Influence of Indoor Temperature Variation on Thermal Comfort in University Classrooms in a Tropical Malaysian Setting

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Abstract. Maintaining a thermally comfortable environment is essential for student well-being and academic performance, especially in regions with consistently high ambient temperatures. The research employs a combination of field measurements and Computational Fluid Dynamics (CFD) simulations to analyse thermal comfort parameters under a range of indoor temperature conditions, from 22 °C to 28 °C. Key indicators such as Predicted Mean Vote (PMV), mean radiant temperature, and operative temperature were evaluated in accordance with ASHRAE 55, ISO 7730, and MS 1525 standards. The results show that PMV values remain within the comfort zone (–0.5 to +0.5) when supply air temperatures are maintained between 22 °C and 24 °C, while values exceed +1.0 at 28 °C, indicating increased discomfort. Zone-by-zone analysis revealed that rear classroom zones experienced higher PMV (up to +1.8), and PMV variability exceeded 0.15 in certain zones, suggesting spatial discomfort. The study also identified a linear increase in PMV with air temperature ($R^2 = 0.91$), and the simulated temperatures deviated from experimental measurements by an average of only 0.32 °C. These findings highlight the critical role of precise HVAC temperature control and air distribution in achieving uniform thermal comfort in tropical university classrooms.

1 Introduction

Thermal comfort plays a critical role in shaping indoor environmental quality, particularly within educational spaces where cognitive performance and occupant satisfaction are closely tied to the thermal environment [1]. In tropical countries such as Malaysia, the challenge of achieving acceptable thermal comfort is heightened due to persistently high ambient temperatures and humidity levels throughout the year [2]. This condition places significant demand on building cooling systems, especially in densely occupied university classrooms where internal heat gains can further exacerbate discomfort [3]. The widely accepted models for assessing thermal comfort, such as Fanger's Predicted Mean Vote (PMV) and Predicted

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Percentage of Dissatisfied (PPD), are often employed to evaluate perceived comfort based on environmental and personal variables, as guided by international standards including ASHRAE 55 and ISO 7730 [4].

Despite the proliferation of research on indoor environmental quality, the specific influence of indoor temperature variation on thermal comfort in university classrooms, under realistic tropical conditions, remains insufficiently explored. Many existing studies either generalise thermal comfort findings across climatic zones or focus narrowly on static environmental conditions. In Malaysia, where classrooms are frequently air-conditioned and densely populated, the interaction between temperature settings, internal heat gains, and actual comfort outcomes warrants more detailed examination. This is particularly relevant in scenarios involving high occupancy densities, such as 61 students seated for extended periods, where each occupant, or their equivalent thermal surrogate—contributes approximately 100 watts of metabolic heat to the room environment. The complexity of this setup, including non-uniform airflow, thermal stratification, and radiant heat interaction from internal fixtures, introduces spatial and temporal variability that challenges the reliability of single-point thermal assessments.

Recent literature has attempted to address various aspects of this issue, yet key limitations persist. For example, Lee and Zakaria [5] investigated PMV distribution in Malaysian classrooms and concluded that neutral comfort conditions are typically achievable at supply air temperatures between 22 °C and 24 °C, though their study did not account for zonal thermal asymmetry caused by internal loads. Similarly, Romero, Miranda [6] compared thermal comfort in air-conditioned and naturally ventilated university classrooms, but their analysis lacked fine-grained thermal mapping across a range of controlled temperatures. Kim and de Dear [7] explored the correlation between PMV and occupant satisfaction under different HVAC modes, while Alghamdi [8] extended this by establishing a linear PMV-temperature relationship for sedentary activities. However, both studies were conducted in less dense settings and without accounting for heat-generating occupants or surrogate models. In another study, Alfano, Pepe [9] evaluated mean radiant temperature fluctuations under mechanically ventilated conditions and observed significant thermal asymmetry at higher indoor temperatures, but their research did not quantify PMV across multiple zones. Elsewhere, Ragab, Hassieb [10] examined the impact of ventilation strategy on occupant comfort, noting that spatial comfort disparities are common but often underrepresented in CFD validation studies. More recently, Hu, Wang [11] examined stratification in zonal supply configurations but did not include internal heat gains nor PMV-based zoning analysis.

These findings underscore a critical research gap: the lack of studies that evaluate the variation of thermal comfort across a full range of indoor temperatures, particularly in highly occupied Malaysian classrooms. Most of the past work either considers comfort in aggregate terms or fails to integrate the effects of occupant-generated heat loads and spatial non-uniformity. Moreover, while the PMV model is often adopted as a standard thermal comfort index, its application in tropical university settings under varying thermal conditions and high-density occupancy has not been sufficiently validated. There is also a methodological gap in combining empirical zone-based measurements with Computational Fluid Dynamics (CFD) simulations to visualise and assess thermal gradients and airflow distribution in these complex environments.

To address these limitations, the present study evaluates thermal comfort in a university classroom located in Malaysia by analysing PMV variation across different indoor air temperature scenarios, ranging from 22 °C to 28 °C. The study adopts a hybrid approach, combining field measurements and CFD simulations at the breathing height of seated occupants (0.8 m), and models heat sources equivalent to 61 occupants each emitting 100 W of sensible heat. The objective is to provide spatially resolved insights into how indoor temperature affects PMV, airflow distribution, mean radiant temperature, and thermal uniformity. Unlike previous works, this study provides a zone-by-zone comfort assessment, allowing for the identification of areas prone to overheating or draft discomfort. The novelty lies in the integration of realistic internal load conditions and supply air temperature variation, offering practical guidance for HVAC design and operation in educational buildings within tropical climates. Ultimately, the findings contribute to a more nuanced understanding of comfort dynamics in high-density academic environments and inform more responsive, energy-efficient indoor climate control strategies.

2 Methodology

2.1 Computational domain

The computational domain for this study was developed using CATIA V5, based on actual classroom dimensions and layout representative of a typical Malaysian university environment. Lecture room at mechanical building of UTM was chosen for the case study site location throughout this study. The model captures essential architectural and interior elements, including walls, ceiling, floor, desks, chairs, a whiteboard, and HVAC components. As illustrated in Fig. 1. The isometric view (Fig. 1a) shows the spatial configuration of the entire room, while the top view (Fig. 1b) highlights the arrangement of furniture and the placement of HVAC outlets and exhaust fans. The side view (Fig. 1c) further details the vertical alignment of the HVAC outlets mounted on the ceiling and their relative position to the occupied zones.

To accurately simulate the airflow and thermal comfort distribution, the model also incorporates HVAC outlets for supply air and strategically positioned exhaust fans near the rear end of the room to represent the mechanical ventilation system. The geometry was simplified by excluding unnecessary details such as small objects or structural decorations that do not influence airflow, thereby reducing computational complexity while preserving the key flow and thermal characteristics. The domain height, width, and length were defined based on in-situ measurements, ensuring geometric fidelity for simulation purposes. This three-dimensional model serves as the foundation for meshing and subsequent CFD analysis aimed at evaluating temperature distribution, airflow patterns, and thermal comfort indices within the occupied classroom space.

