ARCHITECTURE CIVIL ENGINEERING ENVIRONMENT
The Silesian University of Technology No. 4/2022

doi: 10.2478/ACEE-2022-0044

CFD MODELLING OF THERMAL COMFORT IN THE PASSENGER COACH

Agnieszka PALMOWSKA [a*], Izabela SARNA [b]

[a] PhD; Faculty of Energy and Environmental Engineering, The Silesian University of Technology, Konarskiego 20, 44-100 Gliwice, Poland
*Corresponding author. E-mail address: *agnieszka.palmowska@polsl.pl*

[b] MSc Eng.; Faculty of Energy and Environmental Engineering, The Silesian University of Technology, Konarskiego 20, 44–100 Gliwice, Poland

Received: 27.07.2022; Revised: 12.10.2022; Accepted: 14.10.2022

Abstract
This paper presents the results of numerical simulations of thermal comfort in a passenger coach. The numerical model with people's presence was developed and appropriate boundary conditions were prepared. The ANSYS CFX program was used for the simulations. The calculations were carried out for summer and winter conditions. The predicted mean vote (PMV), predicted percentage dissatisfied (PPD) and draft rate (DR) were calculated to assess the thermal comfort of passengers. The requirements of railway standards in terms of passenger comfort assessment were also verified. Based on the simulation results, it was found that the thermal comfort conditions of the passengers in the coach were not fully satisfactory, especially in summer.

Keywords: **Ventilation; Thermal comfort; CFD; Thermal and humidity conditions; HVAC.**

Nomenclature:

t_i – interior temperature, °C

t_{mi} – mean interior temperature, °C

t_e – exterior temperature, °C

t_s – sol-air temperature, °C

$\dot{m}_S$ – mass flow rate of air supply, kg/s

U – heat transfer coefficient, $\frac{W}{m^2 \cdot K}$

RH – relative humidity, %

w – air speed, m/s

w_{max} – maximum air speed, m/s

w_{min} – minimum air speed, m/s

$\dot{q}_{S,C}$ – convection heat flux, $\frac{W}{m^2}$

$\dot{q}_{S,R}$ – radiation heat flux, $\frac{W}{m^2}$

$\dot{W}$ – moisture stream, kg/s

1. INTRODUCTION

The study of thermal comfort is the subject of many scientific studies. They concern residential, public and office buildings [1–5], as well as vehicles due to the increased attention of researchers and vehicle manufacturers on improving thermal comfort conditions in passenger compartments in the last decades [6]. However, comfort research in transport is rather rarely conducted, although comfort study and proper analysis of the results could affect the comfort of people's travel. Thermal comfort for vehicle users must be maintained for a good mental and physical condition of the users as well as air quality due to daily exposure to pollutants. The driver's stress because of low comfort conditions and less visibility caused by the phenomenon of fogging on the windshield influences the safety of the trip [6]. Appropriate design of Heating, Ventilation, and Air-Conditioning (HVAC) systems in vehicles is very important because passengers are often forced to stay in a designated place for several hours. Travellers often complain about the operation of HVAC systems in vehicles, despite the design following current standards. However, the method of their operation also causes passenger dissatisfaction with comfort conditions.

Thermal comfort is a state of thermal balance of the human body with the environment. It is the result of a balance between the amount of heat produced by the body and the heat losses released to the environment. The currently used indicators to assess thermal comfort are: predicted mean vote (PMV) on the 7-point thermal sensation scale: +3 hot; +2 warm; +1 slightly warm; 0 - neutral; -1 slightly cool; -2 cool; -3 cold and predicted percentage dissatisfied (PPD) were proposed by P. O. Fanger [7]. The second group of indicators are indicators characterizing local discomfort: draft discomfort is measured by the draft rate (DR) index, which measures the percentage of people sensitive to faster air movement, and the percentage of dissatisfied (PD) index, which measures the percentage of people dissatisfied with the variability of air temperature between the head and ankles (vertical air temperature difference) or radiation asymmetry of partitions (it takes place when there are partitions in the room with a temperature different from the air temperature). The methodology of determining and interpreting thermal comfort with the use of the calculation of PMV and PPD indices and local thermal comfort criteria was described in the standards [8, 9].

In the HVAC industry, there are no clear guidelines regarding air distribution in coaches, apart from the standards [10] and the UIC leaflet [11]. The most commonly used solutions are mixing and displacement ventilation. In the case of mixed ventilation, the diffusers are located above the windows, while in the case of displacement ventilation, the air is supplied through the diffusers located in the vehicle's ceiling or on the floor [12, 13]. Air conditioning units are usually located so that the heavy compressor, condenser, pumps, electric heater, and fan are under the casing, while the supply fan and evaporator are located in the false ceiling. In some carriages, all devices are located on the roof. Depending on the solution, the outside air can be taken in from the train roof or from below. Often the same channels are used to cool and heat the coach. For additional heat recovery, recirculation is also used [14].

