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Thermal Comfort Simulation for Cold Air Distribution Systems by a User Defined Predictive Mean Vote Index

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ABSTRACT

Thermal comfort is the prime purpose of Heating, Ventilation, and Air Conditioning (HVAC) systems in the indoor environment. Ideally, providing the best thermal comfort with the minimum energy consumption of HVAC systems is most desired. Cold air systems are proven to put forward energy savings with relatively low supply air temperatures ranging between 4 ℃ and 10 ℃, compared to conventional systems, which supply air at around 16℃. However, cold air systems are rarely applied due to cold draft formation and thermal comfort concerns. Thermal comfort is established by the interactions of convective and radiative heat transfers within the zone. These conditions of the thermal environment are incorporated with the occupants' characteristics of metabolic rate associated with their activities as well as clothing (thermal insulation) into the well-established Predicted Mean Vote (PMV) index [1]. Cold air systems were narrowly studied in terms of averaged Air Diffusion Performance Index (APDI), leading to a false representation of the system compatibility to provide thermal comfort. This paper develops a User Defined Function (UDF) combined with a Computational Fluid Dynamics (CFD) model to accurately represent thermal comfort conditions of cold air systems. The model is tested for the two-dimensional indoor zone. A scenario of variable indoor conditions is considered to identify the PMV index on the cell-sized scale in the order of 9 mm x 9 mm of the airflow in the office space. The PMV index is considered for a k–ε turbulence model for indoor airflow [2]. In order to test the validity of the two-dimensional model estimations, PMV indices for cells are compared with the results from Center for the Built Environment (CBE) thermal comfort tool against ASHRAE-55 standard. The CFD model developed has shown the effectiveness of cold air systems for the occupied zone layer with applications of PMV based reduced-order control systems.

Keywords: Thermal comfort; CFD; Indoor environment; PMV index; User defined function (UDF); Steady state model.

NOMENCLATURE

Abbreviations	
APEN	Applied Energy
ASHRAE	American Society of Heating, Refrigeration and Air Conditioning
CBE	Center of the Built Environment
CFD	Computational Fluid Dynamics
CSP	Computer Simulated Person
HVAC	Heating, Ventilation and Air Conditioning
ISO	International Organization for Standardization
UDF	User Defined Function

1. INTRODUCTION

As Coronavirus stay home orders have been enforced globally, the approximation of time percent spent in the built environment has increased rapidly from 90% to

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nearly 100%, where many of these homes are mechanically air conditioned. Consequently, the built environment should be ultimately designed to support its occupants' health and comfort, especially with the increase of annual extreme weather days frequency. Severe physical impacts on individuals are induced by recurring thermal discomfort, such as agitation and lack of concentration [3]. Due to the concerns of indoor air quality with COVID-19 spread, it is expected that new design guidelines will be developed to promote localized / personalized air control. Therefore, it's imperative to develop a deeper understanding of the finer scale thermal comfort distribution and air movement inside offices and mechanically conditioned spaces.

The question is, what constitutes thermal comfort conditions? The Existing standards have identified that thermal comfort for individuals is achieved when thermal equilibrium occurs between human body and the surrounding thermal environment, provided that there is no local discomfort caused by asymmetries of vertical temperature, nonuniform radiant temperature, air velocities and air pockets [4]. Conversely, human thermal comfort is not determined solely by temperature of the air, but by the perceived temperature at the skin responding to the surrounding conditions. Therefore, thermal comfort indices are vital tools to quantify empirically thermal comfort perception for occupants combined with interactions of convective and radiative heat transfers within the zone. PMV developed by Fanger [1], is a commonly used thermal comfort index that accounts for physiological variables and indoor air variables. Given the elevated surface temperature of the human body, the airflow sheds around the human body initially as laminar at the feet level then develops into turbulent around the upper body, leading to regions of airflow recirculation forms around the shoulders and head [5]. Respectively, a temperature gradient is formed around the human body and can be easily missed with averaged PMV index valued, contributing to faulty traditional HVAC system design. An experimental study, using infrared thermography flow visualisation for the thermal plume of human body, concluded that only 1/8 of 40 L/s airflow produced by the human body is evacuated by the HVAC outlet, contributing to poor indoor air quality. Various experimental studies relied on field measurements to evaluate indoor thermal performance [6, 7], to avoid the complexity of computational models; however, this required installations of expansive equipment for measurements.