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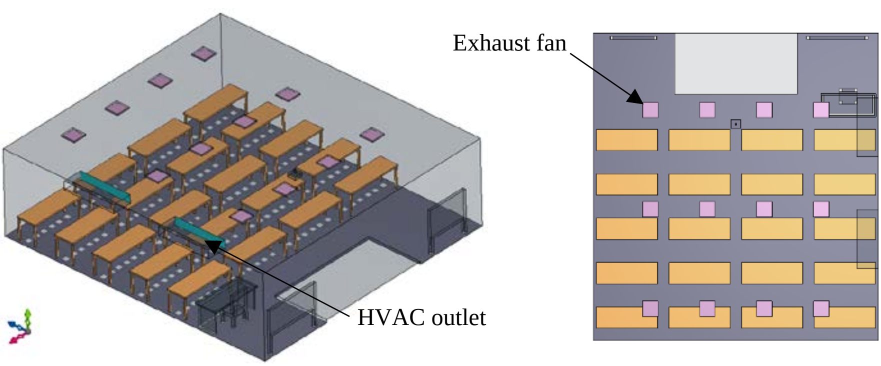
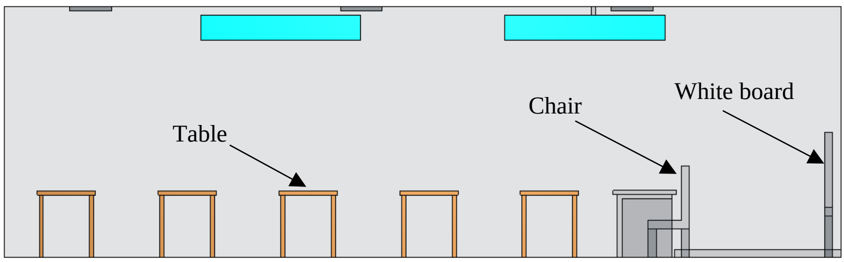


(a) Isometric view (b) Top view

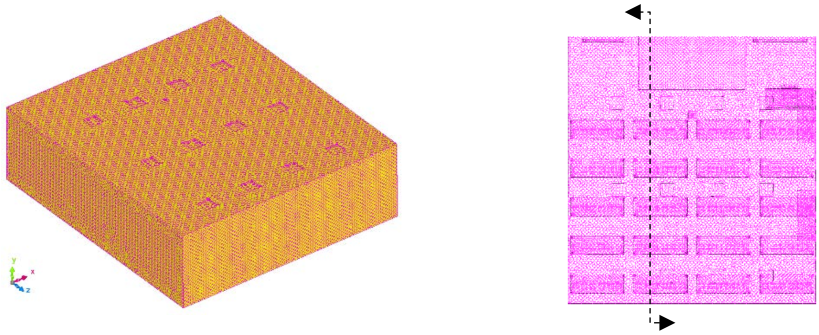


(c) Side view

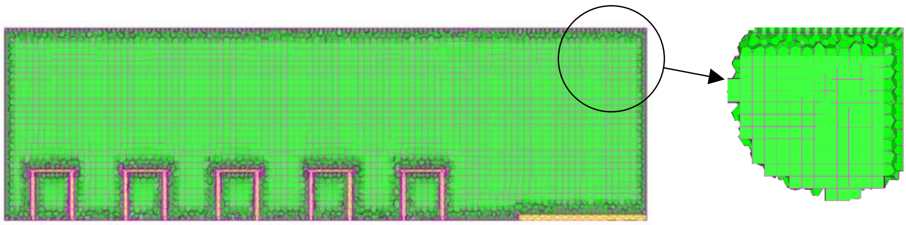
Fig. 1. Modeling of a computational domain

2.2 Meshing

The computational domain, developed in CATIA V5, was discretized using a polyhedral mesh to accurately capture airflow and temperature distribution within the classroom. The meshing process was conducted using a surface-to-volume approach, where surface elements were first generated based on the CAD geometry, followed by volume filling with polyhedral cells to ensure improved accuracy in complex regions and better convergence behaviour. Figure 2a shows the isometric view of the meshed domain, highlighting the uniformity and density of polyhedral elements across the entire volume. Compared to traditional tetrahedral meshes, polyhedral elements offer greater numerical stability and require fewer cells to achieve comparable accuracy, thus reducing computational cost. The top wireframe view (Fig. 2b) reveals the structured distribution of mesh elements over the occupied plane, with finer meshing applied around high-gradient regions such as HVAC outlets, exhaust fans, and near the occupant mannequins. A sectional cut taken along the classroom's central plane (Fig. 2c) illustrates the internal mesh layering, particularly around student desks and chairs, where localised airflow and thermal gradients are expected. To resolve near-wall flow behaviour and ensure accurate prediction of convective heat transfer, prism layers were applied along all solid boundaries, including the floor, ceiling, furniture surfaces, and occupants. These boundary layer meshes are clearly visible in Fig. 2d, which focuses on the mesh refinement at a corner region. A total of five inflation layers were generated with a growth ratio of 1.2 to maintain sufficient resolution of the wall-adjacent cells, essential for turbulence modelling and accurate estimation of PMV indices. The final mesh was validated through a mesh independence study to ensure the fidelity of the solution was not sensitive to further refinement.



(a) Isometric of polyhedral mesh (b) Wireframe top view



(c) Cut off view taken from 2(b) (d) Prism layer at corner

Fig. 2. Discretization of the domain

2.3 Boundary conditions

To evaluate the airflow and thermal comfort distribution in the classroom, appropriate boundary conditions were applied to replicate realistic operating and occupancy scenarios. The simulation was performed under steady-state conditions using the Reynolds-Averaged Navier–Stokes (RANS) approach with the standard k-ε turbulence model. A total of 500 iterations were specified to ensure convergence of residuals and stabilisation of solution variables.

The HVAC inlets were defined with a total volume flow rate of 83 cubic feet per minute (CFM), equivalent to approximately 0.039 m^3/s, and an inflow air temperature of 18 °C to represent typical cooling operation in Malaysian university classrooms. Outlet boundaries were modelled as pressure outlets with zero-gauge pressure to simulate ambient discharge conditions. Each thermal mannequin was modelled with a fixed surface heat flux of 100 W, representing sensible heat emission by a seated human body under sedentary activity. Additionally, exhaled air from each mannequin was simulated via a localised source with a normal velocity of 0.23 m/s and an inflow temperature of 36.7 °C, consistent with the thermal plume typically generated from breathing.

