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COMPARATIVE CFD ANALYSIS ON CONDITIONED AIR FLOW AND TEMPERATURE DISTRIBUTION IN METRO TRAIN FOR DIFFERENT CITIES OF INDIA

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Abstract— The thermal comfort of passengers is a priority concern in India's metro rail system as it affects both psychological and physiological character while traveling in the train compartment. The principal factors affecting human thermal comfort are the air temperature, its relative humidity, the mean radiant temperature and the relative air velocity which varies due to various human load inside cabin. Hence in this work, the CFD tool ANSYS FLUENT 18 software package was used for numerical analysis of cooling load, air flow and heat transfer in our virtual 2-D and 3-D model of the metro train's passenger cabin/compartment so as to analyze the level of comfort that passengers experience in different climate zone cities mainly Chennai, Delhi and Kolkata. This work also tries to validate how the required temperature and air distribution for passengers is maintained for different scenarios like normal time and peak time inside a metro rail coach for the above-mentioned cities. The result, analyzed and discussed in terms of temperature and velocity fields for different scenarios/cases, shows an adequate behavior from the passengers' comfort point of view.

Keywords- thermal comfort; metro train; HVAC; computational fluid dynamics

I. INTRODUCTION

The most common term for referring conditioned air flow and temperature distribution is HVAC. HVAC system elaborates itself as Heating Ventilation and Air Conditioning. The system when implemented for the metro rail compartment becomes more complex due to less space and have a variational load based on the human's sudden entering the zone. In order to maximize the passenger carrying capacity, longitudinal seating arrangement is adopted in metro train. The whole train shall be vestibuled to distribute the passengers evenly in all coaches. For the even distribution of air, the ducting system is implemented that carries the air from the either side of the package unit into the compartment and then delivered by the diffusers as designed by the convectional method. Thus, ensuring thermal comfort inside metro train compartment involves either defining the optimal temperature and acceptable thermal comfort zone, as perceived by passengers, or estimating the thermal sensations of passengers under certain conditions.

In previous literatures, numerical and experimental studies for the air flow and temperature was carried out for limited human load. ASHRAE standard 55 defines thermal comfort as "that state of mind which expresses satisfaction with the thermal environment" [1]. For instance, Oakland did a study on metro stations and carriages in London. They investigated the thermal comfort conditions as well as the air particulates in the studied systems. They realized that air temperature and humidity as well as particulate distribution, significantly affect the thermal comfort conditions of the passengers [2]. Ishihara and Hara measured the air-flow velocity distributions in the car compartment by visualization and then compared them with their numerical simulation results [3]. Several researchers have investigated human comfort level using different viewpoints and modeling schemes including psychological modeling for sitting and standing posture [4], Thermal comfort in hospitals [5], Numerical investigation on the airflow characteristics and thermal comfort in buoyancy- driven natural ventilation rooms [6], Thermal comfort of the surgical staff in the operating room [7], Experimental investigation on transient natural ventilation driven [8], Study on the Thermal Accumulation and Distribution Inside a Parked Car Cabin [9], Interior Airflow Simulation in Railway Rolling Stock [10] and so on.

Hence in this work, we tried to validate how the required temperature and air distribution for passengers was maintained for different scenarios like normal time and peak time inside a metro rail coach for different cities of India namely Chennai, Delhi and Kolkata. During the design process or during usage, the air should be given in low air speeds and homogeneously for user's comfort. For this, virtual 2-D and 3-D model of the metro train's passenger cabin/compartment was modeled using Creo and Pro/E and was analyzed using CFD tool ANSYS FLUENT 18 software packages since Fluent software solves continuum, energy and transport equations numerically with natural convection effects and boundary conditions. Based on the available approach for the estimation of general thermal comfort index; the presented work aims to compare three proposed thermal comfort cases/scenarios so that the results, analyzed and explained in terms of temperature and air velocity, showed an adequate behavior from the passengers' comfort point of view.

II. OBJECTIVES AND COMPUTATINAL METHODLOGY

A. *Objectives*

1. Work involves simulation of air flow and temperature distribution characteristics in passenger compartment for different design scenarios.
2. To determine cooling load calculations manually using ASHRAE Data base and analysis readings.
3. Design optimization of Duct layout on critical areas through CFD.

B. *Challenges*

1. Remodeling CAD model of metro train compartment so as to perform air flow and temperature distribution analysis.
2. Utilizing applicable cooling load and air flow equations.
3. Identifying current duct model in metro train compartment to simulate air flow.

