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Comparative Analysis of Thermal Comfort Surveys with CFD Simulations

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Abstract: In this paper three dimensional numerical simulations are carried out for air-conditioned room environments having wall mounted units for two cases. The actual solar gain using P1 model using the solar irradiance values of day of survey along with actual geo-coordinates and time are incorporated in both cases for accurate solar solstice whereas the air conditioning units air flow details and temperatures are taken from OEM data. The solution is solved for PMV contours in post processing and local PMV values are compared with the actual measurements from instruments, and it is concluded that simulations of solar gain cases from sun exposed ceiling yield closer results as compared to the case of solar gain from the windows.

Keywords: Thermal Comfort, Workplaces, CFD Indoor Simulations, Thermal Sensation Vote, Official Buildings

1. Introduction

Air- Conditioning consumption accounts for 10% of global electricity consumption (Ali *et al.*, 2022). To check the validity of effectiveness of any air-conditioning environment, the ASHRAE Standard 55-2017, ISO 7730:2005 and EN 16798-1-2009 standards point towards thermal comfort surveys, however, these surveys can only be conducted when the building has already been built, AC systems installed and occupied by its residents translating into accrual of capital as well as operating costs.

Building simulations featuring computational fluid dynamics can be used at design stage, however, the accuracy of such software packages is largely dependent on careful problem setup and correct boundary conditions initializations, as multiple methods and equations need to be solved in correct order to obtain an agreeable solution. The need for a robust yet economical CFD methodology warrants a comparative study of field collected real time results. Nevertheless, the rigorous literature review done for the past four decades of indoor CFD simulations rarely addressed this concern.

2. Literature Review

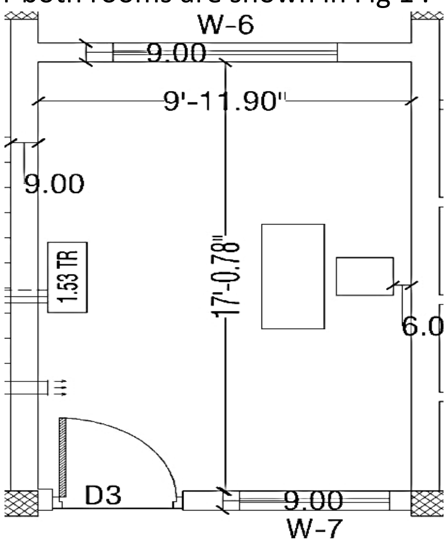
The pioneering study which evaluated thermal comfort through numerical simulations featured 18 cases for 2 buildings and simulated the enclosed space in 3d using K-Epsilon (k-ε)Turbulence Model . Converged solutions for Air Velocity, Temperature , PMV and PPD were found using Under Relaxation Technique.(Yuan *et al.*, 1992) A three dimensional CFD code called Vortex was and the results obtained from CFD code were compared with each other as mechanically ventilated and naturally ventilated however the field assessments and measurements were not the part of scope of this study. (Awbi and Gan, 1994) .

Another research used experimentally validated 3d CFD simulation codes to investigate the effect on PMV and PPD by different mechanical ventilation strategies utilizing simple algorithm which simultaneously solves Navier Stokes Equation, Continuity Equation, and Energy Equation with k- epsilon (k-ε) Turbulence model. (Gan, 1995)Similarly commercial solver Fluent 5.3 was used to simulate the air temperatures and velocity for the Lecture Hall with furniture and all energy sources using epsilon (k-ε) Turbulence model. Thermal sensation was subjectively assessed using questionnaires and objectively measured and calculated as PMV and PPD, however, the aforementioned thermal indices were not simulated in CFD simulations. (Cheong *et al.*, 2003)