All solid surfaces, including desks, chairs, walls, ceiling, and floor, were treated with no-slip velocity and adiabatic thermal conditions, unless otherwise influenced by nearby heat sources. The air was assumed to be incompressible, and Boussinesq approximation was not employed due to the moderate temperature gradients involved. A summary of the boundary conditions is presented in Table 1.

Table 1. Boundary conditions

Region / Boundary	Type	Condition / Value
Simulation type	Flow + thermal (steady)	500 cycles, RANS, standard k-eps
Inlet (HVAC diffuser)	Velocity inlet	83 CFM, Temperature = 18 °C
Outlet (Exhaust fan)	Pressure outlet	Gauge pressure = 0 Pa
Mannequin surface	Heat source	100 W sensible heat per mannequin
Mouth opening (breathing)	Velocity inlet (local)	Velocity = 0.23 m/s, Temperature = 36.7 °C
Solid walls/furniture	Wall (no-slip)	Adiabatic
Gravity	Body force	Enabled in vertical direction

2.4 Governing equations

The numerical simulation in this study is based on the solution of the time-averaged governing equations of fluid flow and heat transfer, assuming incompressible, steady-state, and turbulent airflow within an enclosed classroom environment. The flow field is described using the Reynolds-Averaged Navier–Stokes (RANS) equations, which are derived from the instantaneous Navier–Stokes equations by applying Reynolds decomposition to account for the effects of turbulence. The resulting set of equations includes the continuity equation, momentum equations, energy equation, and turbulence transport equations.
The continuity equation for incompressible flow is expressed as:

$$\frac{\partial u_i}{\partial x_i} = 0 \tag{1}$$

The momentum equations, which govern the conservation of linear momentum in each coordinate direction, are given by:

$$\rho\left(u_j \frac{\partial u_i}{\partial x_j}\right) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) - \rho\overline{u_i' u_j'}\right] \tag{2}$$

where ρ is the fluid density, u_i and u_j are the velocity components, p is the static pressure, μ is the dynamic viscosity, and $\overline{u_i' u_j'}$ denotes the Reynolds stress tensor, which requires turbulence modelling for closure.

The energy equation, used to simulate heat transfer and thermal stratification within the domain, is written as:

$$\rho c_p u_j \frac{\partial T}{\partial x_j} = \frac{\partial}{\partial x_j}\left(k \frac{\partial T}{\partial x_j}\right) + \Phi \tag{3}$$

where c_p is the specific heat at constant pressure, T is the temperature, k is the thermal conductivity, and Φ represents the viscous dissipation term, which is typically negligible for low-speed indoor flows.

To model the turbulence effects, the standard k-ω model was employed in this study. This model solves two additional transport equations, one for the turbulence kinetic energy kk and another for the specific dissipation rate ω and is particularly suitable for wall-bounded internal flows with complex near-wall behaviour. The model offers improved accuracy in capturing low-Reynolds-number effects and is robust in predicting recirculation and separation zones often present in mechanically ventilated indoor environments.

The above set of equations was solved using a pressure-based solver with the SIMPLE algorithm for pressure–velocity coupling. Second-order upwind schemes were applied for the discretisation of momentum, energy, and turbulence equations to ensure numerical accuracy. The governing equations were implemented and solved using a commercial CFD package with convergence monitored through residuals and domain-specific integral quantities.

2.5 Mesh refinement and turbulence model selection

To ensure mesh independence and validate the accuracy of the numerical solution, a mesh refinement study was conducted using three mesh densities: coarse, medium, and fine. The representative cell lengths for these mesh levels were approximately 22 mm, 14 mm, and 10 mm, respectively. The evaluation metric used for this study was the axial velocity at a point located 10 cm downstream from the mannequin's mouth, corresponding to the breathing jet trajectory.

As illustrated in Fig. 3, the predicted velocity values at this location for the coarse, medium, and fine meshes were 0.147 m/s, 0.144 m/s, and 0.138 m/s, respectively. A clear trend of mesh convergence is observed as the cell size decreases. To estimate the grid-independent solution, Richardson Extrapolation was applied to the results obtained from the two finest mesh levels. The extrapolated velocity was calculated to be approximately 0.131 m/s, which serves as a reference value for evaluating the accuracy of the numerical solutions. The deviation between the fine mesh prediction and the extrapolated value was less than 6%, indicating an acceptable level of numerical uncertainty for indoor airflow simulation.

Multiple turbulence models were also assessed at the same location to examine their influence on the predicted velocity. As shown in the figure, the standard *k-ε*, *RNG k-ε*, *realizable k-ε*, and *SST k-ω* models all produced predictions within a narrow range, though variations were noted in comparison to experimental data. Among them, the *SST k-ω* model demonstrated a balanced performance between numerical stability and velocity prediction accuracy and was thus selected for subsequent simulations. The mesh refinement results confirm that the selected grid density and turbulence model provide reliable outcomes with minimal discretisation error.

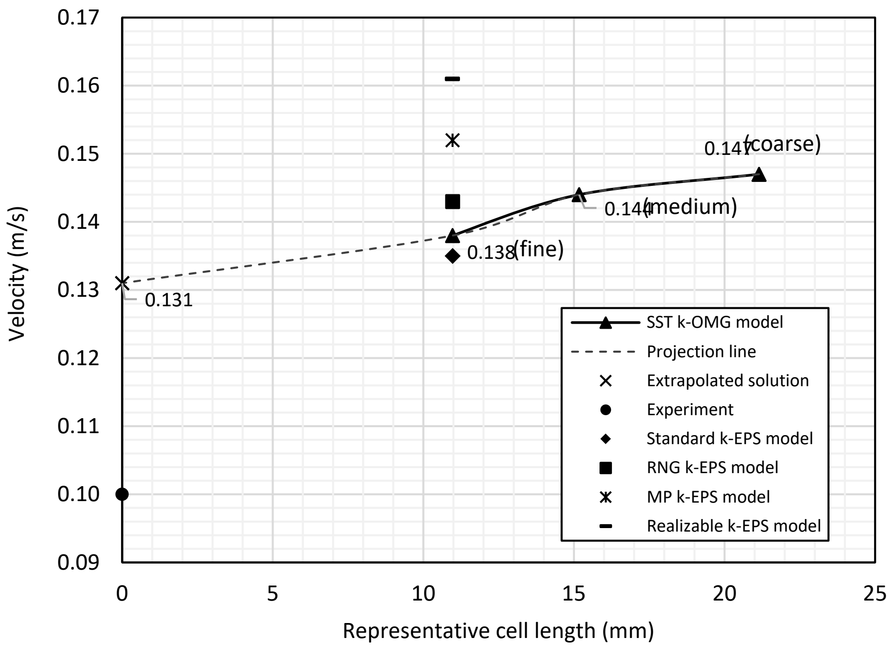


Fig. 3. Mesh refinement study of velocity against cell lengths

2.6 Validation

To ensure the reliability of the numerical model, validation was carried out by comparing the simulated air temperature values against experimental measurements at nine different points within the classroom. These validation points were located at a height of 1 m from the floor to represent the breathing zone of seated occupants. The temperature readings from the simulation were obtained under the same operating conditions as those applied during the field measurements, specifically with a supply air temperature of 18 °C.