C. *Design Methodology*

We created models in Creo and Pro/E by building features. These features have intelligence, in that they contain knowledge of their environment and adapt predictably to change. Each feature asks the user for specific information based on the feature type. For example, a hole has a diameter, depth, and placement, while a round has a radius and edges to round. The essential difference between Pro/E and traditional CAD systems are the models created in Pro/E exist as 3D solids. Other 3D modelers represent only the surface boundaries of the model. Pro/E models the complete solid. This is not only facilitating the creation of realistic geometry, but also for accurate model calculations, such as those for mass properties.

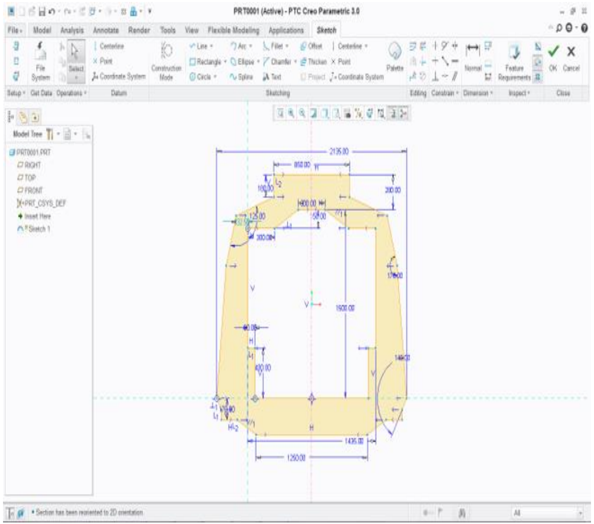


Figure 1. 2D Model of Metro Compartment

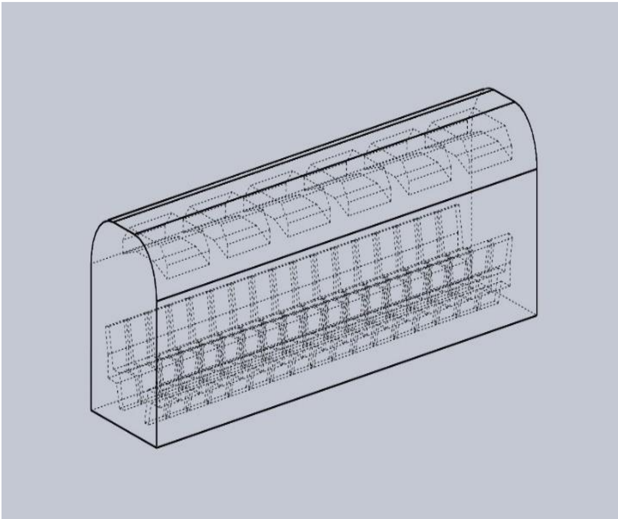


Figure 2. 3D Model of Metro Compartment

D. *Analysis Methodology*

The ANSYS Program has many finite element analysis capabilities, ranging from a simple, linear, static analysis to a complex nonlinear, transient dynamic analysis. The ultimate purpose of a finite element analysis it to recreate mathematically the behavior of an actual engineering system. In other words, the analysis must be an accurate mathematical model of a physical prototype. In the broadest sense, this model comprises all the nodes, elements, material properties, real constants, boundary conditions, and other features that are used to represent the physical system. The strength of parametric modeling is in its ability to satisfy critical design parameters throughout the evolution of a solid model. The concept of capturing design intent is based on incorporating engineering knowledge into a model. This intent is achieved by establishing features and part relationships and by the feature dimensioning scheme. An additional postprocessor, POST26, enables us to evaluate solution results at specific points in the model as a function of time.

A typical ANSYS analysis has three distinct steps:

- Build the model
- Apply loads and obtain the solution
- Review the results.