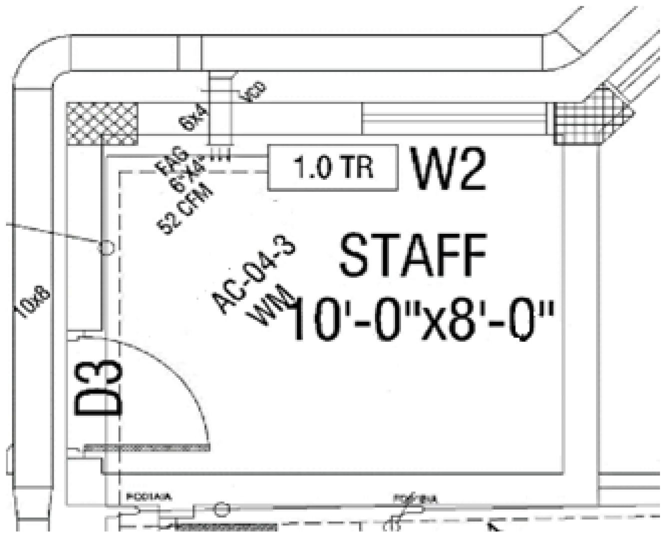
A research studied the effect of aspect ratio of windows sizes of naturally ventilated office buildings on thermal comfort using CFD codes and Network Model and found optimum window opening area ensuring maximum thermal comfort without compromising the strength of buildings(Ravikumar and Prakash, 2009)**.** Another study also conducted simultaneous experimental and simulation studies for the assessment of thermal comfort for a sub-tropic region office utilizing the cooling ceiling for mechanical ventilation. CFD simulation were subsequently reused to address temperature stratification and thermostat strategy was revised so as to achieve thermal comfort and energy savings up to 8%. Humidity levels were assumed in this study(Chiang, Wang and Huang, 2012)**.** A latest study compared 3 operative temperatures of 21^{o}C, 23^{o}C and 25^{o}C by calculating PMV and PPD values through CFD simulations using Ansys Fluent Software Package for a workspace(Ismail *et al.*, 2021)

3. Study Setup

In the current study, two cases were simulated for finding Predictive Mean Vote contours to access thermal comfort through CFD Simulations and subsequent comparison with field assessments. Case A was based on an Office located on the 1st floor, whereas, Case B was based on an office on the 3rd Floor of G+3 academic building of Islamabad. The HVAC layouts for both rooms are shown in Fig 1 .



AC Layout for Room of Case A



AC Layout for Room of Case B

Fig 01: Air Conditioning (AC) Layouts for Case A and Case B.

Room A had two windows named W-6 ,W-7 and a door D3. W6 was exposed to sun, whereas, W7 and D3 were opened in an air-conditioned environment. Room A was provided with 1.5 TR Wall Mounted Unit with a diffuser for fresh air providing 50 CFMs. Room B had one door and one window both opening in the corridors. Being on the 3rd Floor, the ceiling of room was exposed to sun. Room B was provided with 1 TR Wall mounted unit with a diffuser for fresh air providing 52 CFMs.

4. Assumptions

Following assumptions were made for CFD simulations.

1. Heat from electrical items such as lights, PCs, laptops, scanners and printers was ignored, and furniture in the room was not modelled. (Horikiri, Yao and Yao, 2015)
2. For surfaces exposed outside to air-conditioned environment air temperature of 26^{o}C was taken as a thermal boundary condition and for the surfaces exposed to the sun outside, the ambient air temperature was taken as a thermal boundary condition. (Bonefacic, Wolf and Frankovic, 2015)
3. Air conditioner's operational data like volumetric flow rate and rated capacity was taken from the catalogues provided by OEM (DC INVERTER MULTI VRF Service Manual T1/R410A/50Hz, n.d.).

4. Governing equations and mathematical modelling

Continuity equation, energy equation and radiative heat transfer equations were used for mathematical modelling of problem. Realizable k-ε model using standard wall functions for near wall treatments was used for turbulence modelling, whereas, P1 Model using solar calculator with actual geo-coordinates and solar irradiance values was used for simulating solar gain.

Solar Gain Parameters	Case A	Case B
Direct Normal Irradiance	324.63 W/m^2	403.79 W/m^2
Direct Horizontal Irradiance	0.85 W/m^2	231.1943 W/m^2
Outside Air Temperature	35.38^{o}C	23.83^{o}C

5. Results and Discussions

5.1. Field Assessments

Calibrated instruments were setup on a medical trolley for field surveys and data was logged into PC with One Minute Interval for Ten Minutes as shown in Fig 06. During this time Subjective assessments were also conducted with the help of questionnaires. Air Temperature, Air Velocity, Globe Temperature, Humidity was measured through instruments whereas clothing and activity level data was provided from respondents.

