OPTIMIZATION OF SUPPLY OUTLET HEIGHT FOR A FAÇADE-MOUNTED DOWNDRAFT EVAPORATIVE COOLER USING CFD-BASED MULTI-OBJECTIVE ANALYSIS OF THERMAL COMFORT AND INDOOR AIR QUALITY

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ABSTRACT

Evaporative cooling systems are increasingly used as energy-efficient alternative to convention air conditioning in hot climates. However, the thermal comfort and indoor air quality performance of façade-mounted downdraft evaporative coolers strongly depend on the height of the supply outlet. This study investigates the optimal outlet height for a downdraft evaporative cooler installed on the façade of an office space using computational fluid dynamics (CFD). The outlet height was varied from 0.1m to 2.4m above the floor level at increments of 0.1m. For each configuration, the Predicted Mean Vote (PMV) was evaluated at the occupant level (1.1m above the floor) while the Air Change Effectiveness (ACE) was determined in the breathing zone to assess indoor air quality. Multi-objective optimization was conducted with the objectives of achieving maximum ACE while maintaining thermal comfort within 0.5 ≤ PMV ≤ +0.5. Results show that outlet heights between 0.3m and 0.5m satisfy the thermal comfort criteria, while the maximum ACE value (1.1588) occurs at an outlet height of 0.3m. The optimal configuration therefore corresponds to a supply outlet height of approximately 0.3m above the floor level. The findings provide practical design guidance for façade-mounted evaporative cooling systems in office environments located in hot and dry climates.

Keywords: Evaporative cooling, CFD, Thermal comfort, PMV, Air change effectiveness, Optimization

1 INTRODUCTION

Building accounts for a significant proportion of global energy consumption, particularly due to heating, ventilation, and air-conditioning (HVAC) systems. In hot and semi-arid climates, air-conditioning energy demand continues to increase rapidly, raising concerns about energy sustainability and carbon emissions [1, 2, 3]. As a result, evaporative cooling technologies are receiving renewed attention as low-energy alternatives for indoor thermal control. Direct evaporative cooling systems operate by reducing dry-bulb air temperature through the evaporation of water into an airstream. Compared with conventional vapour-compression air-conditioning systems, evaporative coolers consume significantly less electrical energy because they require only fan power and water circulation [4]. Consequently, evaporative cooling has been widely investigated for applications in residential and office buildings in hot and semi-arid climates. The performance of evaporative cooling systems is greatly a function of airflow distribution within the conditioned space. Poor air distribution can lead to thermal discomfort, stagnant zones, and inadequate removal of indoor contaminants [5, 6]. In particular, the location and height of supply air outlets greatly influence airflow penetration and mixing characteristics. Several researchers have used computational fluid dynamics (CFD) to analyze airflow behaviour in ventilated indoor environments. CFD has become a globally accepted tool for predicting indoor air distribution, thermal comfort, and contaminant transport because of its capability in allowing detailed analysis of complex airflow patterns [7, 8]. Studies have shown that supply air diffuser location and jet characteristics strongly influence both thermal comfort and indoor air quality of occupied spaces [9, 10, 11]. Thermal comfort in indoor environments is commonly evaluated using the Predicted Mean Vote (PMV) model developed by Fanger, which predicts the mean thermal sensation of occupants based on environmental and physical parameters. According to ISO 7730 and ASHRAE Standard 55, acceptable thermal comfort conditions occur when PMV lies within the range $-0.5 \leq \text{PMV} \leq +0.5$ (ISO, 2005; ASHRAE, 2020) on the thermal sensation scale. Indoor air quality and ventilation effectiveness can be assessed using the Air Change Effectiveness (ACE) index derivable from mean age of air (MAA), which measures the ability of a ventilation system to deliver fresh air to the breathing zone relative to a perfectly mixed system [12, 13]. Higher ACE values indicate more effective contaminant removal. Previous investigations have focused mainly on airflow distribution from ceiling diffusers or displacement ventilation systems. However, limited research has examined the performance of façade-mounted downdraft evaporative coolers with respect to the optimal height of the supply outlet. The installation height of the supply outlet may significantly influence the airflow regime inside a room. Low-level outlets can promote displacement-type ventilation, whereas high-level outlets tend to produce mixed ventilation patterns. These airflow mechanisms may affect both occupant's thermal comfort and indoor air quality. Despite the growing interest in evaporative cooling systems, optimization of supply outlet height considering both thermal comfort and ventilation effectiveness has not been adequately explored. Therefore, the objective of this study is to determine the optimal supply height for a façade-mounted downdraft evaporative cooler in an office environment using CFD simulation and multi-objective optimization based on PMV and ACE.