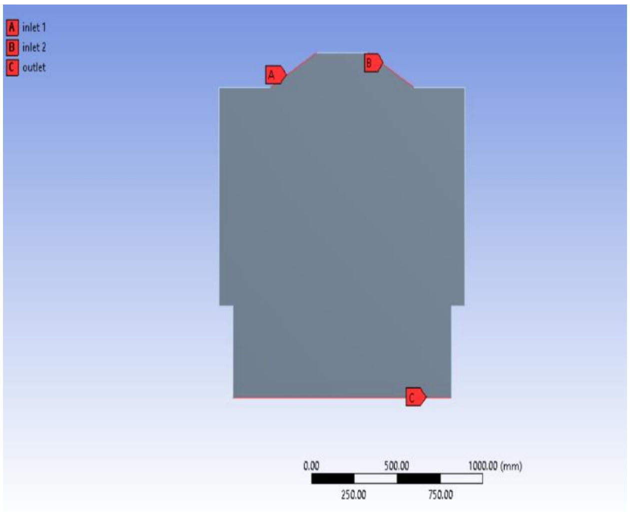


Figure 3. Boundary Conditions

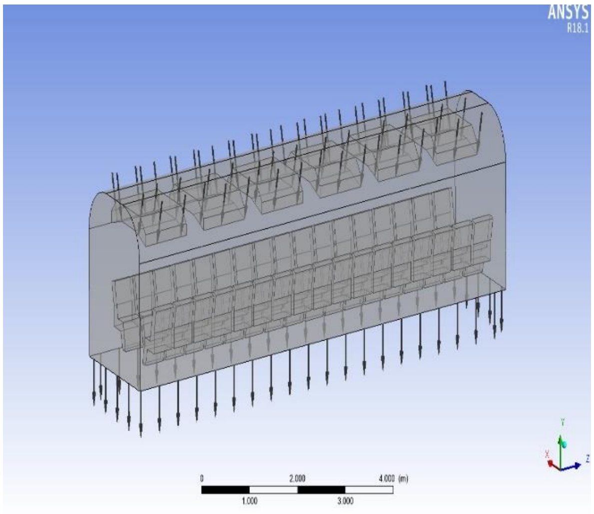


Figure 4. Model Analysis in ANSYS

III. DESIGN INPUT CONSIDERATIONS

A. Environment Conditions and Weather Data:

The comfort air temperature in summer is between 23ºC and 26ºC and the comfort air velocity is between 0.2 m/s and 0.8 m/s. These conditions should be sustained by the HVAC systems of passenger coaches. The design weather data from the ASHRAE handbooks have been used to arrive at the design criteria. The climate pattern in India suggests that the summer season is generally between March to June. During the July and February months the weather generally has temperate conditions.

Table 1. Boundary Conditions Used in Our Study

City	Inlet Air Temperature	Manikins/Passengers Body Temperature	Glass Window Temperature	Ceiling Temperature
Chennai	17 °C	33 °C (Average)	31 °C	28 °C
Delhi	17 °C	39 °C (Average)	37 °C	34 °C
Kolkata	17 °C	35 °C (Average)	33 °C	30 °C
Ideal Thermal Comfort= 23±1 °C			**Ideal Relative Humidity= 25-55%**	

The passengers are modeled in two different positions: standing and seated. For the purpose of this study, the design capacity of the train is 600 passengers, 486 standing and 114 sitting during peak time and 357 passengers, 243 standing and 114 sitting during normal time.

Table 2. Total Carrying Capacity of Designed Metro Train

Description	Driving Motor Car		Trailer Car		3 Car Train	
	Normal	**Crush**	**Normal**	**Crush**	**Normal**	**Crush**
Seated	35	35	44	44	114	114

Standing	78	156	87	174	243	486
Total	113	191	131	218	357	600

NORMAL-2 Person/sqm of standee area
CRUSH -5 Person/sqm of standee area

B. Thermal Load and Cooling Calculation

Requirement of fresh air for non-smoking= 0.35m^3 /passenger/ minute.

Compartments Quantity of ventilating air for 357 passengers during normal time and 600 passengers during crush/peak time.

For Normal Hours

0.35x357 passengers = 124.1 m^3 / minute =124.1x35.3

For Peak Hours

0.35x600 passengers= 210 m^3 / minute = 210x35.3 = 7,413 (CFM)

The following are the wattages considered for various-electrical appliances:

Fluorescent lights 2' long – 17W

VAC blower fan – 29W

- Heat transfer from equipment's and fans = 2545 $\frac{\frac{BTU}{HP}}{Hr}$
- Heat transfer from fluorescent lights = 3.4 $\frac{\frac{BTU}{Watt}}{Hr}$
- Sensible heat per passenger =205 $\frac{BTU}{Hr} = 51.6\ K\frac{Cal}{Hr}$
- Latent heat per passenger =195 $\frac{BTU}{Hr} = 49.12\ K\frac{Cal}{Hr}$
- Refrigerant used = R143a
- 1 Ton of refrigeration $= 12000\frac{BTU}{Hr} = 3024\ K\frac{Cal}{Hr}$
- 1 k-calorie $= 3.97\frac{BTU}{Hr}$
- Coefficient of heat transfers for some materials:
  - Wall end partition = 0.615
  - Roof = 0.65
  - Floor = 0.72
  - Window = 1.92

The calculation is based on all the heat source running, including electrical etc.