Cold Air Distribution systems supply air at relatively low temperatures ranging between 4°C and 10°C, rather than conventional cooling systems which supply air at 16°C. When introducing supply air with lower temperature, the supply volume, needed for a given air conditioning load, is reduced proportionally [8]. Thus, the supply duct size, fan speed and energy consumption will be reduced. According to Youssef et al. [2], the supply airflow for a cold air distribution system can be reduced 30-40% compared to conventional cooling systems. Cold Air Distribution thermal comfort criteria have been evaluated based on Air Diffusion Performance Index (ADPI) [9-11], which considers averaged air velocity as a sole criterion of thermal comfort. This fragmented scope led to misjudging the effectiveness of such an air supply system. In addition, the question of how these systems truly impact the occupants remains unanswered.

This paper develops a computational method to quantify the efficiency of cold air systems from the thermal comfort perception of the occupants. CFD is employed to investigate the air temperature, air velocity and PMV index calculations for a 2D office space with a detailed Computer Simulated Person (CSP), using User Defined Function (UDF) to accurately represent the thermal comfort distribution. Hence, energy savings can be achieved with acceptable thermal comfort conditions for cold air systems.

2. METHODS

2.1 Case Study Room Model

Computational Fluid Dynamics simulations are reliable for indoor air characteristics evaluation. A simplified 2D office geometry is constructed as a representative of the indoor environment. The physical model used is shown in **Figure 1**. The room under consideration is 3 x 3 x 2.6 m, representing the dimensions of potential one-occupant office. Air is supplied through a 0.3 m ceiling inlet and discharged by a 0.5 m ceiling outlet, both placed on the ceiling. To simulate summer working environment in Melbourne, Australia, exterior and interior walls represented the conductive and convective heat transfer processes into the zone. No slip and no penetration boundary conditions were applied. The façade consisted of an exterior wall with temperature of 27 ℃. The interior wall temperature was modelled to be 23 °C. The ceiling and the floor were considered adiabatic surfaces for simplification. A detailed seated CSP was used to ensure the accuracy of near-occupant regions PMV values [12] placements were based on the Craven and Settles experiment of detecting human thermal plume, with a distance between the mannequin's head and ceiling of 1.21 m [5, 13]. The heat generation of the mannequin is 63.96 W/m^2, contributing to the total thermal loads of the office [14].

Unstructured quadratic dominant mesh was used to discretise the spatial domain, in order to capture the irregular geometrical features of the human body. The grid density was adjusted with greater density around the boundary layers. Inflation layers were specifically included for the CSP to detect the critical PMV index, shown in **Figure 1**. To ensure accurate convergence, grid sensitivity study was performed with three grids in the simulation as shown in **Table 1**. To reduce computational cost, Grid 3 was chosen with acceptable orthogonality quality.

Table 1 Grid sensitivity analysis

Grid	Cell number	N_x (m)	Outlet $V_{average}$ (m/s)
1	14,163	0.03	0.1250
2	28,240	0.02	0.1241
3	111,366	0.009	0.1323

To better understand the air distribution homogeneity and zone occupancy variation, the finite volume solver ANSYS Fluent 20 R2 is used to solve the governing equations and simulate the airflow velocity and temperature profile, with the RNG $k-\varepsilon$ turbulence model. The Coupled scheme is used to solve the coupling between the temperature and velocity, where the coupled scheme obtains a robust and efficient single-phase flows, with superior performance compared to the segregated solution schemes [15].