1.1 The Study Area and System Characteristics

The characteristics of the study area, the building, and the downdraft evaporative cooler pertinent for Design Builder CFD analysis are presented in Table 1.

Table 1. Characteristics of the Case Study

Title	Characteristics
Building type	Office building
Location	Bayero University Kano
Climate	Latitude: 12.05ºN
	Longitude: 8.53ºE
	Elevation: 481m above sea level
Floor height	3.0m
Occupancy (person/m^2)	0.068
Office hours	8.00 am to 4.00 pm
Office dimension	4.0m long, 3.7m wide, and 3.0m high
Office equipment	Laptop and printer
Outlet cross-section of cooler	0.6m x 0.6m

2 METHODOLOGY

2.1 Creation of 3D model of the office building

Design Builder software was employed in the creation of the office model dimensioned 4m x 3.7m x 3m. Figure 1 shows the office building model at zone level while Figure 2 shows the integrated model of the office building with the downdraft evaporative cooler.

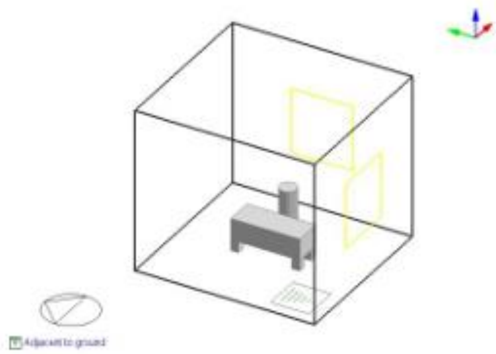

Figure 1. 3D model of the office at zone level

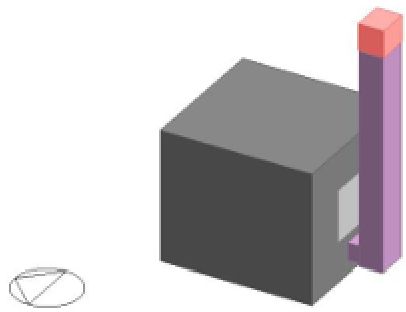

Figure 2. Integrated cooler-office building model

2.2 Mesh generation and independency study

Meshes were generated using Design Builder parameters for coarse, medium, and fine grids as shown in Figure 3.

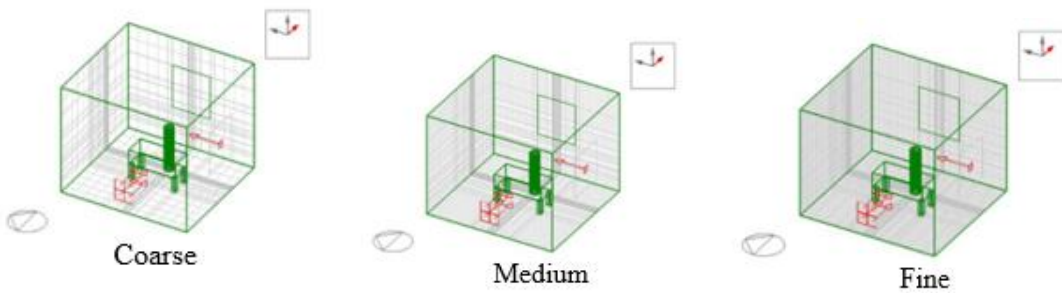

Figure 3. Coarse, medium and fine grids of the geometrical model

To ensure numerical accuracy, simulations were conducted using three grid densities as presented in Table 2. The predicted average air velocity and temperature at the occupant level (1.1m above the floor level) were compared for the three mesh levels. The relative deviation between the medium and fine meshes was less than 2%, indicating that the medium mesh provided sufficient accuracy while maintaining reasonable computational cost. Therefore, the medium mesh configuration (780,000 cells) was selected for all subsequent simulations.