As shown in Fig. 4, the predicted temperatures closely follow the experimental trend across all measurement locations. The maximum deviation between simulation and measurement was less than 0.6 °C, occurring at points with high spatial gradients near thermal plumes or supply diffusers. The average temperature difference across all points was found to be approximately 0.32 °C, indicating strong agreement between the numerical model and actual measured conditions.

The results demonstrate that the model is capable of accurately capturing indoor temperature distribution, including localised variations caused by airflow recirculation or internal heat sources. The slight discrepancies can be attributed to uncertainties in sensor placement, local turbulence effects, and simplifications made in the geometry or boundary conditions. Nonetheless, the level of agreement confirms the validity of the CFD setup and supports its use for further thermal comfort and airflow analysis within the same classroom environment.

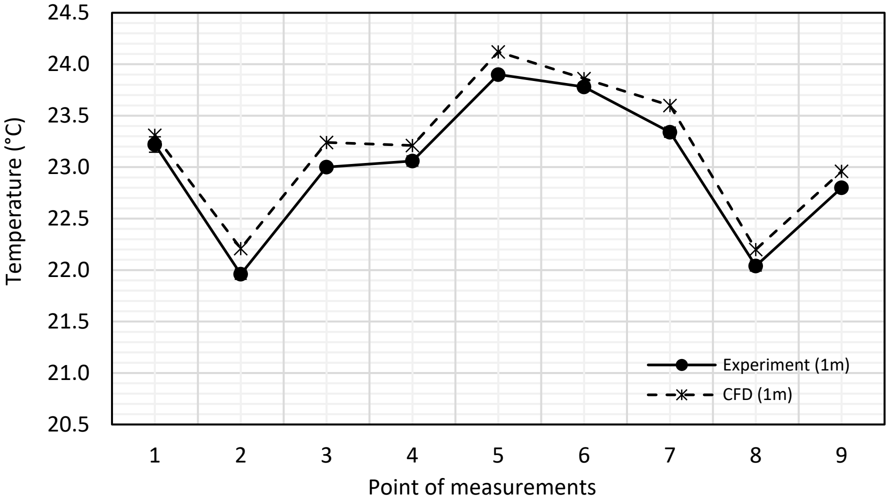


Fig. 4. Validation of temperature across different points

3 Result and Discussion

3.1 Thermal comfort in classroom

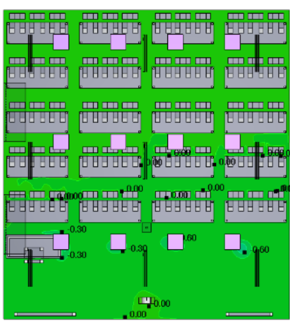
The PMV contour plots at 0.8 m height for supply air temperatures (T_n) from 22 °C to 28 °C are shown in Fig. 5, illustrating how thermal comfort evolves across the classroom plane with changing inlet temperature.

It was evident that cooler supply temperatures (22–24 °C) delivered a mostly neutral PMV distribution (between –0.5 and +0.5), indicating comfortable conditions throughout. As

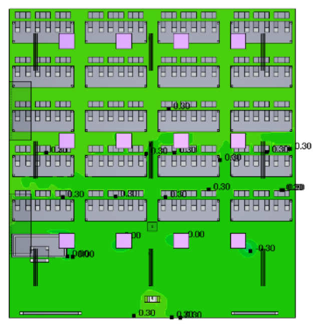
T_n increased beyond 24 °C, comfort deteriorated: PMV values shifted positive, suggesting occupants would feel increasingly warm. Particularly at 27–28 °C, large portions of the room displayed PMV near +1.0, indicating "slightly warm" and approaching discomfort.

Detailed trends support this: at $T_n = 22$ °C the PMV range remained tightly between –0.2 and +0.3 across student desks. At $T_n = 25$ °C, PMV rose to between +0.4 and +0.6 near the instructor zone. By $T_n = 28$ °C, peak PMV exceeded +1.0 in pockets around ceiling lamps and near exhaust vents—signifying a notable thermal gradient of ~1.5 PMV units across 0.8 m height plane.

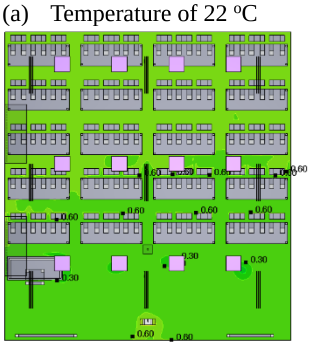
In summary, optimal thermal comfort at seated level (0.8 m) is maintained when supply air is kept between 22–24 °C, keeping PMV within the neutral range. Inlet temperatures at or above 26 °C lead to elevated PMV levels (+0.6 to +1.0), indicating thermal discomfort and asymmetric gradients across the space. The data demonstrate that maintaining T_n between 22–24 °C ensures uniform and comfortable thermal conditions at occupant level in the classroom. Supply temperatures above 26 °C compromise comfort, particularly near heat sources or poor circulation areas. These findings reinforce established HVAC design principles for educational environments.