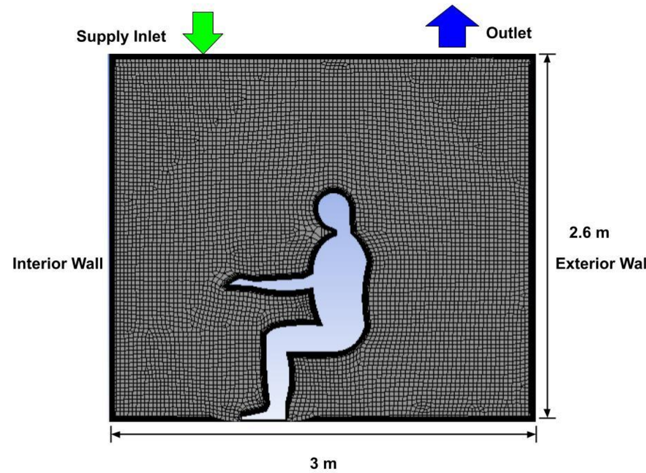


Figure 1 2D Geometry of a single office space

The nonlinear terms are modelled using second order upwind algorithms [16]. Convergence conditions are set up to reduce the error of the difference between the intermediate solution of the discretisation partial differential equations and the exact solution of the algebraic system equations. In addition, the convergence conditions mean the absolute criteria for the maximum difference between two conductive iterations for the flow variables. To determine the iterative convergence conditions, the relative residuals stemming from solving the governing equations is determined to be 10^{-3} for continuity, X and Y velocities, k and ε. For the energy equation residuals, the convergence condition is 10^{-6}. The simulation was performed using 8 CPUs with 32 GB RAMS, 512 GB disk and NVIDIA P40_4Q GPU (4GB VRAM).

2.2 Predicted Mean Vote

According to Fanger's continuous investigation to quantify thermal comfort, PMV is determined by six factors: zone air temperature, mean radiant temperature, air velocity and relative humidity; in addition to clothing insulation and human activity. It can be quantified by a scale of 7 points:- 3 (Cold), - 2 (Cool), - 1 (Slightly cool), 0 (Neutral), +1 (Slightly warm), +2 (Warm), +3 (Hot) [1]. PMV indices are categorised into three classes of satisfaction according to ISO 7730 standard: Class A: PMV ranged from - 0.2 to +0.2 showing the highest satisfaction to the environment; Class B: - 0.5 < PMV < + 0.5 for the moderate satisfaction; Class C: - 0.7 < PMV < + 0.7 for minimum requirement for thermal comfort [17]. Several other analytical methods can be used to quantify the thermal comfort, such as Predicted Percentage Dissatisfied (PPD) index, the (Centre of Built

Environment) CBE comfort model, Daily Discomfort Score (DDS) and adaptive thermal comfort model [18] [19, 20].

To calculate the PMV index, offices are identified as the targeted controlled spaces with determined representative occupant's metabolic rate of 63.96 W/m2 for typing while seated activity, and insulation of 0.58 clo for a seated individual with trousers and short sleeve shirt. These values are determined from AHRAE 55-2017 [14]. Metabolic rate represents the heat produced by occupants, depending on their physical activity. Averaged metabolic rate is used to present occupants with metabolic rate difference less than 0.1 met. ASHRAE 55-2017 recommends using time-weighted averaging for varying metabolic rate, except for activities that persist for more than one hour. For clothing insulation, averaged clothing insulation is used to represent multiple occupants, provided by AHSRAE 55 -2017 [14]. The PMV model is presented as the following:

$$\begin{aligned} PMV = {} & [0.303 \times \exp(-0.036M) + 0.028] \\ & \times \{M - W \\ & - 3.05[5.733 - 0.007(M - W) - Pa) \\ & - 0.42(M - W - 58.15) \\ & - 1.73 \times 10^{-2} \times M(5.867 - Pa) \\ & - 0.0014M(34 - ta) \\ & - 3.96 \times 10^{-8} fcl(tcl + 273)^4 - (tr + 273)^4] \\ & - fclhc(tcl - ta)\} \end{aligned}$$