Table 2. Model Mesh Densities

Mesh Type	Number of Cell
Coarse	420,000
Medium	780,000
Fine	1,250,000

2.3 Governing Equations of the Computational Model

The airflow distribution and heat transfer within the office building were simulated using CFD module of Design Builder software, which solves the Reynolds-Average Navier-Stokes (RANS) equations using finite volume method. The indoor airflow was assumed to be three-dimensional, turbulent, incompressible, and steady-state, which is consistent with airflow conditions in mechanically ventilated buildings [14].

2.3.1 Continuity Equation

The conservation of mass for incompressible flow is expressed as:

$$\nabla . u = 0 \tag{1}$$

Where u is the velocity vector in m/s.

2.3.2 Momentum Equation

The conservation of momentum in Cartesian Coordinates is given by:

$$\rho(u . \nabla)u = -\nabla p + \mu\nabla^2 u + F \tag{2}$$

Where: ρ = Air density (kg/m^3), p = Pressure (Pa), μ = Dynamic viscosity (kg/m.s), F = Body forces (Such as buoyancy).

2.3.3 Energy Equation

The temperature distribution within the office space is governed by the energy conservation equation.

$$\rho C_p(u . \nabla T) = k\nabla^2 T + S_T \tag{3}$$

Where T = Air temperature (K), C_p = Specific heat capacity of air (J/kg K), K = Thermal conductivity of air (W/mK), S_T = Heat source term.

2.3.4 Turbulence Model

Indoor airflow is typically turbulent due to the interaction between supply air jets and room surfaces. To account for turbulence effects, the standard $k - \varepsilon$ turbulence model was employed. The model introduces two additional transport equations: one for turbulent kinetic energy (k) and another for the turbulent dissipation rate (ε).

2.3.5 Turbulent Kinetic Energy Equation

$$\frac{\partial(\rho k)}{\partial t} + \nabla.(\rho k u) = \nabla.\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\nabla k\right] \tag{4}$$

Where k = Turbulent kinetic energy (m^2/s^2), μ_t = Turbulent viscosity, G_k = Production of turbulence due to velocity gradients.

2.3.6 Dissipation Rate Equation

$$\frac{\partial(\rho\varepsilon)}{\partial t} + \nabla.(\rho\varepsilon u) = \nabla.\left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\nabla\varepsilon\right] \tag{5}$$

Where ε = Turbulent dissipation rate (m^2/s^3), $C_{1\varepsilon}, C_{2\varepsilon}$ = Empirical model constants

2.3.7 Turbulent Viscosity

The turbulent viscosity is computed as:

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon} \tag{6}$$

Where C_μ is model constant.

2.3.8 Model Constants

For the standard $k - \varepsilon$ model, the following empirical constants presented in Table 3 were used.

Table 3. Empirical Constants for Standard $k - \varepsilon$ Model

Constant	Value
C_μ	0.09
$C_{1\varepsilon}$	1.44
$C_{2\varepsilon}$	1.92
σ_k	1.0
σ_ε	1.3

These values are widely adopted in indoor airflow simulations and have been validated for ventilation studies in buildings.

2.3.9 Thermal comfort model

Thermal comfort was evaluated using Fanger's PMV model [15]:

$$PMV = (0.303e^{-0.036\,M} + 0.028)[(M - W) - E_d - E_r - C - R] \quad (7)$$

Where $M =$ Metabolic rate, $W =$ Mechanical work, $E_d =$ Heat loss by diffusion, $E_r =$ Evaporative heat loss, $C =$ Convective heat loss, $R =$ Radiative heat loss. PMV values range from -3 (cold) to +3 (hot) on the scale of thermal sensation. According to International Organization for Standardization standard ISO 7730 and ASHRAE Standard 55, occupants are thermally comfortable when: $-0.5 \leq PMV \leq +0.5$.

2.4 Air change effectiveness (ACE)

Ventilation performance was evaluated using Air Change Effectiveness [6, 15]:

$$ACE = \frac{T_n}{T_b} \quad (8)$$

Where $T_n =$ Nominal air age, $T_b =$ Local air age in the breathing zone. The interpretation of the ACE is presented in Table 4 [6]. It is worthy to note that higher ACE indicates better contaminant removal.