The thermal comfort in the passenger compartment was examined in the study Aliahmadipour et. al. [15]. The calculations were conducted for three cases: seated and sleeping manikins or without passengers. The comfort conditions were assessed based on simulated air temperature and velocity, which were validated with the conducted measurements. It was found that the non-symmetrical airflow entering from the HVAC system caused discomfort in a part of the compartment both for seated and sleeping passengers. Due to the unsatisfactory results obtained, simple improvements to the HVAC installation (addition of a direct channel to the compartment, modification of upper inlet) were carried out to improve the thermal comfort conditions in the compartment. In the study, Goelz and Orellano performed [16] simulations of transient thermal comfort in a metro coach. The impact of a door opening on comfort conditions was investigated. The results of the calculation showed that during the opening of the door warm outdoor airflow was entering the upper parts of the coach and cool conditioned air was leaving the vehicle at floor level. After that transient phenomenon, an increase in temperature in the coach was observed (especially in the upper part), which resulted in a deterioration of comfort. The authors also mentioned other transient behaviour: pre-cooling and preheating, however, they were not the subject of research. Furthermore, the methods with respect to speed of unsteady simulation should be improved to use on an industrial basis. The analysis of comfort in a subway coach was also the subject of study by an Indian research team Karthik et. al. [17]. The comfort, with passengers during normal hours and peak hours, was assessed on the basis of air flow and temperature distributions simulated in the ANSYS Fluent program. In the final conclusions, the authors will emphasize the validity of CFD calculations at the final stages of designing HVAC devices in all rail vehicles. Schmeling and Bosbach [18] investigated the thermal comfort and ventilation efficiency in the lower-level of a double-deck high-speed train with displacement ventilation. The authors focused on examining the amount of heat released by humans and checking how it affects the conditions of comfort in accordance with standard EN 13129 [10]. The thermal manikins with set variable sensible heat release were used to verify the simulation results. The conducted experimental studies proved the advantages of displacement ventilation in the train compartment. First, the airflow induced by thermal convection arises only near the heat source. Secondly, there is a low risk of drafts due to low air velocities. The temperature was also found to depend on the latent and sensible heat released by passengers. The temperature stratification increases with increasing heating power, but it does not significantly affect the speed distribution. In the next article Meyenberg et. al [19] examined the possibility of using a moving thermal manikin for comfort tests in aircraft or trains. The moving manikin was supposed to represent walking passengers, who can cause a blockage of airflow. The experimental studies were conducted for the same coach and ventilation models as in the study [18]. The results showed that moving manikin causes a temperature decrease and velocity increase at the shoulder level of seated passengers and velocity increase at head level. However, the disturbances to passenger comfort were present for a short time. The comparison of various, simulated air distribution systems: displacement, personalized and mixing ventilation were presented in the study [20] by Zhang et. al. The simulations were carried out for sleeping spaces in the transport vehicle. The research aimed to determine the thermal environment and air quality for these cases. The results of simulations showed that the personalized system provided the best values of air velocity with little draft risk. The research on the efficiency of the ventilation system was also carried out for the driver's cabin of a modern rail vehicle. The evaluation of thermal comfort in high speed trains was presented by Yang et. al. [21] and Zhang and Lu [13]. In study [13], the authors compared the traditional sidewall air supply and bottom return mode with the underfloor air supply mode in terms of ensuring comfort and verification of air parameters: airflow, temperature, CO_2 concentration and humidity field. The better thermal conditions were obtained for underfloor supply mode according to CFD calculation. In the study [22], Palmowska and Walczyk analysed indoor thermal and humidity conditions for six different cases of air distribution. Thermal comfort in the kitchen environment of a non-air-conditioned railway pantry coach in Indian Railways was the subject research in the study Alam and Salve [23]. The evaluation of thermal comfort in a double-decker train cabin for summer and winter conditions was conducted in the study of Konstantinov and Wagner [24]. To access thermal comfort FIALA-Manikin model was used, which made it possible to predict the human thermoregulatory responses to indoor and personal conditions. The obtained velocity and temperature distributions in the cabin were inhomogeneous, the better thermal comfort conditions were in the lower deck. Summing up, most of the studies [13, 16–20, 24], were done to analyze comfort in vehicles with displacement ventilation. Fewer studies on the operation of mixing ventilation were conducted [12, 20]. In only two presented studies [13, 21], Fanger's indices were used to assess comfort in the passenger coach. The research [13] was conducted in relation to China's climate and the study [21] was concerned only with summer conditions. Usually, the assessment of comfort is based only on the analysis of the air speed and temperature distribution and a possible reference to whether they meet the requirements of the standard EN 13129 [10]. The assessment of comfort according to the Fanger indices for the deluxe cabin of a high speed train was made in work by Ghosh et. al. [25], however, using the program Autodesk Ecotect [26], not the CFD technique.

This study is a continuation of the scientific research presented in the previous article by Sarna and Palmowska [12]. The purpose of the research was the analysis of thermal comfort in the passenger coach with mixing ventilation for winter and summer conditions while providing real data on the parameters of the internal and external environment. The evaluation of thermal comfort was based on Fanger indicators and railway standards. The implementation of this goal required supplementing with human models, adapting the existing numerical model of the coach in the Ansys CFX code and fine-tuning the appropriate boundary conditions.

2. REQUIREMENTS FOR THERMAL COMFORT IN PASSENGER COACH

In the previous study, Sarna and Palmowska [12] described current and general requirements of ventilation and air conditioning systems in passenger coaches. This paper focused on the requirements of thermal comfort in such vehicles.

The comfort parameters defined in the standard EN 13129 [10] and UIC leaflet 553 [11] include air temperature, surface temperatures, air speed, and relative humidity of air. The air temperature parameter encompasses the mean indoor temperature and criteria for a horizontal and vertical range of the extreme interior air temperature to reduce areas of local thermal discomfort to a minimum. Due to providing thermal comfort conditions in passenger coach a mean interior temperature should be at least 21°C during the heating period and the maximum mean interior temperature should not exceed 27°C for I and II climatic zones or 25°C for III climatic zone during the cooling period. In the railway standards [10, 11], climate zones (separately for winter and summer) are assigned to each country in which the [10, 11] vehicle is homologated. Based on climate zones, the selection of HVAC devices for rail vehicles is carried out.

Table 1.
Requirements of temperature parameters in the comfort zones [10]

Parameters	Quality limit 1	Quality limit 2
Range of mean interior temperature concerning the interior temperature setting	1 K	1.5 K
Horizontal range of the extreme interior air temperatures	2 K	2.8 K
Vertical range of the extreme interior air temperatures for seated passengers	3 K	5 K if the foot in the warmest point 3.5 K otherwise cases