$$\begin{aligned} tcl = 35.7 - 0.028(M - W) & \\ - Icl\{3.96 \times 10^{-8} \times fcl[(tcl + 273)^4 & \\ - (tr + 273)^4] + fclhc(tcl - ta) & \end{aligned}$$

$$hcl = \begin{cases} 2.38|tcl - ta|^{0.25}, & for\ \ 2.38|tcl - ta|^{0.25} > 12.1\sqrt{var} \\ 12.1\sqrt{var} \quad , & for\ \ 2.38|tcl - ta|^{0.25} < 12.1\sqrt{var} \end{cases}$$

$$fcl = \begin{cases} 1.00 + 1.290Icl, & for\ Icl \leq 0.078m^2.K/W \\ 1.05 + 0.645Icl, & for\ Icl > 0.078m^2.K/W \end{cases}$$

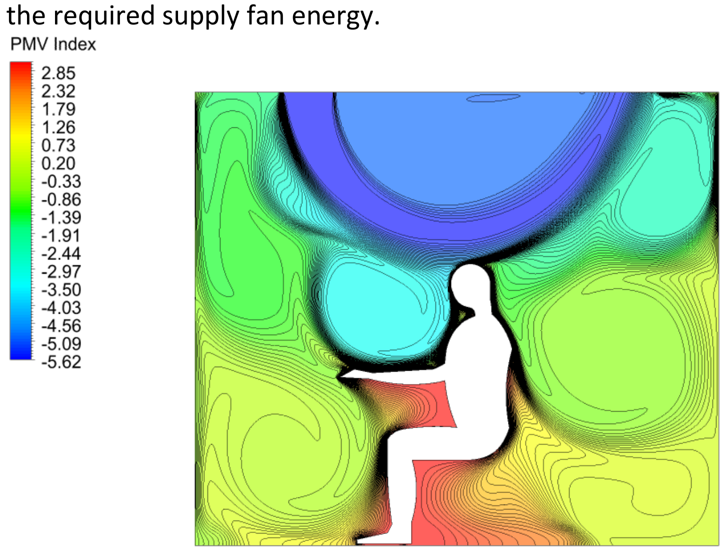
Where M (W/m^2) is the metabolic rate produced by the body during certain activity; W (W/m^2) is effective mechanical power by movement; $hcl\left[\frac{W}{m^2}.K\right]$ is convective heat transfer coefficient which can be calculated using temperature of clothing and air temperature or the air velocity; tcl (℃) is clothing surface temperature which can be obtained from ASHRAE 55; tr (℃) is mean radiant temperature; ta (℃) is air temperature, provided by the CFD simulation; fcl is clothing surface area factor which also can be obtained from ASHRAE 55 according the assumptions of the scenario season; Icl is the clothing insulation; and var $\left(\frac{m}{s}\right)$ is the relative air velocity, provided by the CFD simulation.

2.3 *PMV User Defined Function*

Introduced as a C function, a PMV User Defined Function (UDF) is dynamically loaded with the ANSYS Fluent solver to adjust the output values and calculate the thermal comfort distribution in the indoor zone. The PMV UDF is interpreted from the C source file, and this occurs during the runtime of the model. To specify the function of PMV calculation, DEFINE-ADJUST macro is selected to modify the air flow variables over the domain of air distribution. The air distribution domain argument provides access to all cells in the mesh, with specific focus on every cell's total temperature, X and Y velocities, and total pressure. This macro enables the PMV UDF to be executed at every iteration and time step. After interpreting the PMV UDF on the ANSYS Fluent solver, the PMV values needs to be stored by linking the UDF to an allocated user defined memory, through using C_UDMI macro.

3. RESULTS

Figure (2) indicates the PMV distribution for introducing the supply air with 10℃ and 0.2 m/s, and total heat loads of 68.2 W/m^2 boundary conditions. These conditions were recommended by [2] to achieve an average room velocity of 0.323 m/s and an ADPI of 98.2%. According to the study, cold draught formation is reduced with the assumption of perfectly mixed air. These supply air conditions are comparatively less than conventional supply air velocity, consequently reducing the required supply fan energy.