A.1. Dimensions of Coach:

- Length of the coach (A) = 15.2m
- Width of roof (B) = 3.245m
- Width of floor (C) = 3.04m
- Height of the coach (D) = 2.03m
- Area of side wall (A x D) = 30.856m^2
- Area of roof (A x D) = 49.324m^2
- Area of floor (A x C) = 46.208m^2
- Area of end partitions = 6.17m^2
- Height of window (E) = 0.56m

- Width of window (F) = $0.61m$
- Area of window (E x F) = $0.3416m^2$
- No. of windows per side wall = 16
- Total area of windows per side wall = $0.56 \times 0.61 \times 16 = 5.466m^2$
- Area of side wall excluding windows = $30.856 - 5.466 = 25.239m^2$

A.2 Connected Electrical Loads Inside A.C. Compartment

- Fluorescent lights 2' long = 20 Nos.
- LED screens = 15 Nos.
- Blower fan motors (0.65 HP) = 2 Nos.
- VAC fans = 8Nos

A.3 Heat Load Calculations

1. Heat Gain Due to Conduction= $\boldsymbol{A \times K \times TD \times \frac{3.97BTU}{Hr}}$

1.1. Sidewall = $50.78 \times 0.615 \times 20 \times 3.97 = 2479.64 \frac{BTU}{Hr} = 624.59\ K\frac{Cal}{Hr}$

1.2. Roof = $49.324 \times 0.65 \times 20 \times 3.97 = 2545.61 \frac{BTU}{Hr} = 641.21\ K\frac{Cal}{Hr}$

1.3. Floor = $46.208 \times 0.72 \times 20 \times 3.97 = 2641 \frac{BTU}{Hr} = 665.4\ K\frac{Cal}{Hr}$

1.4. End partition =$2 \times 6.17 \times 0.615 \times (20 - 3) \times 3.97 = 512.288 \frac{BTU}{Hr}$

1.5. Window = $5.466 \times 2 \times 1.94 \times 20 \times 3.97 = 1683.8 \frac{BTU}{Hr}$

Total = $2479.64 + 2545.61 + 2641.62 + 512.288 + 1683.8 = \mathbf{9862.954} \boldsymbol{\frac{BTU}{Hr}}$

2. Solar Heat Gain= $\boldsymbol{A \times K \times TDS \times 3.97 \frac{BTU}{Hr}}$

2.1. Sidewall =$25.39 \times 0.615 \times 9 \times 3.97 = 557.92 \frac{BTU}{Hr} = 140.53\ K\frac{Cal}{Hr}$

2.2. Roof =$49.324 \times 0.65 \times 10.55 \times 3.97 = 1342.81 \frac{BTU}{Hr} = 338.24\ \mathrm{K}\frac{\mathrm{Cal}}{\mathrm{Hr}}$

2.3. Window =$5.466 \times 5.34 \times 95.55 \times 3.97 = 11071.34 \frac{BTU}{Hr} = 2788.75\ K\frac{Cal}{Hr}$

Total =$557.92 + 1342.81 + 11071.34 = \mathbf{12972.069} \boldsymbol{\frac{BTU}{Hr}}$

3. Heat Gain due to Passengers (BTU/Hr)

S.H. = 205 x No. of passengers
L.H = 195 x No. of passengers
S.H + L.H = 400 x No. of passengers

3.1. **For Normal Hours**:

S.H + L.H= $400 \times 357 = \mathbf{142800} \boldsymbol{\frac{BTU}{Hr}} = \mathbf{36009.0764}\ \boldsymbol{K\frac{Cal}{Hr}}$

3.2. **For Peak Hours:**

S.H + L.H= $400 \times 600 = \mathbf{240000} \boldsymbol{\frac{BTU}{Hr}} = \mathbf{60519.4562}\ \boldsymbol{K\frac{Cal}{Hr}}$

4. Heat Gain due to Ventilation (BTU/Hr)

S.H.= $1.08 \times Q \times TD \times \frac{9}{5} = 1.08 \times 568.33 \times 20 \times \frac{9}{5} = 22096.67 \frac{BTU}{Hr} = 5565.91\ K\frac{Cal}{Hr}$

L.H.= $0.68 \times Q \times Gd = 0.68 \times 568.33 \times 26 = 10048.07 \frac{BTU}{Hr} = 2531\ K\frac{Cal}{Hr}$