Mean Radiant Temperature was further calculated from Globe Temperatures and all 10 minutes data was averaged was plugged in PMV Equation (1) to obtain Local PMV values and corresponding Local PPD

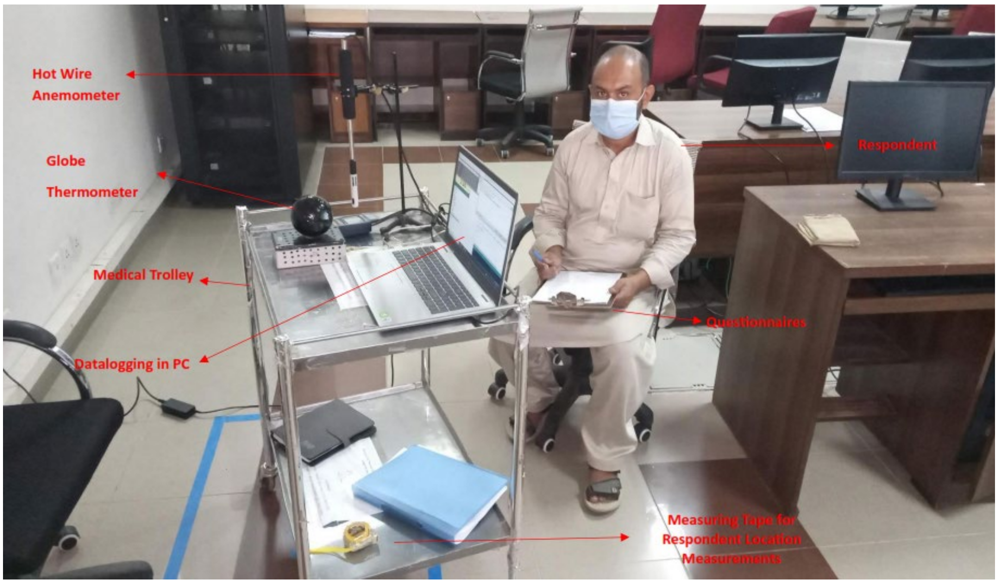


Fig 06.Field Assessment Typical Setup with Instrumentation and Respondent

values were calculated from Equation (2).(Fanger, 1970)

$$PMV_{Local} = [0.303e^{-0.036.M} + 0.028].L \quad .....(1)$$

$$PPD_{Local} = 100 - 95e^{(-0.03353(PMV)^4 - 0.2179(PMV)^2)} \quad .....(2)$$

5.2. CFD Results

During field surveys the dimensions for position of equipment was noted and later in CFD simulation same position PMV and PPD is compared with PMV and PPD calculated from field surveys as shown in Table 01 and Fig 04 for Case A and Fig 05 for Case B.

Thermal Indices	**Case A**		**Case B**	
	Simulations	**Field Surveys**	**Simulations**	**Field Surveys**
PMV (Maximum)	2.20	-	0.45	-
PMV (Minimum)	-3.0	-	-1.28	-
Coordinates for Local PMV	0.77,1.01,0.57	-	2.89,1.04,-1.43	-
PMV_{Local}	-0.12	0.13	0.72	0.85
PPD (Maximum)	95.0	-	39.3	-
PPD (Minimum)	5.0	-	5.0	-
PPD_{Local}	5.3	5.4	5.1	20.4

Table 01: Comparison of PMV and PPD found from simulations and field Surveys of Case A