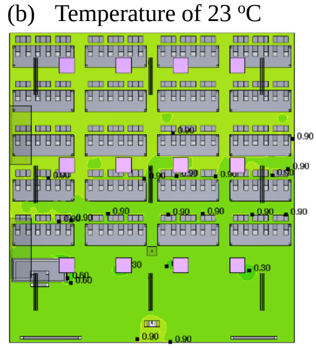
(a) Temperature of 22 °C



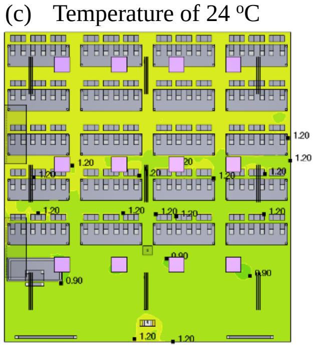
(b) Temperature of 23 °C



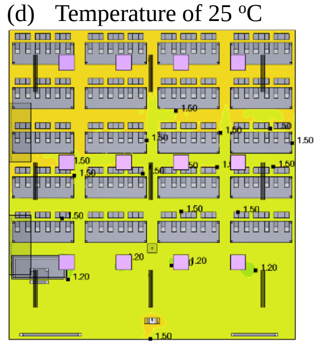
(c) Temperature of 24 °C



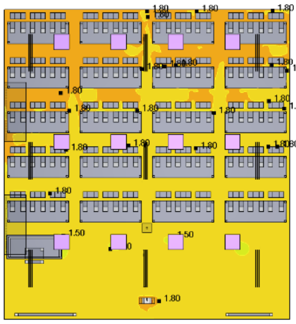
(d) Temperature of 25 °C



(e) Temperature of 26 °C



(f) Temperature of 27 °C



(g) Temperature of 28 °C

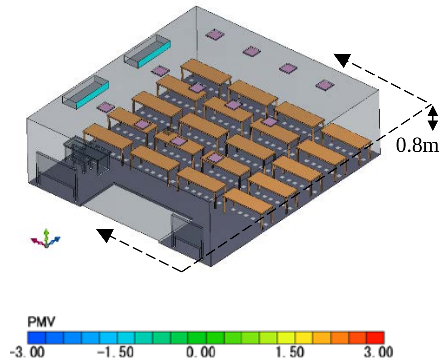


Fig. 5. contour plot on plane view of 0.8m from ground

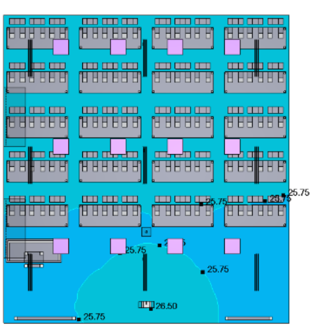
3.2 Spatial mean radian temperature (SMRT) in classroom

The SMRT (Spatial Mean Radiant Temperature) contour plots at the 0.8 m plane for supply air temperatures T_n = 22–28 °C are shown in Fig. 6, illustrating how radiant heat exposure evolves across the classroom at seated level.

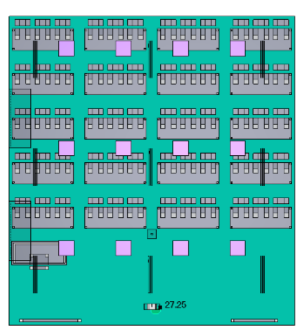
It was evident that lower supply temperatures (22–24 °C) maintained SMRT values within a comfortable range (approximately 25–29 °C) throughout the seating area. As T_n rose above 24 °C, SMRT gradually increased: at 25–26 °C, values peaked near 30–31 °C, particularly around heat sources such as ceiling lamps and near exhaust vents. At T_n = 28 °C, expansive zones exhibited SMRT values exceeding 31–32 °C, with localized maxima reaching close to 33–34 °C under obstructions and lamp clusters.

Detail trends reveal that at T_n = 22 °C, SMRT was uniform across the room, with tight variation (~25–27 °C). When T_n reached 26 °C, local hotspots emerged SMRT reached 31 °C near exhaust outlets and lamp shadows, indicating strong radiant gradients despite uniform air temperature. At T_n = 28 °C, the room saw large zones at 31–33 °C, highlighting diminished radiant comfort.

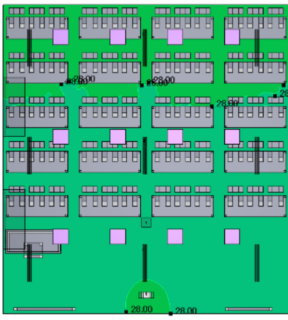
In summary, maintaining $T_n \leq 24$ °C ensures uniform SMRT (~25–29 °C) across the classroom, delivering radiant comfort at seated height. In contrast, supply temperatures of 26 °C and above progressively elevate radiant exposure, with 31–34 °C SMRT hotspots, which can contribute to discomfort and thermal asymmetry. To preserve radiant comfort at occupant level, supply air should be kept between 22–24 °C, thereby preventing local overheating and sustaining uniform SMRT—a critical factor often overlooked in classroom HVAC design.



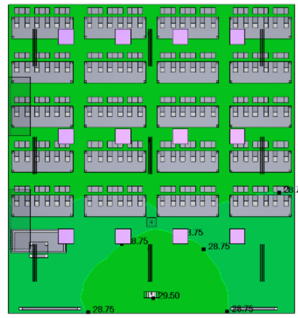
(a) Temperature of 22 °C



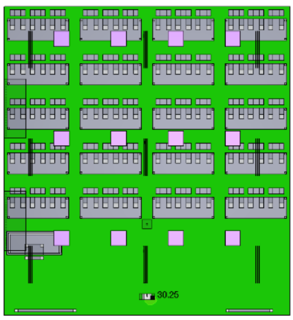
(b) Temperature of 23 °C



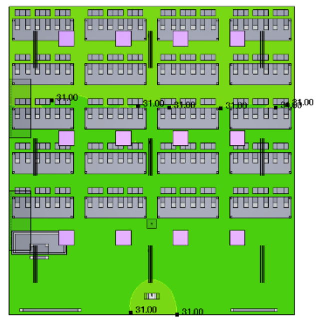
(c) Temperature of 24 °C



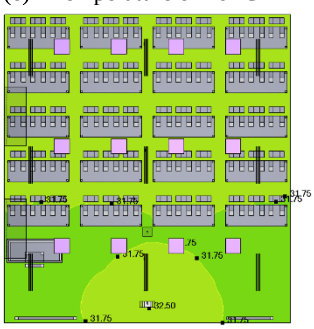
(d) Temperature of 25 °C



(e) Temperature of 26 °C



(f) Temperature of 27 °C



(g) Temperature of 28 °C

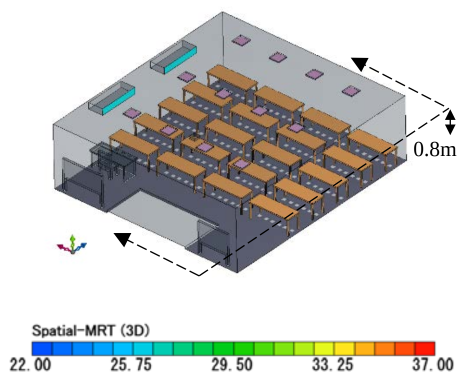


Fig. 6. contour plot on plane view of 0.8m from ground

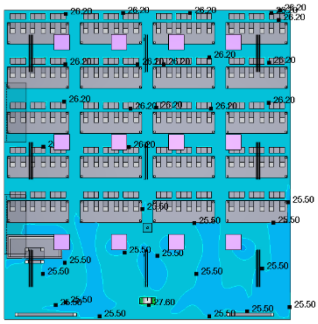
3.3 Temperature in classroom

The temperature contour plots at 0.8 m of height for supply air temperatures ranging from 22 °C to 28 °C are presented in Fig. 7, offering insight into the thermal field experienced by seated occupants across the classroom plane.