Total = $22096.67 + 10048.07 = \mathbf{32144.7} \boldsymbol{\frac{BTU}{Hr}} = \mathbf{8096.91}\ \boldsymbol{K\frac{Cal}{Hr}}$

5. Heat Gain due to Electrical Appliances=Wattage x 3.4 BTU/Hr

5.1 Fluorescent lights $= 20 \times 1.2 \times 3.40 \times 30 = 2448\,\frac{BTU}{Hr} = 616.62\ K\frac{Cal}{Hr}$

5.2 LED screens $= 15 \times 16 \times 3.40 = 816\,\frac{BTU}{Hr} = 205.54\ K\frac{Cal}{Hr}$

5.3 Blower fan $= 0.65HP \times 2 \times 2545 = 3308.5\,\frac{BTU}{Hr} = 833.37K\,\frac{Cal}{Hr}$

5.4 VAC fan $= 29W \times 8 \times 3.4 = 788.8\,\frac{BTU}{Hr} = 198.69\ K\frac{Cal}{Hr}$

Total $= 2448 + 816 + 788.8 + 3308.5 = \mathbf{7361.3}\,\boldsymbol{\frac{BTU}{Hr}} = \mathbf{1854.22}\ \boldsymbol{K\frac{Cal}{Hr}}$

6. Total of (1+2+3+4+5)

For Normal Hours: $= 205141.023\,\frac{BTU}{Hr} = 51729.2631\ K\frac{Cal}{Hr}$

For Peak Hours: $= 302341.023\,\frac{BTU}{Hr} = 76239.6428\ K\frac{Cal}{Hr}$

7. Heat Gain due to Infiltration@10% =$\mathbf{8074.10}\,\boldsymbol{\frac{BTU}{Hr}} = \mathbf{2033.78\ K\frac{Cal}{Hr}}$

8. Gross Total Heat Gain

For Normal Hours: $= 205141.023 + 8074.10 = 213215.123\,\frac{BTU}{Hr} = \mathbf{53765.263}\ \boldsymbol{K\frac{Cal}{Hr}}$

For Peak Hours: $= 302341.023 + 8074.10 = 310415.123\,\frac{BTU}{Hr} = \mathbf{78275.643}\ \boldsymbol{K\frac{Cal}{Hr}}$

A.4. Refrigeration Capacity (TR):

For Normal Hours: $53765.263\ K\frac{Cal}{Hr} = \mathbf{17.8}\ \boldsymbol{TR}$ **needed.**

For Peak Hours: $78275.643\ K\frac{Cal}{Hr} = \mathbf{25.9\ TR}$ **needed.**

Table 3. Refrigeration Capacity

Description	Normal Hour	Peak/Crush Hour
No. of passengers	357	600
Refrigeration Capacity	**17.8 TR**	**25.9 TR**

Thus, for passengers travelling in normal state (357 passengers) - a unit of 17.8 TR and for passengers travelling in crush/peak state (600 passengers) - a unit of 25.9 TR is needed by the HVAC system to maintain a constant and comfortable temperature of 24±1 °C inside the metro train compartment during summer and winter seasons.

IV. Results

The ultimate purpose of a finite element analysis it to recreate mathematically the behavior of an actual engineering system. Analysis set up of all cases is carried out in FLUENT Solver. Results are presented in three different sections. Firstly, the air distribution within the train volume is described briefly, with a focus on the main inlets and outlets of air and pointing out the singularities of the velocity fields in its proximities. Secondly, results are discussed in terms of the temperature fields in different representative planes in the train. Finally, the results are also quantitatively analyzed in the train volume.

A. Air Distribution

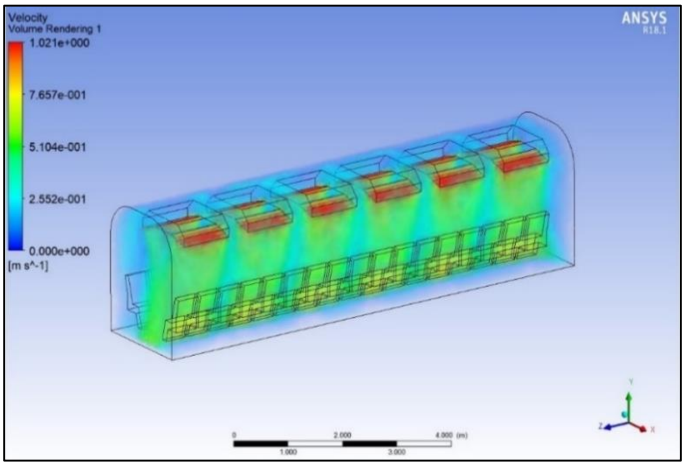


Figure 5. Air Velocity Contour Without Passengers

With Passengers:

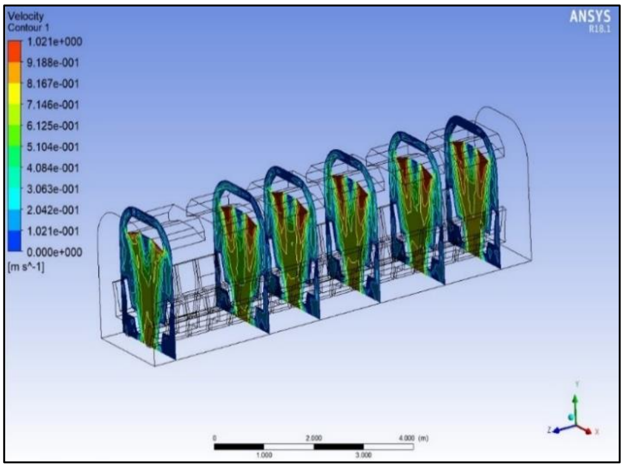


Figure 6. Air Velocity Contour (Normal hour)

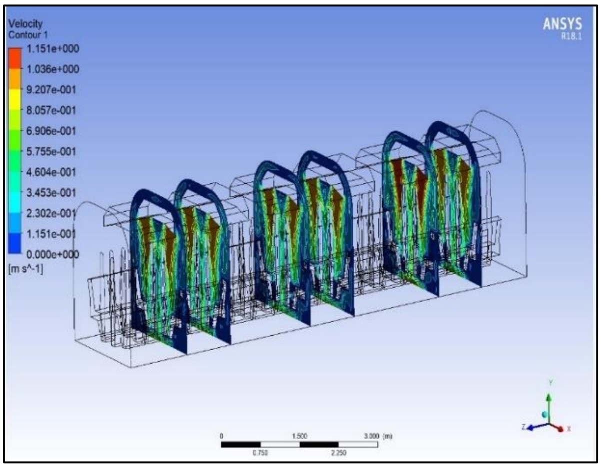


Figure 7. Air Velocity Contour (Rush hour)

The air pressure and velocity fields for Figure. 5 shows that air is driven from the supply inlets located at the ceiling to the doors' natural openings located near the floor. Even though the conditioned air is not supplied near the ceiling extractions air distribution and circulation inside the train is adequate, with no death zones without air circulation.

It is observed again that for passengers in compartment during normal and peak hour (Figure. 6 & 7), air is driven to the ceiling forced extractions, reaching maximum velocities values near them. Air is driven from the supply inlets located at the ceiling to the door's natural openings located near the floor. Again, the maximum velocities are reached in the proximities of inlets and outlets. Even though the conditioned air is not supplied near the ceiling extractions air distribution and circulation inside the train is adequate, with no death zones without air circulation.

A.2 Temperature Distribution

Case 1: Compartment Without Passengers

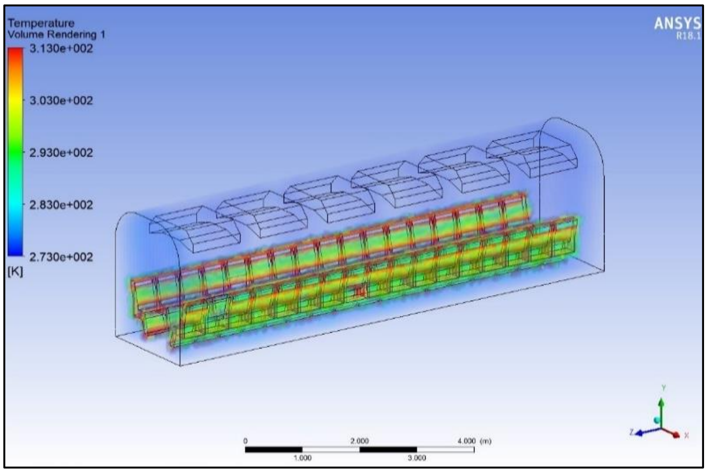
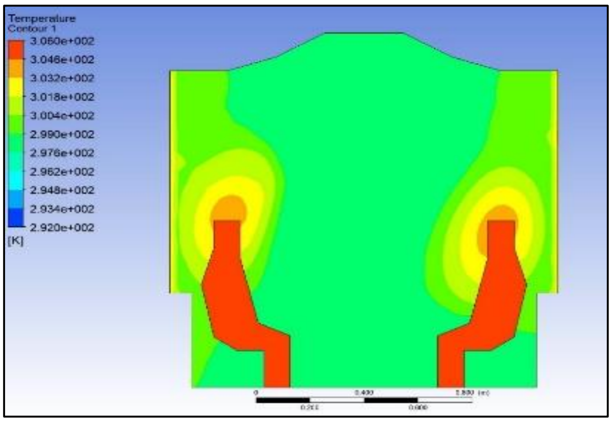


