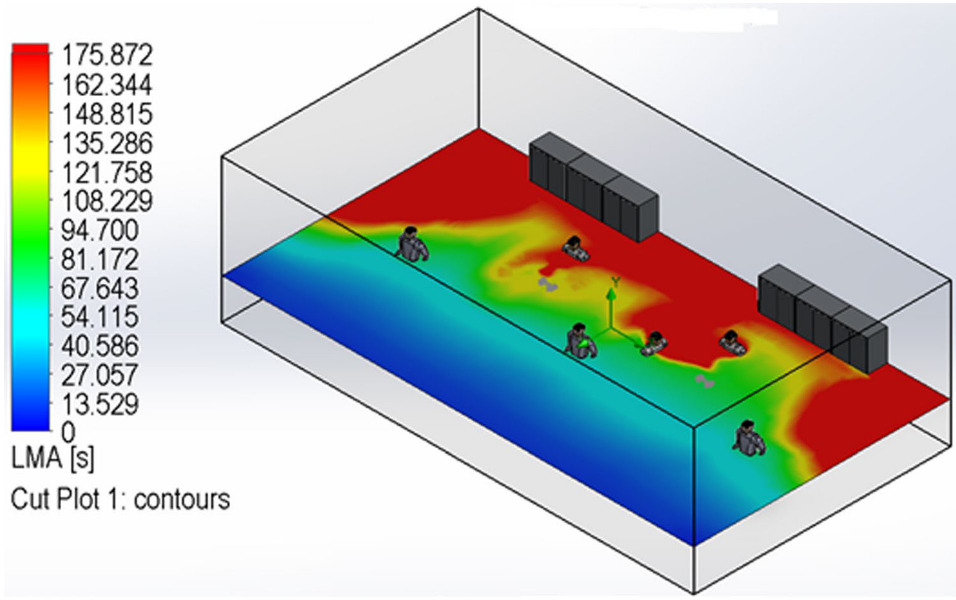

Fig. 11 Pantry LMA contour isometric view plane at $y = 1$ m

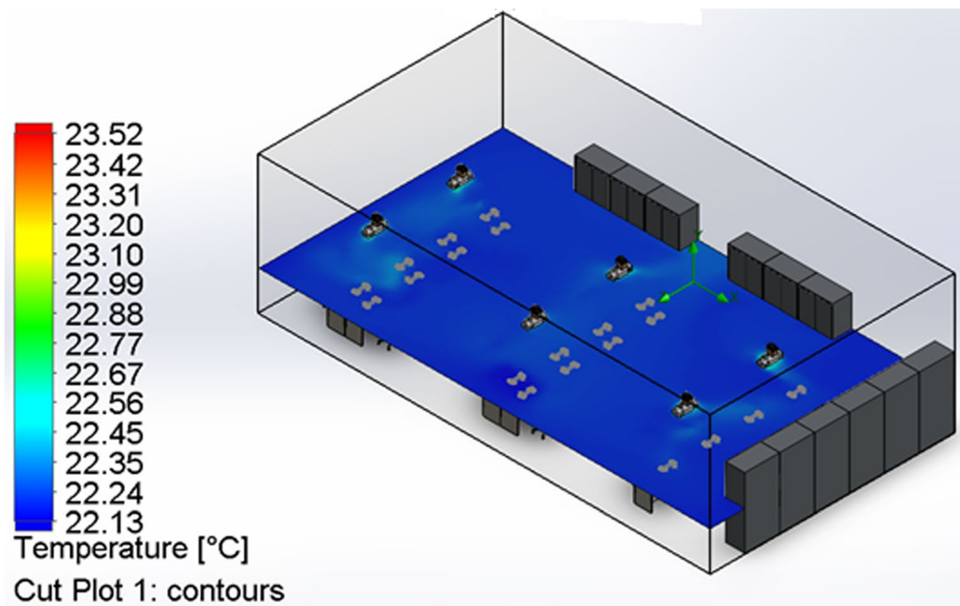

Fig. 12 Office space temperature contour isometric view plane at $y = 1$ m

5 Conclusions

The study of the UFAD system for thermal comfort has been conducted successfully in a high-rise building located in the tropics, where the examined areas were located in the 33rd, 34th, and 35th floors of the building. Based on the fieldwork results, the PMV and PPD results have been computed and have indicated that the range of PPD falls in between 27.4 and 67.5%, while the average PMV is approximately − 1.5 for each examined area. The results obtained clearly show that the occupants have perceived the examined areas to be too cool. The obtained PPD result indicates that about more than half of the occupants are dissatisfied with the indoor conditions in the examined building.