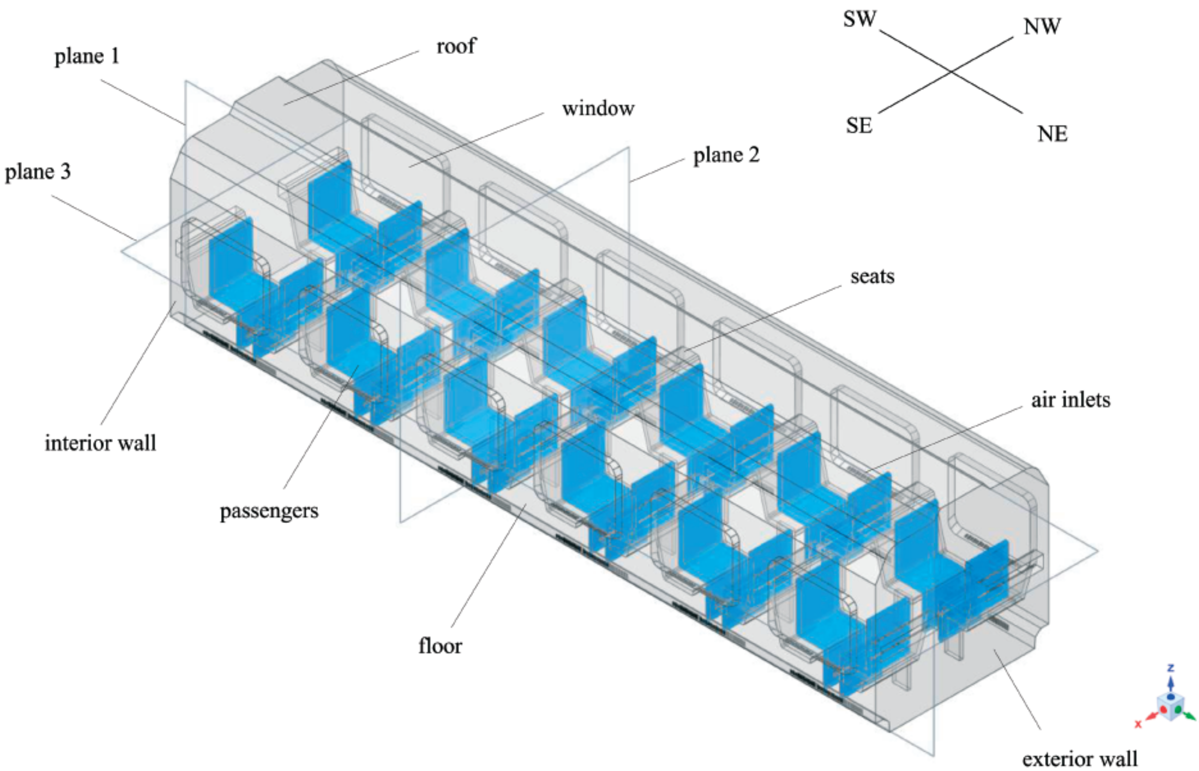

Figure 1.
The numerical passenger coach model – geometry

According to the Polish standard, it is located in the 2nd climate zone both in winter and summer. Detailed interior temperature settings are defined by a regulation curve as a function of mean exterior temperature [27]. Horizontal temperature distribution means an absolute difference of the exterior air temperature measured at 1.1 m from the floor, while vertical is an absolute difference of the exterior air temperature in a vertical direction at different heights. For standing passengers, it is the difference between the extreme interior air temperature at 0.1 m and 1.7 m above the floor. The interior temperature is measured in a few locations specified in accordance with the standard [10]. For seated passengers, the differences in the extreme values should be measured at the position of the head, shoulders, knees and feet defined in the standard [10]. Different requirements of surface temperatures were specified. The requirements were defined so that they not only meet the subjective feelings of passengers but also are achievable in practice. Table 1 presents the main requirements of temperature in comfort zones. Two quality limits were shown: 1) target quality limit that if fulfilled results in 100% fulfillment and 2) quality limit required to be fulfilled.

To reduce the risk of draught, acceptable values of air speed were determined based on limiting curves as a function of local air temperature. Maximum air speed for t_i = 22°C is w_{max}=0.25 m/s and for t_i = 27°C is w_{max}=0.60 m/s.

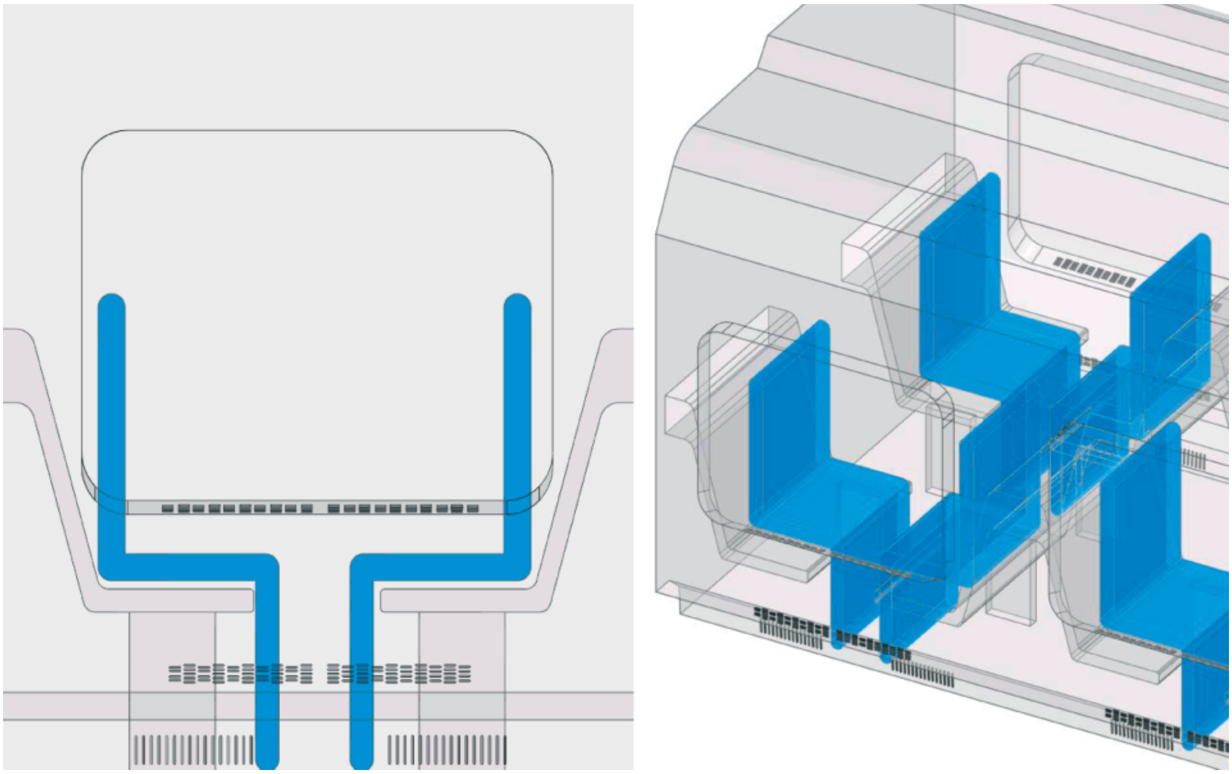

Figure 2.
The human model: side view (left) and axonometric view (right)

To ensure adequate humidification in passenger coaches the range of humidity of the air is specified in the form of graphs. Absolute and relative humidity should not exceed 11.5 g/kg and 65% irrespective of the interior air temperature of comfort zones [10, 11].

The standard requirements of the minimum amount of fresh air and heat transfer coefficient for the coach at standstill were defined. It was described in the previous article [12].