Table 4. ACE Characteristics

ACE Value	Ventilation Quality
ACE < 1	Poor ventilation
ACE = 1	Perfect mixing
ACE > 1	Displacement-type ventilation

2.5 Boundary Conditions

The principal boundary conditions used in the CFD simulations conducted with Design Builder are summarized in Table 5. These conditions represent the airflow characteristics of the façade-mounted downdraft evaporative cooler and the physical conditions within the office building. The results obtained concur with [5].

Table 5. Principal Boundary Conditions for the CFD Simulations

Boundary Location	Type of Boundary Condition	Specification
Evaporative cooler outlet	Velocity inlet	Supply air from downdraft evaporative cooler; outlet height varied from **0.1 – 2.4m in 0.1m** increments
Supply air temperature	Fixed temperature	Temperature of air delivered by evaporative cooler
Return/exhaust opening	Pressure outlet	Atmospheric pressure (0 Pa gauge)
Walls	No-slip wall	Stationary wall with defined thermal properties
Floor	No-slip wall	Adiabatic or constant temperature surface
Ceiling	No-slip wall	Constant temperature boundary
Occupant heat source	Internal heat gain	Sensible heat from occupant activity
Turbulence model	Turbulence enclosure	Standard $k - \varepsilon$ model
Thermal comfort evaluation point	Monitoring point	**1.1m** above floor level
Ventilation performance zone	Occupied zone	**1.0 – 1.2m** above floor

2.6 Pareto Optimization Analysis

The optimization of the supply outlet height was performed using a multi-objective Pareto optimization approach considering two performance indicators: Thermal comfort (PMV). Indoor air quality (ACE). The optimization objectives were defined as: Maximize ventilation effectiveness: Maximize ACE: Subject to the thermal comfort constraints: $-0.5 \leq PMV \leq 0.5$. Where the acceptable PMV range corresponds to the thermal comfort limits recommended by ASHRAE Standard 55 and ISO 7730.

3 RESULTS AND DISCUSSION

3.1 Influence of Supply Outlet Height on Thermal Comfort

The downdraft evaporative cooler supply outlet height determines how the cooled and humidified air jet reaches the occupants as shown on the 2D PMV contour plots at representative heights of 0.6 m, 1.2 m, 1.8 m, and 2.4 m above the floor level in Figure 4. This clearly shows how the variation of the cooler's outlet height affect the distribution of the cooled and humidified air in the office space which, ultimately, affect the thermal sensation of the occupants. The results obtained concur with [13].

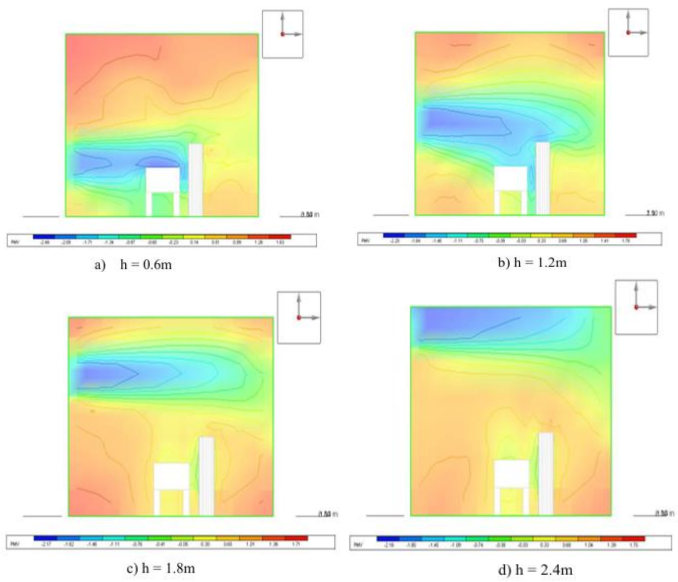

Figure 4. 2D Contour plot of PMV at representative cooler's outlet heights

The numerical results of the PMV of the thermal comfort of the occupant at the specified height of 1.1m above the floor level due to variation in the cooler's supply outlet height is shown in Figure 5. The results obtained is similar with a study of [14].