The obtained average indoor air temperature and relative humidity are in the ranges of 21.0–23.4 °C and 61.5–72.3%, respectively. As for the air velocity, the result obtained is in the range of 0.01–0.12 m/s. In addition, the LMA obtained from the simulation shows that most of the examined areas have low ventilation effectiveness and hence the air is not fresh. These results clearly show that the existing UFAD system is unable to provide an adequate thermal comfort to the occupants.

As for the airflow pattern, both simulation and vapor test show that the circular swirl-type diffuser is the most suitable

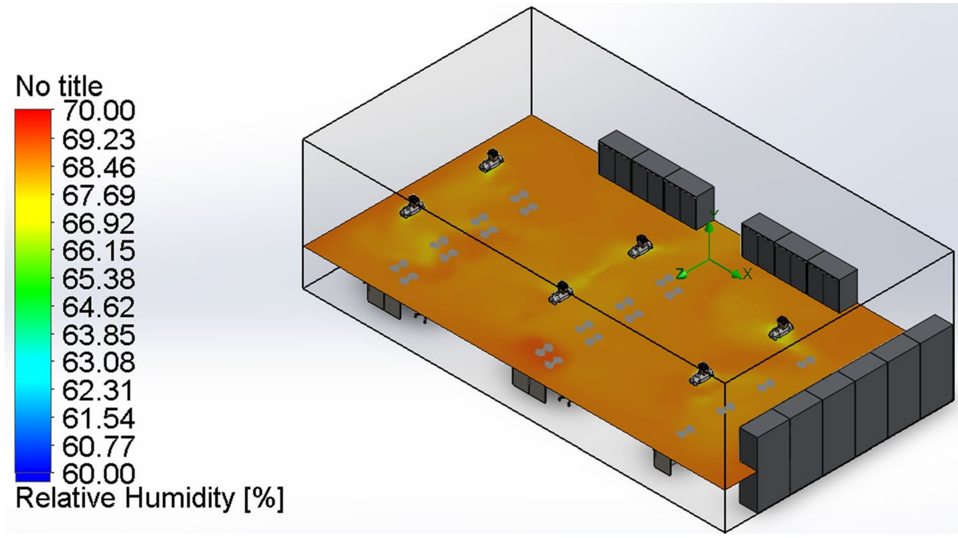

Fig. 13 Office space relative humidity contour isometric view plane at $y=1$ m

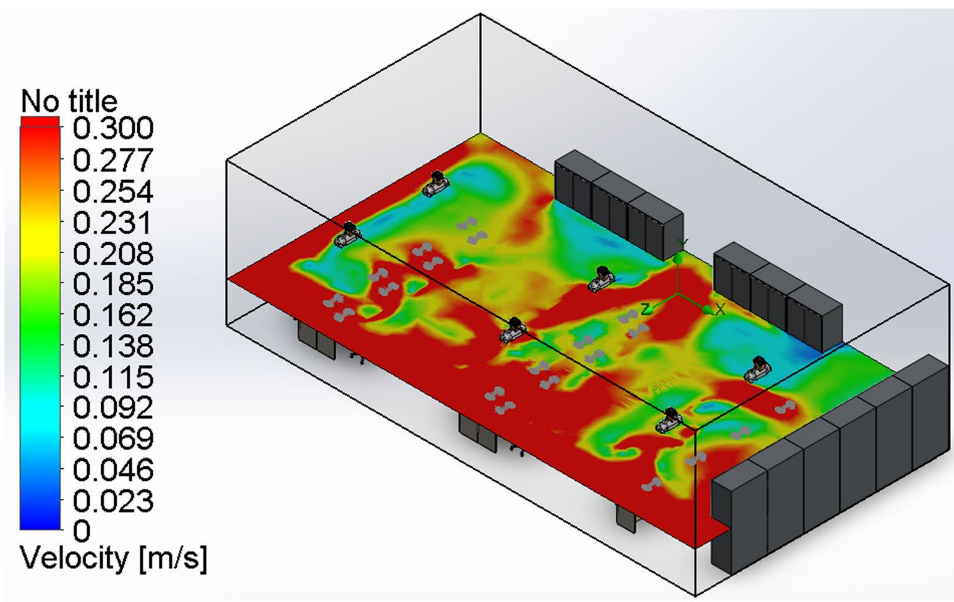

Fig. 14 Office space air velocity contour isometric view plane at $y=1$ m

diffuser to be used with the UFAD system compared to the normal circular diffuser. This is because, when compared with the circular swirl diffuser, the normal circular diffuser will cause an air stagnancy in the higher region (near the ceiling), which eventually makes the occupants feel stuffy after long hours of working.