3. METHODS

3.1. Description of the numerical passenger coach model

The numerical passenger coach model (Fig. 1) was an improvement of the model prepared in the previous study [12]. The exterior geometry of the model was not changed, thus the overall dimensions of the coach were 10.6×2.6×2.5 m, and the construction of the model was symmetrical. The investigated model was a part of the coach, which consisted of a restaurant part, an open compartment and sanitary facilities. The walls of the compartment were exterior except for the partition wall. There were six identical, double-glazed windows on longitudinal walls. To research thermal comfort, human models were introduced. There were 48 passengers in this coach. Due to the limited computing power of the computer and the size of the model people were modelled in a simplified manner using a simple human body model. Two people sitting side-by-side were created as one model (Fig. 2).

3.2. Grid data

An unstructured grid was used to mesh the numerical model of the passenger coach in all cases. The total number of elements was 2,902,873, including 10,220,446 tetrahedral elements. The basic mesh size was in the range of 15–50 mm, with local refinement around inlets, outlets and human models, the maximum edge length was 2 mm. The inflated boundary was built with five layers consisting of prismatic elements with a maximum thickness of 200 mm. It was used for all elements of the numerical model, except diffusers. The mesh has average orthogonal quality and skewness of 0.79 and 0.37, respectively. Before performing the CFD simulations, a grid independence study was conducted.

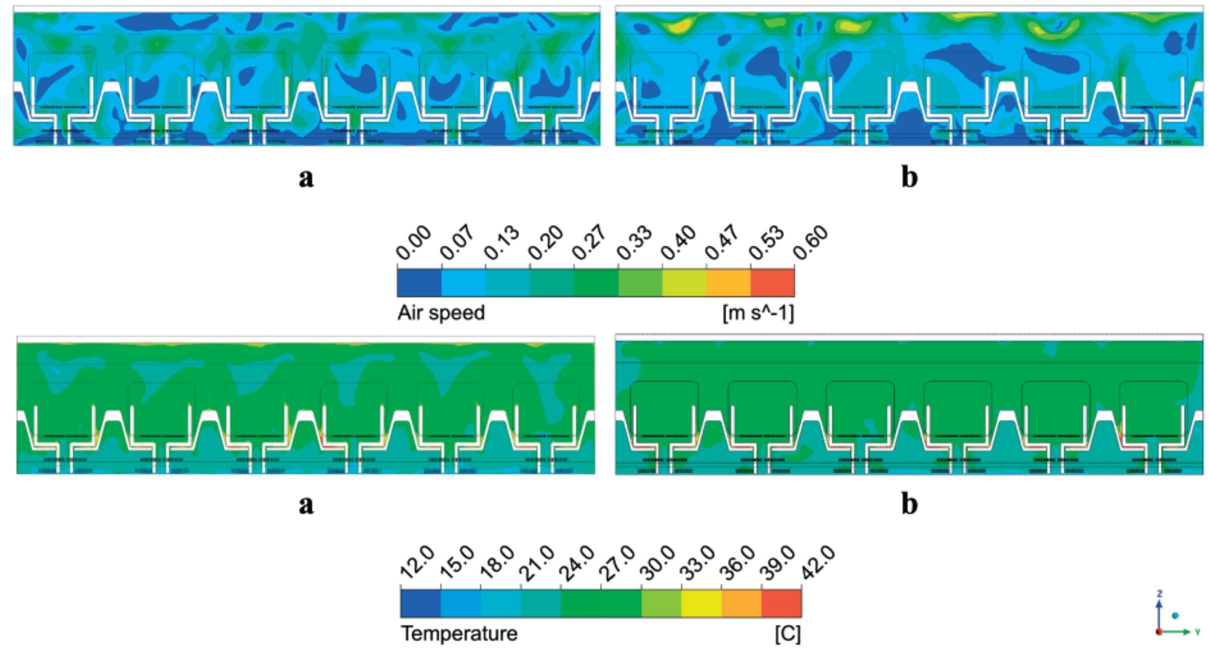

Figure 3.
The comparison of the distribution of air parameters: speed, temperature, in the vertical plane YZ, X=2.55 m (Plane 1), a) summer, b) winter

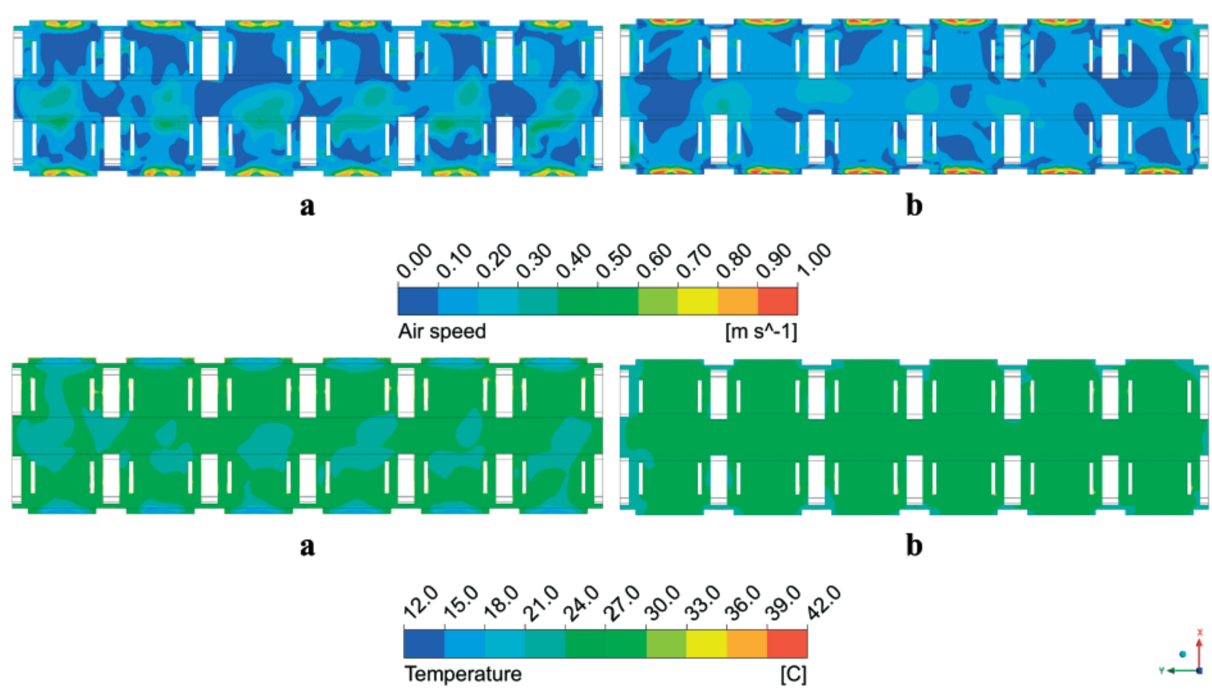

Figure 4.
The comparison of the distribution of air parameters: speed, temperature, in the horizontal plane YX, Z=-0.85 (Plane 3), a) summer, b) winter