Figure 8. Temperature Distribution

Case 2: Compartment with Passengers-Normal Hour



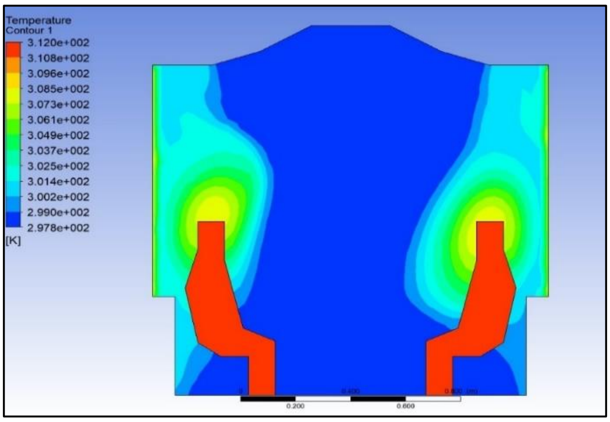


Figure 9. Temperature Distribution (Chennai) Figure 10. Temperature Distribution (Delhi)

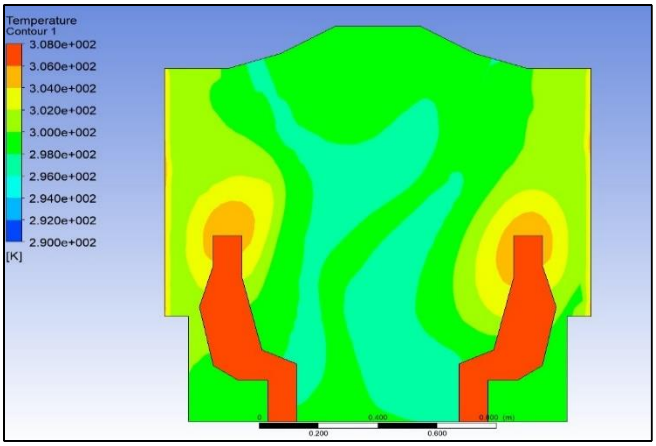
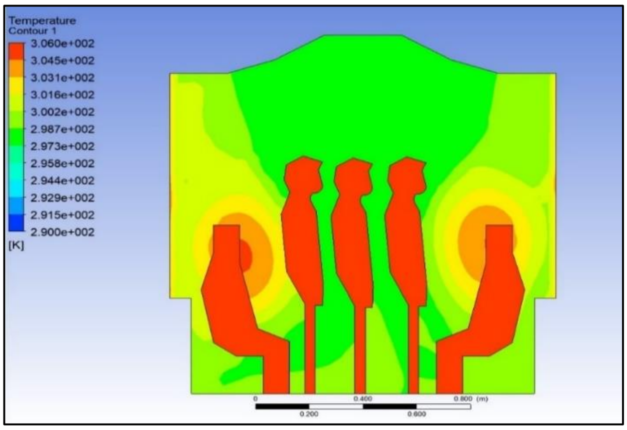


Figure 11. Temperature Distribution (Kolkata)

Case 3: Compartment with Passengers-Peak Hour



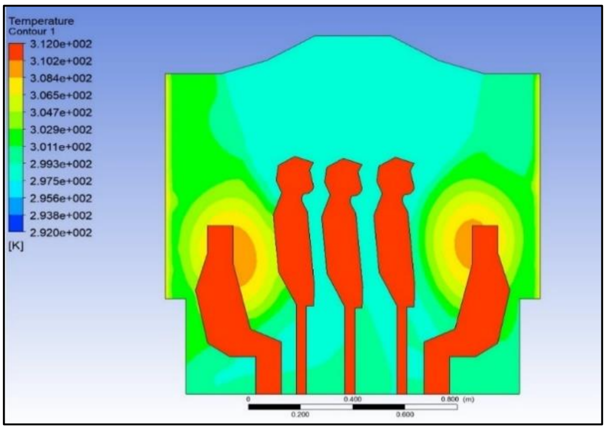


Figure 12. Temperature Distribution(Chennai) Figure 13. Temperature Distribution(Delhi)