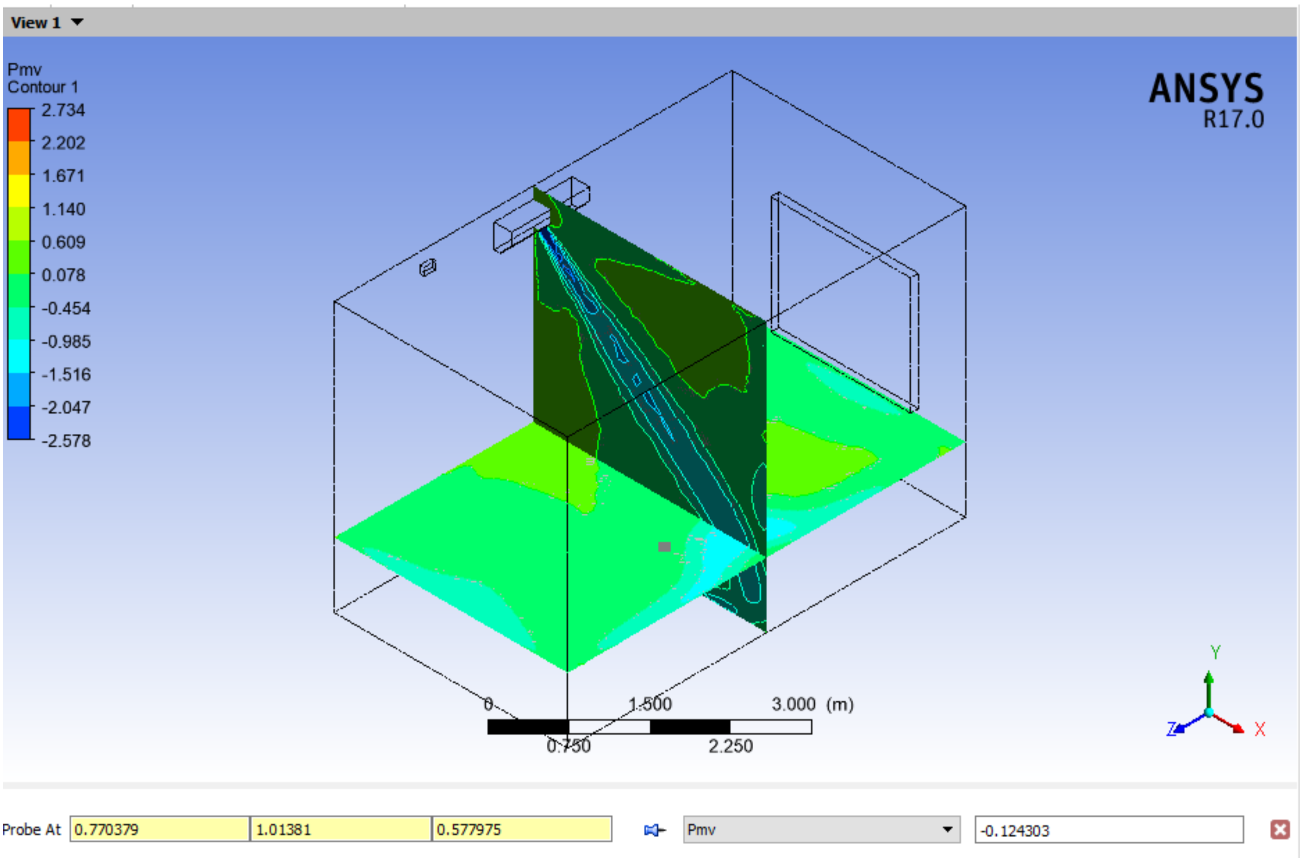


Fig 04. Local PMV at x= 0.77,y=1.01 and z=0.57 from simulation results for Case A