3.3. Boundary conditions and design assumptions

The calculations were carried out for two cases: 1) summer conditions and 2) winter conditions. Boundary conditions were assumed based on Polish standard PN-EN 13129 [10] and the UIC leaflet 553 [11] for II climatic zone, to which Poland is assigned. The design conditions in a passenger coach were as follows: t_i = 27°C, RH = 45%, t_e = 35°C in summer; t_i = 22°C, RH = 95%, t_e = -20°C in winter. As in the previous study [12], mixing ventilation was modelled to ensure comfort. At each window, there were two lower air inlets located at leg level, and two window air inlets responsible for the distribution of air in the coach. The cooling airflow was distributed to the diffusers through a system of ducts located on the floor. Then, 20% of the mass flow rate flowed into the coach through the lower diffusers, the remaining part of the stream mixed with the air sucked in from the coach was supplied by the window diffusers. The amount of the air-conditioning air stream was determined for one real HVAC unit, used for a passenger coach of similar size and purpose, which equalled 1815 m^3/h.

The temperature of the supplied air was determined based on the heat and humidity balance, including human heat and moisture gains, heat transfer losses, and solar gains. The balance does not take into account the accumulation of heat in partitions and heat gains from lighting, due to the small impact on the value of total heat gains. For the summer it was 11.65°C by lower diffusers and 12.60°C by upper diffusers, for winter 26.91°C and 26.61°C, respectively. The solar gains for windows were assumed as the equivalent solar load of 700 W/m^2 with a solar factor of 60% and converted from surface gains into solar-air temperature in accordance with railway standards [10, 11]. Due to the lack of information on the solar gains for opaque partitions in standards [10, 11], t_s was determined on design calculations. In winter the solar gains were not taken into account, thus only the exterior temperature was assumed. The value of the heat transfer coefficient was the same for each partition, equalling 1.6 W/(m^{2}K) [10, 11]. Human heat gains were introduced as convection and radiation heat fluxes and a moisture stream was also established in accordance with [28]. It was as follows:

$\dot{q}_{S,C} = 27.32\ \frac{\mathrm{W}}{\mathrm{m}^2}$, $\dot{q}_{S,R} = 27.2\ \frac{\mathrm{W}}{\mathrm{m}^2}$, $\dot{W} = 0.00107\ \frac{\mathrm{W}}{\mathrm{m}^2}$ for summer, and $\dot{q}_{S,C} = 35.55\ \frac{\mathrm{W}}{\mathrm{m}^2}$, $\dot{q}_{S,R} = 35.34\ \frac{\mathrm{W}}{\mathrm{m}^2}$, $\dot{W}$=0.00072 kg/s for winter.

3.4. Numerical method

The numerical simulations were carried out using the ANSYS CFX software. The Reynolds - averaged Navier-Stokes were solved by the Finite Volume Method. The CFD model used the Shear Stress Transport (SST) turbulence model, which is a combination of k-ε and k-ω models [29]. The thermal radiation in the passenger coach was simulated using the Discrete Transfer Model (DTM). Non-slip conditions in near-wall boundary layer were taken into account. The simulations were performed in steady-state, three-dimensional and non-isothermal conditions. The monitoring points of air parameters were introduced to control the convergence of calculation results. The required stabilization of observed variables was achieved after about 2000 iterations. At the same time, the convergence for heat transfer and mass fraction was about 1.0E-3 and for other variables was less than 1.0E-4.

4. RESULTS AND DISCUSSION

The results of simulations were prepared in a graphical form and parameter values using the ANSYS CFX-Post module. Three planes were selected to present the distribution of air parameters in terms of initial overall ensuring thermal comfort conditions for passengers. The vertical YZ plane 1 was along the passenger seats: X=2.55 m (Fig. 1), and the vertical ZX plane 2 was passed through the body of the passengers: Y=-11.85 (Fig.1). As in the previous study [12] the range of air parameters in the comfort zone was between 0.1-1.1 m of height for seated passengers. The horizontal YX plane 3 was located at a height of 1.1 m above the floor: Z=-0.85 (Fig. 1). This is a representative height to access thermal comfort in accordance with railway standards [10,11] and relates to the height of the passengers' heads. The comparison of the distribution of air parameters: speed, temperature in the vertical and horizontal plane was shown in Fig. 3 and 4.

In the comfort zone for seated passengers, the air speed did not exceed the maximum value of 0.6 m/s (for t_{im}=27°C) that is allowed in the summer period on any of the discussed vertical planes 1, 2. Maximum value was 0.51, 0.45 m/s, respectively. For winter, areas, where the air speed was higher than the recommended value (w_{max}=0.25 m/s for t_{im}=22°C) were observed, however, these were areas above the seated passengers and did not affect their comfort. The exact locations of the higher speeds were shown in Fig. 5. The occurrence of airflow dead zones was also