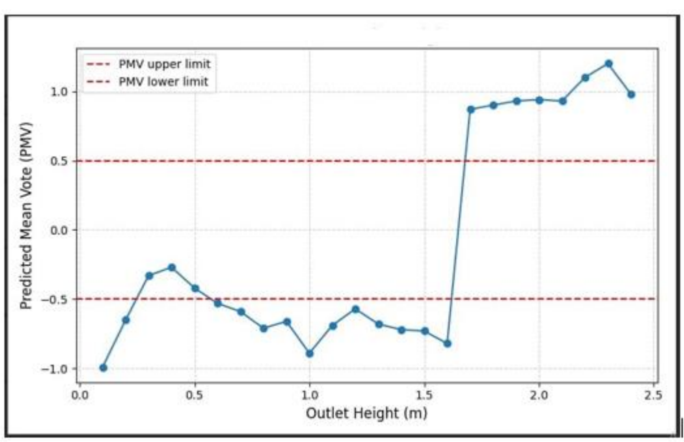

Figure 5. PMV versus cooler's supply outlet height

The PMV values predicted in the study varied substantially with changes in the supply outlet height, indicating that the location air supply strongly affects the thermal comfort experienced by occupants. Lower outlet heights (0.3m – 0.5m) produced PMV values within the acceptable comfort range recommended by ASHRAE Standard 55 and ISO 7730. This suggest that the airflow produced by the evaporative cooler effectively reached the occupant level while maintaining moderate air velocity and temperature conditions. Previous studies have shown that supply air jets introduced at low heights often promote displacement-like airflow patterns, where cooler air spreads across the floor and gradually rises due to buoyancy effects possibly created by thermal plumes around the occupants [3]. Such airflow behaviour enhances thermal stratification and can improve perceived comfort in occupied zones. Similar observations were reported by Rahmati *et al*. (2018), who demonstrated that low-level supply air systems often provide improved comfort conditions compared with high-level mixing ventilation systems. The PMV values predicted for higher outlet heights (> 1.5m) exceeded the acceptable comfort limits, indicating that the airflow was less effective in delivering cooled air to the occupant zone. This finding aligns with the work of [16] who reported that when supply air is introduced at higher elevations, the jet tends to mix with room air before reaching the occupant, thereby reducing its cooling effectiveness. This also aligns with the work of Haruna *et al*., (2022a) who reported that low-level supply air systems can provide improved thermal comfort due to enhanced stratification and reduced turbulence intensity near occupants.

3.2 Influence of Supply Outlet Height on Ventilation Effectiveness

ACE values obtained in this study ranged between 0.75 and 1.16 as shown in Figure 7. This indicates varying levels of ventilation effectiveness depends on the cooler's outlet configuration. An ACE value greater than unity indicates that fresh air reaches the breathing zone more efficiently than in a perfectly mixed ventilation system [6]. The highest ACE value (1.1588) occurred at an outlet height of 0.3m, suggesting that this configuration enhanced contaminant removal in the occupied zone. The improved ventilation performance at low outlet heights can be attributed to the displacement airflow mechanism created by the downdraft evaporative cooler. In this airflow regime, cool air introduced near the floor displaces warmer air upward, promoting efficient removal of heat and contaminants through the upper region of the room. The displacements of warmer air due to the introduction of cooler air at the representative outlet heights of 0.6m, 1.2m, 1.8m, and 2.4m above the floor level are shown on the 3D contour plots in Figure 6.