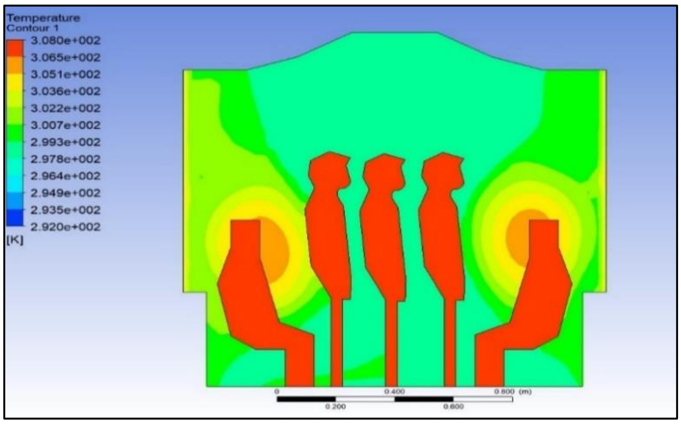


Figure 14. Temperature Distribution (Kolkata)

The quantitative results in the defined comfort volume for CASE 1 show a good behavior in terms of the defined comfort parameters for temperature and air velocity removal rate since there is no passengers to dissipate heat energy. In CASE 2, heat dissipation from passengers/manikins during normal hours, it can be seen that there is a good flow for the air between them indicated by meter scale where we can see continuous flow of cool air carrying away the heat from them resulting in more ride comfort to the passengers/manikins inside the metro train compartment. However, there is a slight variation in heat removal rate in different cities due to their varying climate zones. In CASE 3, the manikins/passengers seated in seats have almost the same temperature difference compared to the non-symmetrical case. However, in case of passengers in standing position due to rush hours, the velocity of air was somewhat low between manikins/passengers in standing position shown by meter scale indicating low heat dissipation by the HVAC system between them mostly in Delhi and Kolkata due to their higher outdoor temperature.

V. Conclusion

A CFD analysis of air flow and temperature distribution in a metro rail vehicle equipped with a specific HVAC system has been presented for nine different scenarios with different load conditions and cities with convective boundary condition considered on the glass surfaces and outer surfaces of the cabin. The results showed that due to more people in the compartment during peak/crush hours, the airflow entered from the HVAC system mostly inclines on the left side of the compartment and this causes the airflow to deviate from the thermal comfort conditions in some parts of the compartment and makes the seated/standing passengers uncomfortable.

The simulation result of air distribution for various human load (with and without passengers) computed velocity value changes between 3.14 m/s and 0.65 m/s at the inlet and a value of 1.34 m/s to 0.18 m/s at the outlet vent. There was only a slight variation in air velocity (±0.5m/s) when compared between cities like Chennai, Delhi and Kolkata. The simulation result of temperature distribution for the human load in CASE 1 showed a good behavior in terms of the defined comfort parameters (23±1ºC) for temperature and air velocity removal rate since there is no passengers to dissipate heat energy. However, temperature distribution for the human load in CASE 2 with 357 passengers (114 seated and 243 standing) during normal hour indicated by meter scale shows about 23.2 ˚C while the near foot was around 24 ˚C in all the three cities, predicting a comfortable environment around the passenger as per comfort conditions. So, to maintain the current heat removal rate the VRF system have to maintain 17.8 TR. In CASE 3 the air temperature around all the passengers during peak hour was around 23.5 ˚C while near the foot was about 26-27 ˚C which was not in the range of comfort temperature.so to overcome this discomfort the VRF system have to maintain 25.9 TR so as to dissipate the extra heat from crowded passengers. The overall results thus obtained, analyzed and explained in terms of temperature and air velocity, showed an adequate behavior from the passengers' comfort point of view.

The main contribution of this work to the state of the art is the detailed CFD analysis of a representative railway vehicle for different load conditions and different cities, to the best of the knowledge of the authors there are no previous studies specifically related to air flow and temperature distribution in metro rail vehicles with passengers during normal hour and peak hour. The present work introduces this analysis and establishes a methodology and starting point for future studies. It is expected that HVAC systems designers and integrators will increasingly require CFD analysis of the final systems (railway vehicles and other means of public transport) in order to finely adjust the system configuration and provide more satisfactory comfort levels to final users. It is, therefore, necessary to establish an appropriate methodology for such CFD analysis. The design of the air flow and temperature distribution system in the railway coach will also require CFD analysis of the air ducts for ensuring better performance (lower pressure drop and better flow distribution).

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