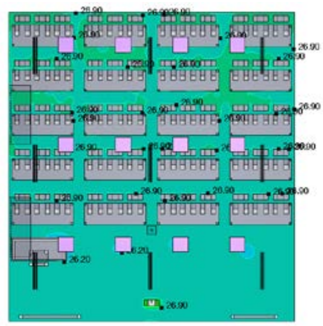
It was apparent that lower supply temperatures (22–24 °C) maintained the occupied plane in a thermally stable and comfortable air environment, with near-uniform temperatures spanning approximately 25–28 °C. As inlet temperatures increased, the thermal map exhibited progressive warming and reduced uniformity, signalling potential discomfort.

In detail, at $T_n = 22$ °C, most of the seating plane recorded temperatures between 25–26 °C. At 23–24 °C, slight warming was visible, with temperatures drifting to 26–27 °C, notably near the instructor's area and ceiling fixtures. Once T_n reached 26 °C, temperature inhomogeneity became more pronounced: readings fluctuated between 27–29 °C, especially around exhaust vents and shadow zones. For 27–28 °C inlet conditions, large sections hit 30–32 °C, with local hotspots reaching 33–36 °C near heat sources, indicating a temperature gradient of 8–10 °C across the plane.

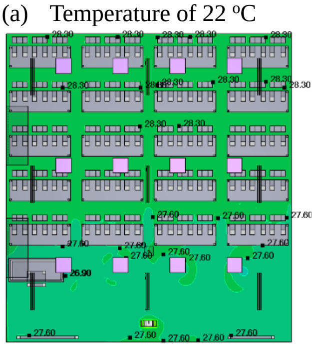
In summary, air temperatures at seated height remain comfortable and well-regulated for supply air between 22–24 °C, while T_n at or above 26 °C triggers substantial warming and uneven thermal fields, with potential discomfort due to elevated local temperatures. To ensure occupant comfort and thermal uniformity at seated level, HVAC systems should supply air within the 22–24 °C range. Supply temperatures beyond 25–26 °C induce unacceptable temperature gradients (8–10 °C), undermining comfort and efficiency in classroom environments.



(a) Temperature of 22 °C

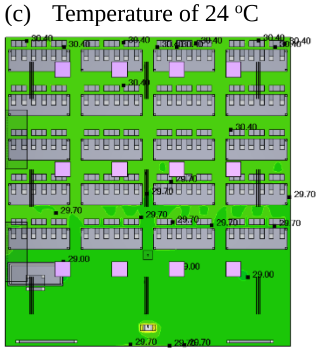


(b) Temperature of 23 °C

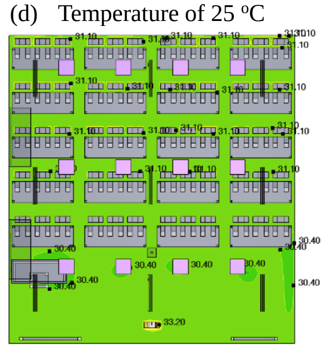


(c) Temperature of 24 °C

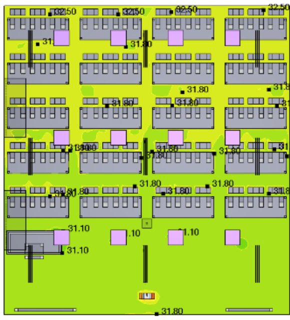
(d) Temperature of 25 °C



(e) Temperature of 26 °C



(f) Temperature of 27 °C



(g) Temperature of 28 °C

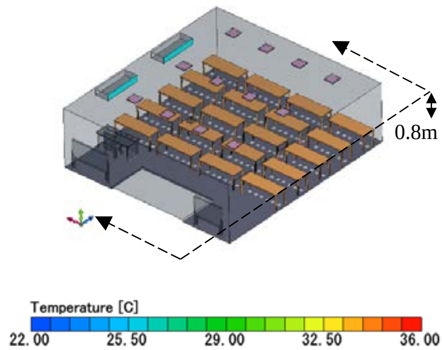


Fig. 7. Temperature contour plot on plane view of 0.8m from ground

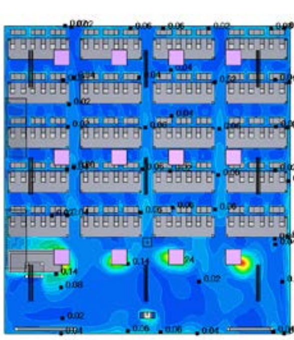
3.4 Velocity flow in classroom

The velocity contour plots at 0.8 m height for supply air temperatures $T_n = 22–28$ °C are shown in Fig. 8, revealing how airflow magnitude evolves across the classroom with increasing inlet temperature.

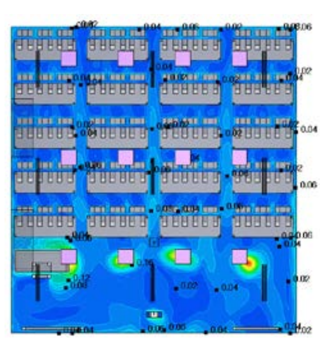
It was observed that lower supply temperatures (22–24 °C) produced smooth, low-speed airflow patterns (< 0.10 m/s) throughout the seated zones. As T_n increased, airflow jets intensified especially around active heat sources like the instructor desk and lamp areas—leading to localized velocities reaching 0.25–0.30 m/s at $T_n = 26$ °C and rising to peaks of 0.35–0.40 m/s at $T_n = 28$ °C.

Specifically, at $T_n = 22$ °C, the airflow remained uniformly low with negligible velocity gradients. By $T_n = 26$ °C, stronger jets under the instructor zone and at the rear occupied desks produced velocity spikes (~0.30 m/s), exceeding typical summer comfort limits. At $T_n = 28$ °C, these jets intensified further, and zones near exhaust vents saw velocity spikes up to 0.40 m/s, while the central aisle retained lower speeds (~0.10 m/s), indicating non-uniform circulation and potential draft risk.

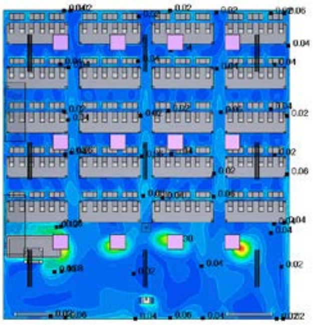
In summary, maintaining T_n between 22–24 °C yields gentle and uniform airflow (< 0.10–0.15 m/s) throughout the seating area. In contrast, raising T_n to 26 °C or above resulted in localized airflow intensities (0.30–0.40 m/s) that may cause drafts and discomfort, especially near occupants and fixtures. To ensure occupant comfort and compliance with draft standards, classroom supply air temperature should be limited to ≤ 24 °C. Higher temperatures produce strong, uneven airflow that increases draft risk, a critical consideration in HVAC design for educational spaces.