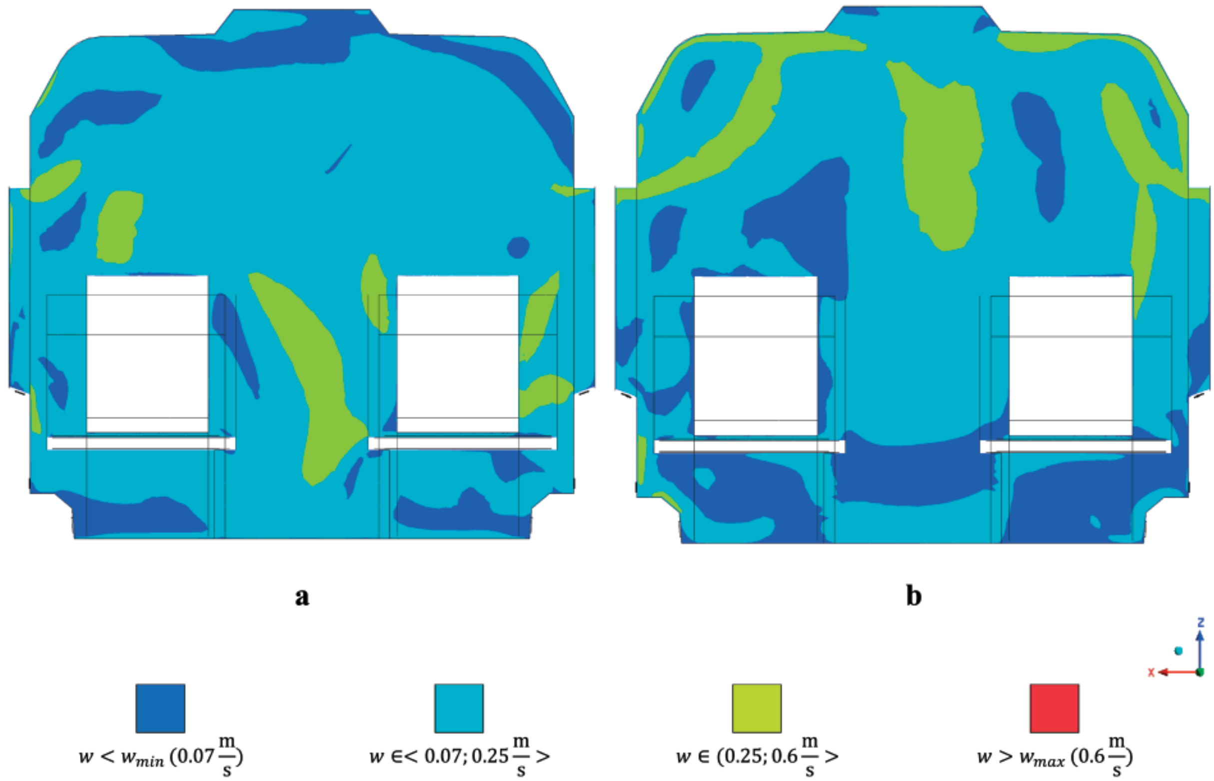

Figure 5.
The comparison of comfort zone in the vertical plane ZX, Y=-11.85 m (Plane 2), a) summer, b) winter

Table 2.
Results of air parameters and thermal comfort indicators at monitoring points 1-12

Monitoring point	Summer					Winter				
	metabolic rate = 1 met clothing insulation = 0.5 clo					metabolic rate = 1 met clothing insulation = 1.25 clo				
	t_i	RH	PMV	PPD	DR	t_i	RH	PMV	PPD	DR
	°C	%		%	%	°C	%		%	%
1	24.7	54.7	1.8	66.9	8.3	23.5	42.1	-0.2	5.8	8.4
2	24.0	55.8	1.3	41.4	7.6	24.4	40.6	0.2	6.1	7.7
3	24.5	56.6	1.1	30.1	7.6	24.3	40.4	0.2	6.0	10.6
4	24.6	55.0	0.8	17.8	6.2	23.8	42.3	0.0	5.0	8.7
5	25.0	54.2	2.2	83.4	6.4	23.8	43.6	0.1	5.1	4.4
6	24.5	55.1	1.8	65.7	4.7	24.7	39.5	0.4	7.7	7.5
7	22.9	59.1	0.0	5.0	16.7	24.6	39.2	0.4	7.6	8.8
8	26.3	51.6	0.8	17.6	8.9	24.2	41.1	0.2	5.7	3.5
9	24.8	54.1	1.8	68.3	7.8	23.8	41.5	0.1	5.2	5.1
10	23.6	57.0	1.0	24.5	11.3	25.0	38.9	0.4	8.0	7.7
11	23.0	58.3	0.2	5.7	13.2	24.7	38.9	0.4	8.1	9.4
12	24.3	54.6	0.7	14.2	4.8	23.9	41.0	0.2	5.7	5.0
Mean	**24.4**	**55.5**	**1.1**	**36.7**	**8.6**	**24.2**	**40.8**	**0.2**	**6.3**	**7.2**

observed, mainly around the passengers' ankles. This is somewhat advantageous because passengers feel the cooling airflow from the diffusers to a lesser extent.

Although the temperature in the coach was largely within the acceptable range from 21°C to 27°C, passengers could experience discomfort in the ankle area due to the occurrence of temperature variations and its uneven distribution (Fig. 3). Both the summer and winter periods, the determined value of a global vertical range of the extreme interior air tempera-

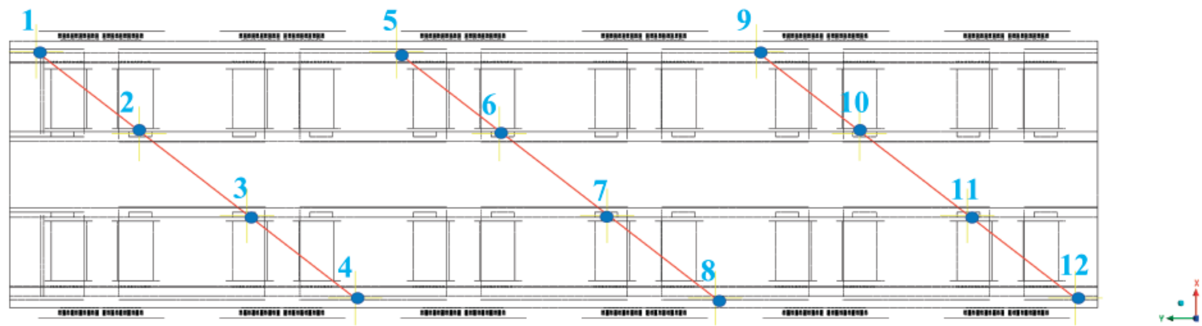

Figure 6.
Location of monitoring points used to determine air parameters and thermal comfort indicators in the coach

tures did not meet the requirements set out in Tab. 1 and amounted to 5.3 K and 5.0 K.

Values of t_{mi} in the comfort zone were determined according to EN 13129 [10] as the arithmetic mean of monitoring points located 1.1 m above the floor (Fig. 6). The coach was divided into 3 equal parts with 4 monitoring points located on marked diagonals. Table 2 presents the results of air parameters and thermal comfort indicators in the monitoring points.

In summer mean value of t_{mi} =24.4°C was too low because in accordance with Tab. 1 it should be 27°C±1 K, in turn, in winter the mean value of t_{mi} =24.2°C was too high in relation to the assumptions. It was noticed that the value of t_{mi} for both periods differed only by 0.2°C. In both summer and winter, the relative humidity meets the requirements of the EN 13129-1 standard (the maximum value of humidity at a temperature of 22°C is 65% and for 27°C is 54%).