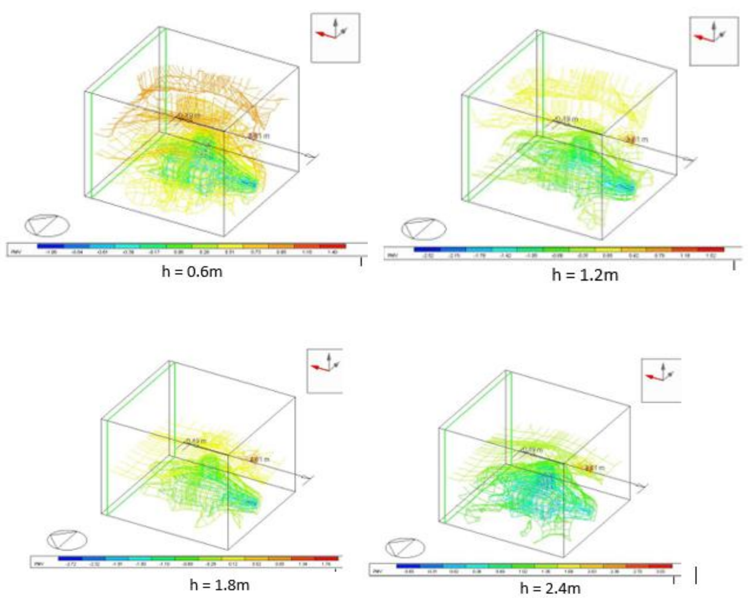

Figure 6. 3D contour plots showing displaced warmer air at selected outlet heights

Such airflow patterns have been shown to improve indoor air quality in comparison with traditional mixing ventilation systems [17]. This phenomenon aligns with the study conducted by Rahmati *et al*., (2018) who found that supply air introduced near the floor significantly improved contaminant removal in office environments due to enhanced airflow stratification.

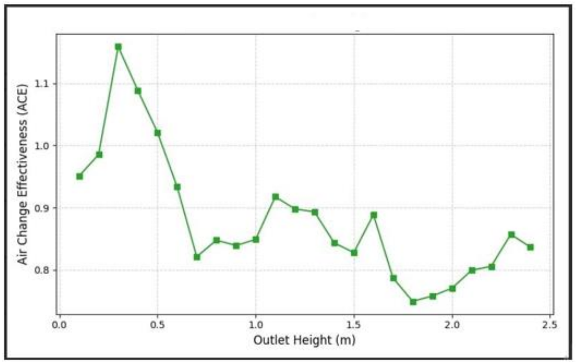

Figure 7. ACE versus cooler's supply outlet height

3.3 Multi-Objective Optimization and Pareto Analysis

The Pareto analysis performed in this study showed that only a limited range of outlet heights satisfied the thermal comfort constraint while simultaneously providing high ventilation effectiveness as shown in Figure 8. Among these configurations, the outlet height of 0.3m produced the highest ACE value while maintaining PMV within the acceptable range. This finding is in consonance with the work of [3] who reported that in an office building conditioned by façade-mounted hybrid downdraft evaporative cooler, the optimum height of the cooler supply outlet height is 0.4m.

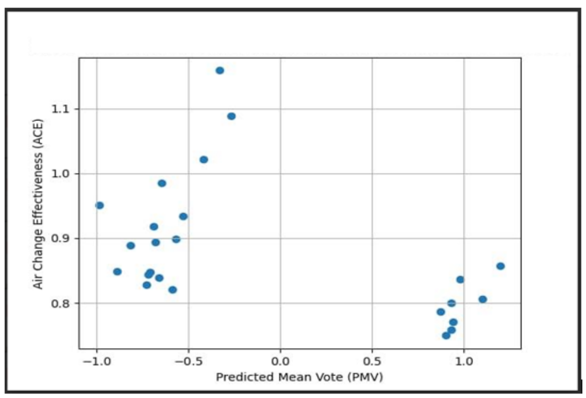

Figure 8. Pareto trade-off between thermal comfort and ventilation effectiveness

4 CONCLUSIONS

This study investigated the influence of façade-mounted downdraft evaporative cooler outlet height on indoor thermal comfort and ventilation performance in an office building using computational fluid dynamics simulations implemented in Design Builder. The outlet height was systematically varied from 0.1m to 2.4m in order to examine its effect on the airflow distribution within the occupied zone. Thermal comfort was evaluated using the PMV index while the indoor air quality was assessed using ACE. The results revealed that the vertical outlet height of the cooler significantly influences airflow distribution and thermal conditions within the office space. The Pareto optimization method employed revealed that the optimum cooler's supply outlet height of 0.3m above the floor level produced the optimum comfort with PMV of -0.33 while at the same time enhances indoor air quality with ACE of 1.1588. This demonstrates that, in hot and dry climates, appropriate positioning of evaporative cooler outlets on façade of office buildings or other buildings with similar characteristics can substantially improve indoor environmental quality.

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