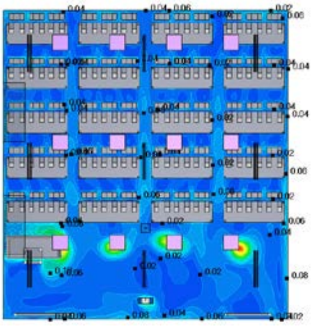
(a) Temperature of 22 °C



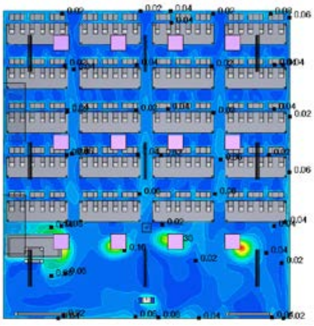
(b) Temperature of 23 °C



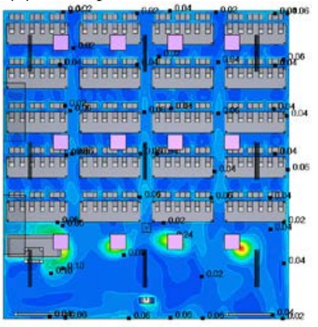
(c) Temperature of 24 ^{o}C



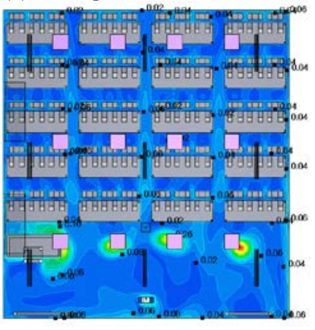
(d) Temperature of 25 ^{o}C



(e) Temperature of 26 ^{o}C



(f) Temperature of 27 ^{o}C



(g) Temperature of 28 ^{o}C

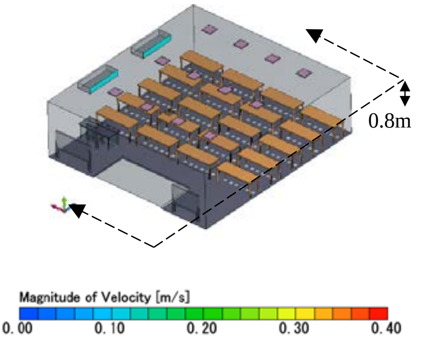


Fig. 8. Velocity contour plot on plane view of 0.8m from ground

3.5 Mean operative temperature on thermal comfort

The relationship between mean air temperature and Predicted Mean Vote (PMV) is shown in Fig. 9. This result presents how thermal comfort, as perceived by occupants, responds to variations in supply air temperature in the classroom environment.

It was found that the PMV value increased linearly with rising air temperature, indicating a strong correlation between indoor temperature and perceived thermal sensation. The PMV remained near neutral (PMV ≈ 0) at 22 °C but rose progressively to values exceeding +1.6 at 28 °C, indicating increasing thermal discomfort.

A detailed observation showed that PMV crossed the +0.5, comfort threshold (ASHRAE Standard 55) between 24 and 25 °C, suggesting that temperatures above this range likely result in occupants feeling "slightly warm" to "warm". The regression analysis further confirms this relationship, with the linear trend given by PMV = -0.1777x - 0.7587, and a coefficient of determination R^2 = 0.9096, signifying a strong statistical fit between temperature and PMV.

In conclusion, the data show that indoor air temperature has a direct and significant effect on thermal comfort, with optimal PMV performance (0 to +0.5) achievable when the classroom temperature is maintained between 22 °C and 24 °C. Temperatures above this range lead to higher PMV values, increasing the risk of discomfort and reducing thermal satisfaction among occupants.

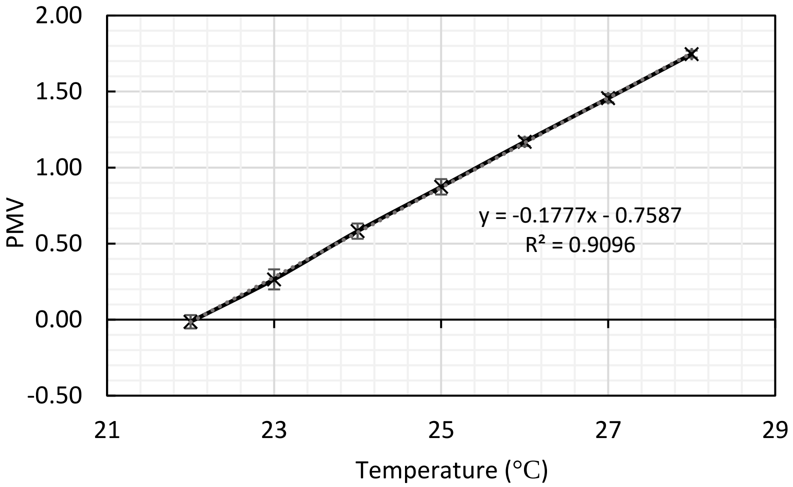


Fig. 9. Mean air temperature on PMV

3.6 Operative temperature on thermal comfort zone by zonez

The variation of mean PMV across different classroom zones with respect to supply air temperature is illustrated in Fig. 10. This result highlights the spatial thermal comfort disparity within the classroom from T_n = 22 °C to 28 °C, segmented into six zones as previously defined.

It was found that PMV values increased steadily in all zones as temperature rose, but the degree of comfort varied slightly between zones. Zones near the rear and corners (Zones 5 and 6) consistently recorded higher PMV values, indicating a warmer sensation, while front and central zones (Zones 1 and 2) maintained relatively lower PMV values, suggesting better thermal comfort under the same supply conditions.

In detail, at T_n = 22 °C, all zones reported PMV close to 0.0 (neutral), indicating ideal conditions. At 24 °C, PMV ranged from ~0.4 in Zone 2 to ~0.6 in Zone 5, approaching the upper comfort limit. At 28 °C, PMV peaked between 1.6–1.8, with Zone 5 and Zone 6 exhibiting the highest thermal discomfort, while Zone 2 remained the coolest. These trends suggest that zones further from inlets or closer to heat sources (e.g., lighting or occupants' heat) were more vulnerable to temperature-induced discomfort.

In summary, the results show that Zone 2 (front-left) consistently provided the most stable and comfortable thermal condition, while Zone 5 and Zone 6 (rear areas) were most affected by rising temperatures, leading to higher PMV and potential discomfort. To ensure balanced thermal comfort across all zones, HVAC designs must account for zonal discrepancies. Optimising air distribution to mitigate rear zone overheating and maintain PMV values below

+0.5 across the classroom is essential—especially when supply air temperatures exceed 24 °C.