The indoor environment is considered comfortable if the PMV value is between -0.5 and +0.5. This requirement was met at all monitoring points only in winter. For summer, higher values were obtained, the mean of which was PMV = 1.1. Thus passengers may feel warm during this period, however, this does not apply to all points. The highest PMV value of 2.2 was obtained in point 5, and PMV = 1.8 was obtained in three places: point 1, point 6 and point 9. This is due to the influence of insolation on this side of the coach, which affects the high radiation temperature of up to 39°C. For these PMV values, the percentage of dissatisfied people exceeds 50%. In winter, the PPD value did not exceed 8%. In both summer and winter, the percentage of people sensitive to faster air movement was not high. DR index oscillated around the mean value equal to 8.6% and 7.2%, for summer and winter, respectively.

It was also decided to compare the set temperature (27°C for summer and 22°C for winter) with the average air temperature (t_i) in the entire coach. Calculations of the heat balance were made to obtain the average internal temperature of the entire zone, not only at selected points located at one height (1.1 m) specified in standards [10, 11]. As shown in Fig. 3, the temperature stratified in the coach also influences the mean value of the internal temperature. In winter, temperature equalled t_i = 23.8°C which is slightly lower than t_{mi}, however, in summer, it was even lower: t_i = 22.1°C.

At a height of 1.1 m (Plane 3), the mean value of indoor temperature was 25.7°C (in summer) and 24.7°C (in winter). More uniform temperature distribution in the range of 24–27°C was obtained in winter. In the summer period, areas of cooler air coming in the range of 18–24°C from the diffusers were noticed. In winter, there was also heat loss from the exit walls, which reduced the comfort of passengers sitting near the exit. The horizontal temperature gradient was 1.5 K, thus it did not exceed the permissible value of 2 K (Tab. 1). However, in summer, due to the less homogeneous temperature distribution, the horizontal gradient slightly exceeded the permissible value and was 3.4 K.

Additionally, according to [8], monitoring points were introduced into the coach at the height of the feet, abdomen and head of passengers to assess their thermal comfort. These points were located on the axes in the center of the seat space between passengers at heights of 0.1, 0.6 and 1.1. m, which gave a total of 12 measuring axes for the entire coach (Fig. 7). A total of 36 monitoring points on the axes are numbered as follows: on axis 1 there are points 1, 2, 3 at heights of 1.1 m, 0.6 m and 0.1 m respectively; on axis 2 there are points 4, 5, 6 at heights of 1.1 m, 0.6 m and 0.1 m respectively; and so on up to as axis

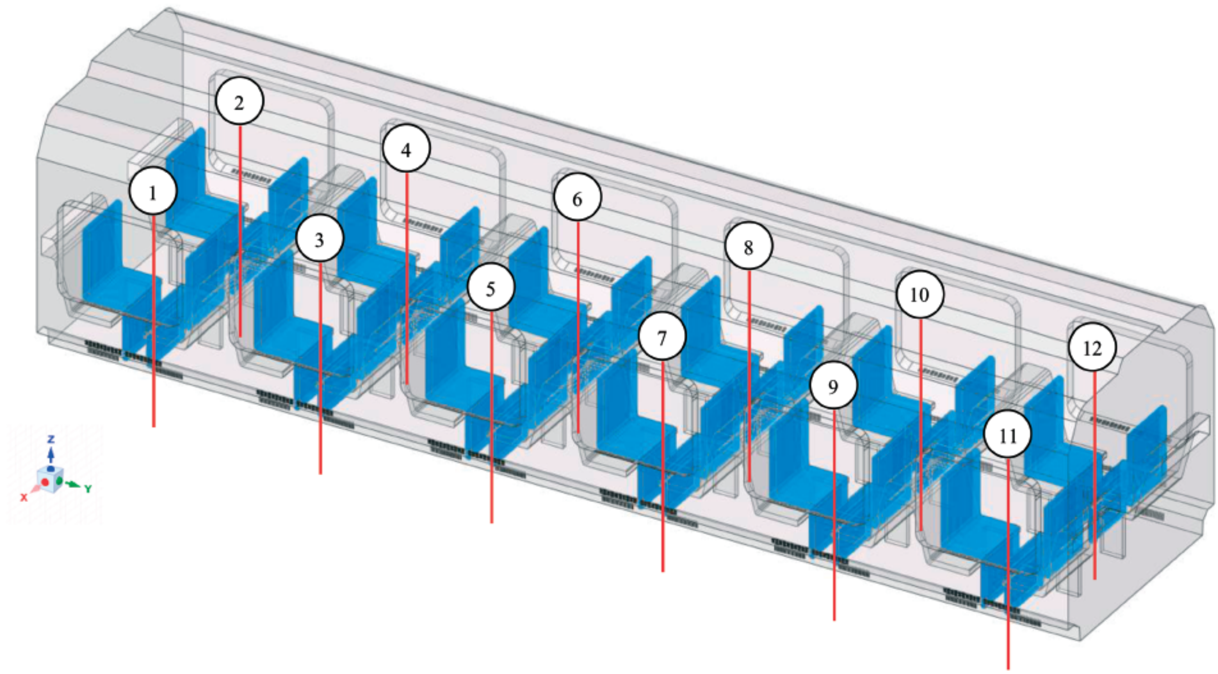

Figure 7.
Location of the monitoring axes used to determine thermal comfort for seated passengers