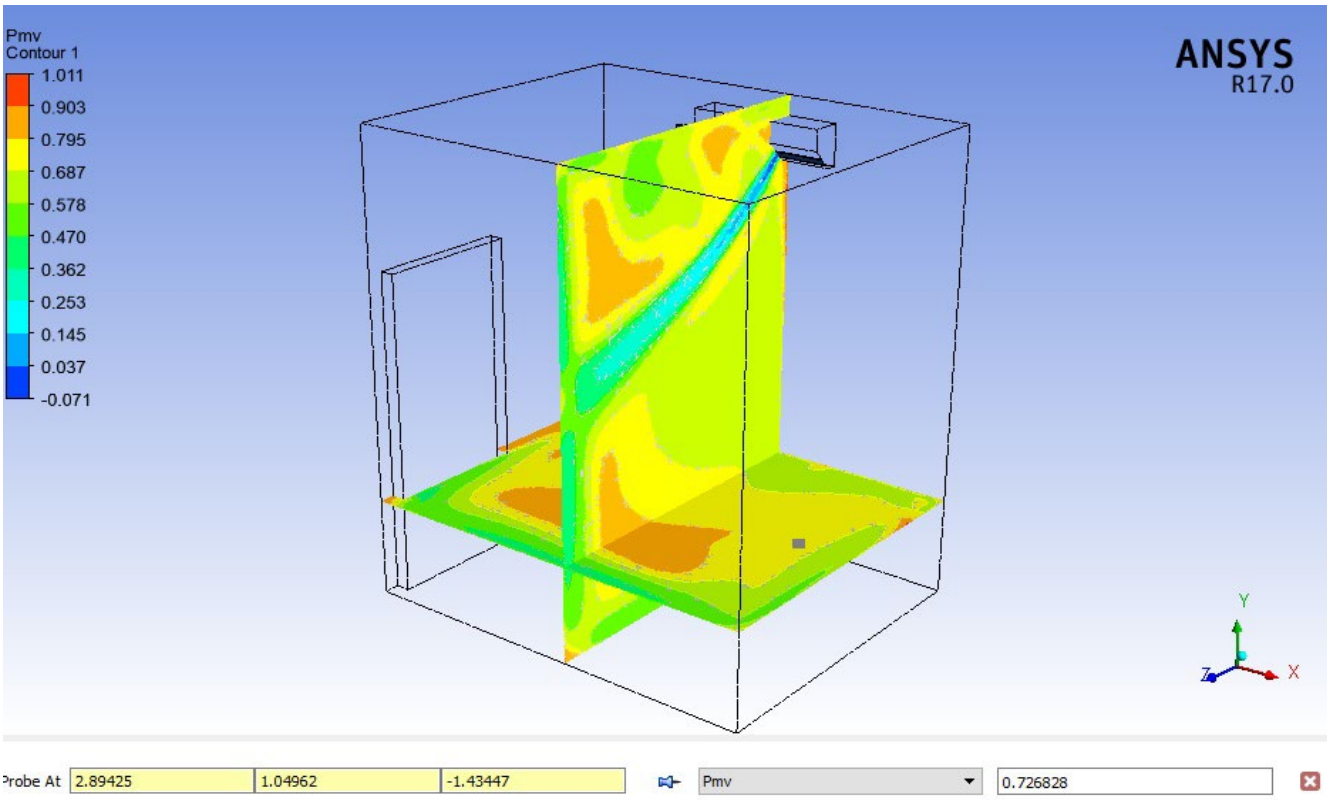


Fig 05. Local PMV at x= 2.89, y=1.04 and z=-1.43 from simulation results of Case B.

The PMV contour is dispersed for Case A varying between a range of 2.2 to –3.0, however, the variation is less in PMV contours for Case B as 0.45 to -1.28. The local PMV for both cases show an inclination towards cooler side of thermal sensation scale whereas in actual field measurements the calculated PMV for both Cases was higher. The local PMV for Case B is closer to actual data as compared to Case A.

Room A was situation on first floor of eastern wing whereas room B was situated on the 3rd floor of central wing which is the topmost floor of the building. The sun facing component for Case A is Wall with a Window whereas the sun facing component for Case B is the roof.

The installed capacity in Room A is 1.5 TR for 170 Sq-ft whereas the installed capacity in Room B is 1 TR for 75 Sq-ft making the load factors as 113 Sq-Ft/TR and 75 Sq-ft/TR respectively.

Both Rooms had individual thermostat controls for occupants.

The lower values of PMV values calculated in contrast to field survey calculated PMV values can be attributed to the mechanical wear of air-conditioners blower fans over the years hence providing flow rate lower than that of rated values as noted in OEM Manuals.

6. Conclusion and Recommendations

Thermal comfort evaluated from CFD simulations with real time field survey experimental data and it is found that simulations yield better results for cases where ceiling is exposed to sunlight as compared to the cases where only walls are exposed to sun. It was also found that within a single room, Predictive Mean Vote experiences variations in PMV values. Better results can be drawn for future studies if:

- ✓ Actual mass flow rate for the air-conditioners vents is considered.
- ✓ Actual mass flow rate for the diffuser vents is considered.
- ✓ Actual temperature for the AC outlet vent is taken.
- ✓ Actual temperature for AC diffuser vent is taken.

- ✓ Actual values for thermal resistance, absorptivity, transmissivity and emissivity is calculated for materials like glass and wall and incorporated in CFD simulations
- ✓ Wall temperatures are measured and incorporated in the studies.
- ✓ Heat sources from office appliances are considered.
- ✓ Boussinesq approximation model is used to consider the buoyancy effects.

7. Nomenclature

ASHARE	American Society Of Heating, Refrigerating And Air-Conditioning Engineers
CFD	Computational Fluid Dynamics
CFM	Cubic feet per minute
EN	European Norm
G+3	3 Floors Excluding Ground Floor
HVAC	Heat Ventilation and Air Conditioning
L	L is a function of Activity Level, Air Temperature , Mean Radiant Temperature, Air Velocity, Relative Humidity and Clothing Insulation Level.
M	Metabolic Rate in W/m^2
ISO	International Organization for Standardization
OEM	Original Equipment Manufacturer
PMV	Predicted Mean Vote
PPD	Predicted Percentage Of Dissatisfied
SIMPLE	Semi-Implicit Method For Pressure Linked Equations
Sq-Ft	Feet Squared
TR	Ton Of Refrigeration
TWh	Terawatt-Hour
e	Euler's Number 2.718281828459045

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