In short, the data collected in the present work can be used as an important guide for UFAD designs in the tropics. From the vapor test assessment, the airflow pattern produced by the underfloor swirl diffusers is distributed more evenly in the conference room compared to the underfloor normal diffusers. The airflow patterns from the CFD simulations also support the current assessment. It is recommended that the system be modified to improve the air distribution with a solution such as the installation of swirl diffusers with a correct air attack angle. It is also suggested that the air velocity of the space be increased and the latent load be reduced to enhance the comfort level in the space. Implementation of these recommendations will certainly enhance the thermal

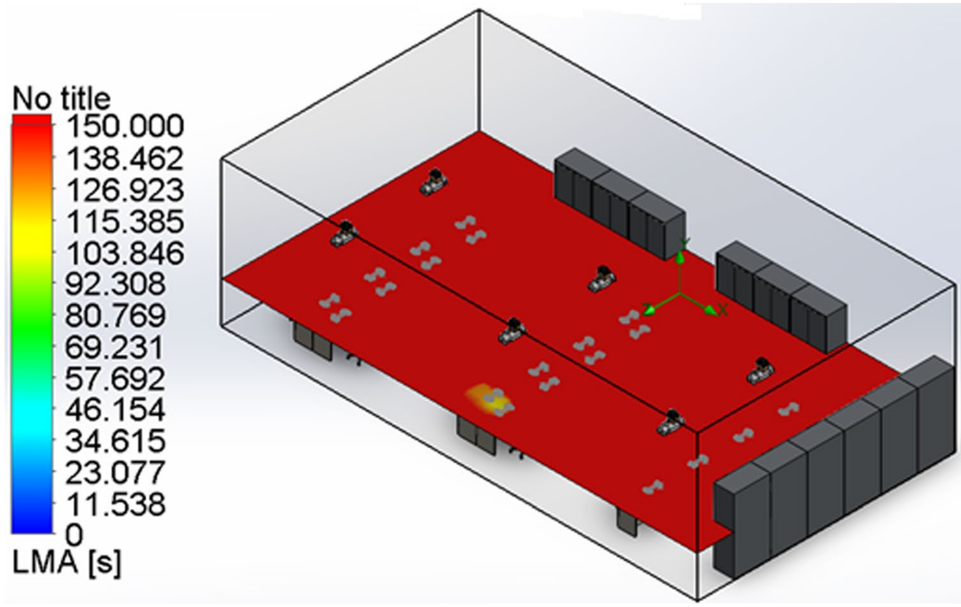

Fig. 15 Office space LMA contour isometric view plane at $y = 1$ m

Table 8 Uncertainty analysis between the simulation and fieldwork results

Zone	Simulation	Fieldwork	Bias uncertainty	Percentage uncertainty (%)
Average air temperature (°C)				
1	22.21	21.70	0.26	1.16
2	22.20	21.90	0.15	0.68
3	22.20	21.70	0.25	1.14
4	22.21	21.50	0.36	1.62
P	22.00	21.40	0.30	1.38
M6	22.34	22.10	0.12	0.54
MRT (°C)				
1	23.30	23.30	0	0
2	23.30	23.30	0	0
3	23.30	23.30	0	0
4	23.30	23.30	0	0
P	23.30	23.30	0	0
M6	23.30	23.30	0	0
Average air velocity (m/s)				
1	0.13	0.01	0.06	85.40
2	0.16	0.05	0.06	52.38
3	0.16	0.04	0.06	60.00
4	0.13	0.04	0.04	52.10
P	0.04	0.09	0.03	42.86
M6	0.03	0.01	0.01	51.22
Average relative humidity (%)				
1	68.52	67.70	0.41	0.60
2	68.41	66.10	1.16	1.72
3	68.41	65.80	1.31	1.94
4	68.52	65.60	1.46	2.18
P	69.93	65.20	2.37	3.50
M6	68.60	69.70	0.55	0.80

comfort conditions and minimize the amount of contaminants in the office space.

Despite some limitations, the present research has given a significant new insight. The findings illustrate the significance of air stagnancy in the occupants' zone, and the LMA values are high in these areas. The future study of the current subject is recommended in order to achieve more accurate results using high-sensitivity devices such as the laser Doppler velocimeter.

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Compliance with Ethical Standards

Conflict of interest The authors declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

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