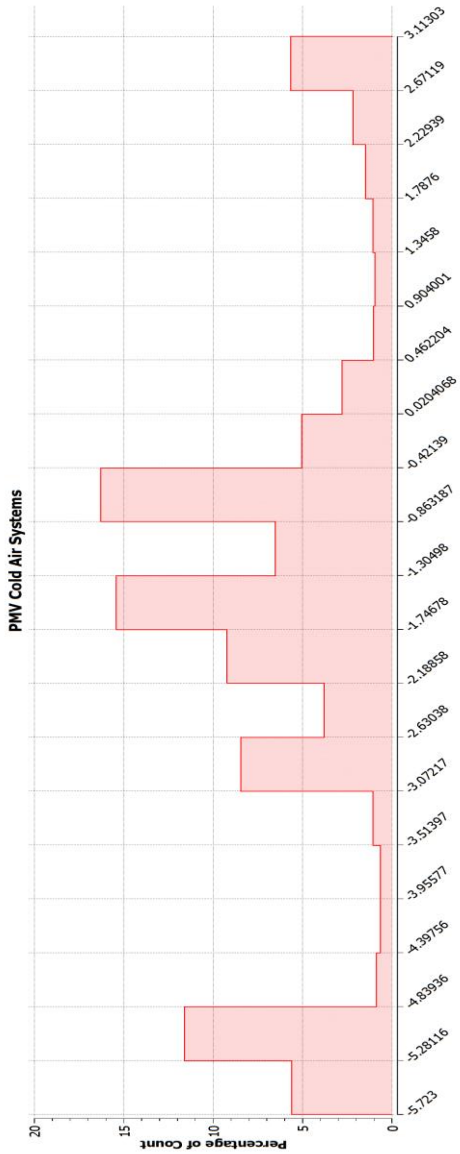


Figure 2 PMV Distribution Profile

To validate the results of the PMV UDF values, random cells' total temperature and air velocity magnitude from the model were input into the Center of the Built Environment CBE thermal comfort tool [21], to receive PMV values varying by %0.64 of the calculated PMV values in the CFD model.

Referring to the PMV profile, approximately 22% of the room discretised cells fell in the ISO class A and B PMV index range for highest and moderate thermal satisfaction, and 7% of the cells fell in the ISO class C minimum satisfaction. 71% of the airflow indicated poor thermal comfort performance. Specifically, thermal comfort indices ranged from –4.82 to –2.69 around the head area, pinpointing the severe cold thermal discomfort impacting the upper body. Cold air systems thermal comfort performance was compared to conventional cooling systems; however, due to the page limit of conference publication, the results will be reported in an extended version.

For a typical PMV reading, averaged temperature of 22.47 ℃ and air velocity of 0.0367 m/s of the office conditions reflect a reading of -0.73, which meets approximately the minimum acceptable thermal comfort conditions, using CBE thermal comfort tool. Averaged air temperature and velocity readings lead to misjudging the thermal comfort distribution of the model. In a single occupant office, one sensor is installed on the interior wall indicating acceptable thermal comfort conditions. From **Figure 2**, variations of thermal comfort distribution are better demonstrated, leading to potential discovery on how to improve the thermal comfort in the built environment.

4. CONCLUSIONS

This paper developed a method to quantify the thermal comfort sensation of cold air systems, in order to maximise the benefits of energy consumption reduction; however, the PMV CFD results showed that the cold air system caused thermal discomfort for a single occupant office space. The methods used demonstrated that PMV CFD models can be a reliable tool to judge the effectiveness of HVAC systems. These findings are used to further develop a thermal comfort based adaptive control system, that reduces the nonlinearity of the PMV calculation, leading to coupling the benefits of energy consumption reduction and acceptable thermal comfort conditions.

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