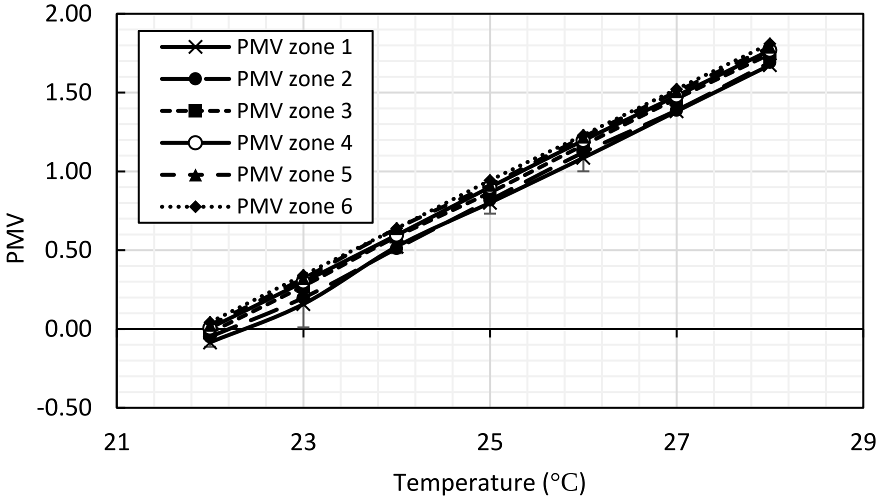


Fig. 10. Mean air temperature on PMV for zone by zone

3.7 Standard deviation of PMV for zone by zonez

The standard deviation of PMV for each zone at different supply air temperatures is illustrated in Fig. 11. This analysis reflects the thermal uniformity within each zone, where lower standard deviation (SD) indicates more consistent comfort levels.

It was found that Zone 1 exhibited the highest variability, especially at 23 °C, where its PMV SD peaked at approximately 0.15, significantly higher than other zones. In contrast, Zones 2, 4, and 5 consistently showed lower PMV variability, with SD values mostly below 0.02, indicating more stable thermal conditions across those zones.

Detail trends revealed that at 23 °C, both Zone 1 and Zone 3 experienced notable spikes in variability—SD reached 0.15 in Zone 1 and ~0.075 in Zone 3. This could be attributed to airflow interactions and localized heat sources, such as instructor movement or ceiling lamp proximity. From 24 °C to 28 °C, Zone 1 continued to show moderate fluctuations (0.06–0.08), while Zone 3 also maintained SD around 0.06–0.07, suggesting non-uniform temperature distribution or airflow within those zones. Conversely, Zones 2 and 4 maintained uniform PMV throughout the range, indicating efficient air mixing and minimal gradient.

In summary, the analysis reveals that Zone 1 (front-right) and Zone 3 (middle-right) are more prone to thermal non-uniformity, particularly at intermediate temperatures (23–25 °C). Meanwhile, Zone 2 and Zone 4 consistently exhibited stable and uniform comfort. For optimal comfort consistency, HVAC designs should address the airflow dynamics in Zone 1 and Zone 3, where PMV variability is highest. Reducing localised heat gains or adjusting diffuser placement may help mitigate uneven thermal perception across those zones.

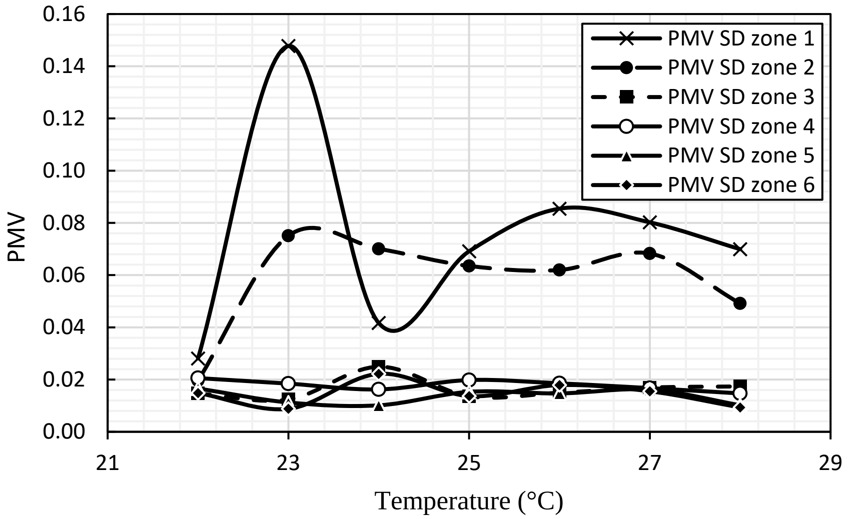


Fig. 11. Standard deviation of PMV for zone by zone

4 Conclusion

This study evaluated the impact of indoor temperature variation on thermal comfort in a mechanically ventilated university classroom in Malaysia using a combination of CFD simulations and field measurements. The analysis focused on seven temperature scenarios ranging from 22 °C to 28 °C, with thermal comfort assessed via PMV, SMRT, and operative temperature indices.

The results indicate that optimal thermal comfort at seated occupant level (0.8 m) is maintained when the supply air temperature is kept between 22 °C and 24 °C. Within this range, PMV values remained in the neutral zone (–0.2 to +0.3), with minimal spatial variation. However, when supply air temperature increased to 26 °C and above, PMV values exceeded +1.0 in several zones, indicating thermal discomfort. At 28 °C, the maximum PMV recorded was approximately +1.8, accompanied by a thermal gradient of up to 1.5 PMV units across the classroom. The SMRT results further confirmed that higher inlet temperatures elevated surface temperatures by up to 34 °C, particularly around ceiling fixtures and stagnant zones.

Zone-based analysis revealed significant spatial disparity in comfort levels. Rear zones (Zone 5 and Zone 6) consistently exhibited higher PMV values and greater thermal gradients than front zones. The standard deviation of PMV reached as high as 0.15 in certain zones, indicating thermal non-uniformity due to uneven air distribution and heat accumulation. Additionally, a strong linear relationship was established between supply air temperature and PMV ($R^2 = 0.9096$), reinforcing the temperature-dependence of perceived comfort in tropical classrooms.

Mesh refinement and model validation supported the accuracy of the simulation results. Richardson extrapolation yielded an axial velocity of 0.131 m/s at 10 cm from the mannequin's mouth, with the selected *SST k-ω* turbulence model achieving less than 6% deviation from the extrapolated value. Experimental validation against nine measurement points resulted in an average temperature deviation of just 0.32 °C from CFD predictions.

In conclusion, this study demonstrates that thermal comfort in tropical classrooms is highly sensitive to supply air temperature and airflow uniformity. It is recommended that HVAC systems maintain inlet air temperatures at or below 24 °C and apply zone-based air distribution strategies to avoid discomfort in thermally disadvantaged regions of the classroom. These insights provide actionable guidance for enhancing indoor environmental quality and energy efficiency in educational buildings across hot-humid climates.

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