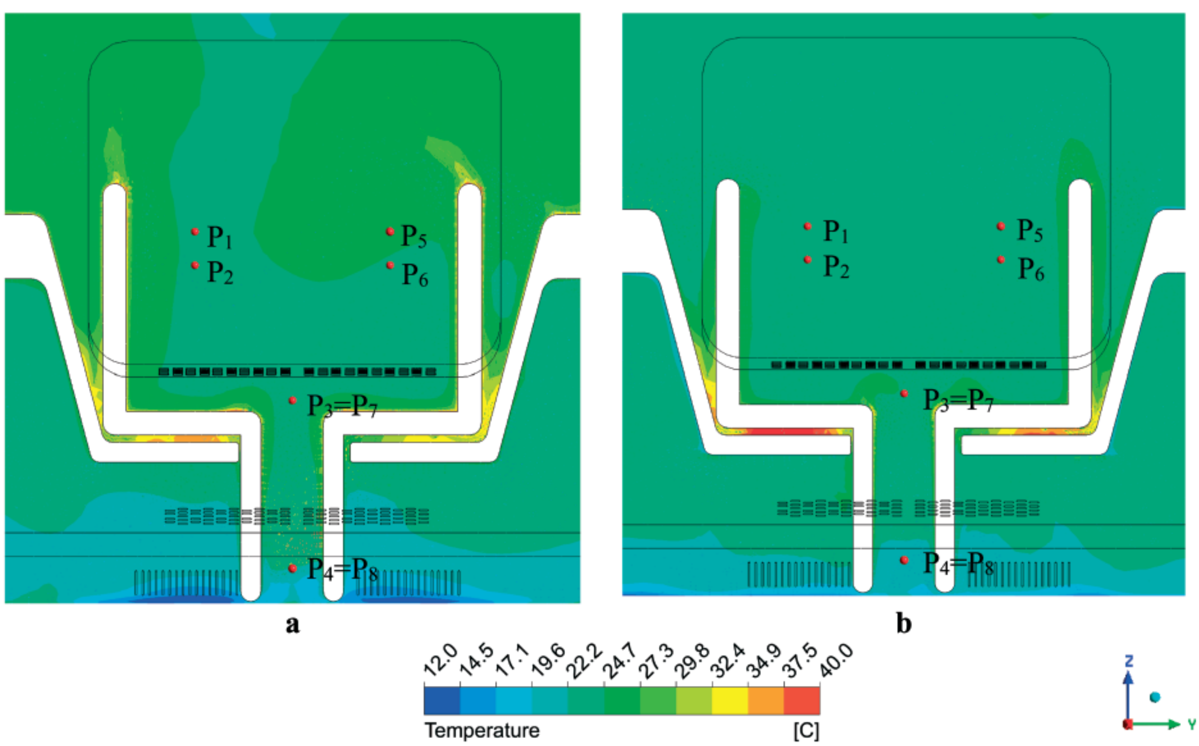

Figure 8.
The comparison of the local distribution of air temperature for passengers, in the vertical plane YZ, X=2.55 m (Plane 1), a) summer, b) winter

Table 3.
Results of air parameters and thermal comfort indicators at monitoring points 1–36

Axis	Monitoring point	Summer						Winter					
		metabolic rate = 1 met clothing insulation = 0.5 clo						metabolic rate = 1 met clothing insulation = 1.25 clo					
		t_i	w	RH	PMV	PPD	DR	t_i	w	RH	PMV	PPD	DR
		°C	m/s	%		%	%	°C	m/s	%		%	%
1	1	24.5	0.13	54.7	1.9	72.0	6.3	24.8	0.13	39.2	0.4	8.5	6.3
	2	23.9	0.23	56.2	1.3	40.7	10.9	24.5	0.12	40.6	1.0	24.9	5.5
	3	21.8	0.04	60.6	0.3	7.2	0.0	21.9	0.16	45.9	0.6	12.8	9.8
2	4	24.8	0.15	56.2	0.9	22.9	6.8	24.8	0.10	38.9	0.5	9.8	4.7
	5	23.0	0.29	59.9	0.0	5.0	14.2	23.9	0.10	41.6	0.9	23.2	4.9
	6	23.6	0.18	57.6	0.4	7.9	9.5	22.8	0.06	43.5	0.9	21.3	2.4
3	7	24.9	0.11	53.6	2.1	82.8	4.8	25.0	0.13	39.0	0.4	8.7	6.0
	8	24.3	0.19	55.0	1.7	62.0	8.9	24.3	0.12	42.0	0.9	23.0	5.9
	9	21.8	0.11	59.1	0.4	8.7	6.3	23.0	0.08	43.0	1.0	24.9	3.8
4	10	24.9	0.11	58.2	1.3	42.9	4.8	24.7	0.17	40.4	0.4	7.9	7.8
	11	23.7	0.23	58.2	0.5	11.4	11.3	25.4	0.09	40.4	1.1	30.4	3.8
	12	23.5	0.24	57.3	0.2	6.3	11.8	22.6	0.05	44.8	0.9	22.9	0.6
5	13	24.7	0.13	54.1	2.0	78.7	6.1	24.3	0.20	39.5	0.3	6.5	9.2
	14	24.2	0.23	55.6	1.5	53.1	10.6	23.8	0.19	43.4	0.7	15.8	9.4
	15	22.2	0.03	59.7	0.5	10.0	0.0	22.3	0.13	45.2	0.6	12.8	7.8
6	16	24.6	0.16	56.5	1.0	24.8	7.8	24.7	0.13	39.6	0.4	8.5	5.9
	17	24.0	0.19	58.0	0.8	17.6	9.5	24.7	0.13	39.5	0.9	24.0	6.4
	18	23.8	0.20	57.0	0.4	7.8	9.8	21.9	0.12	45.8	0.7	15.5	7.0
7	19	24.3	0.10	55.8	2.0	78.4	5.0	24.6	0.14	39.6	0.4	7.7	6.6
	20	23.9	0.27	56.1	1.3	41.3	12.5	25.1	0.16	40.9	0.9	24.1	7.1
	21	19.7	0.03	66.8	-0.1	5.3	0.0	21.2	0.03	50.1	0.7	14.0	0.0
8	22	24.9	0.20	55.6	0.9	22.0	8.8	24.5	0.15	40.1	0.3	7.2	7.3
	23	23.7	0.18	58.7	0.7	14.7	9.4	24.1	0.11	42.6	0.9	22.8	5.5
	24	23.8	0.20	57.6	0.5	10.1	10.0	21.5	0.11	47.6	0.7	14.5	6.8
9	25	24.9	0.10	54.1	2.1	83.1	4.3	24.4	0.14	39.5	0.3	6.5	6.9
	26	24.4	0.16	55.4	1.9	70.5	7.8	24.6	0.15	41.7	0.8	19.0	7.3
	27	22.1	0.21	59.3	-0.1	5.1	12.1	21.6	0.04	47.5	0.7	14.1	0.0
10	28	24.6	0.14	56.4	1.1	28.5	6.9	24.3	0.18	40.2	0.2	6.0	8.6
	29	24.3	0.13	56.5	1.2	33.4	6.4	23.8	0.11	46.1	0.8	20.1	5.7
	30	22.7	0.32	60.2	-0.4	8.2	15.7	20.5	0.04	51.3	0.5	10.1	0.0
11	31	24.0	0.10	56.2	1.8	69.2	5.0	24.4	0.13	40.6	0.3	6.8	6.3
	32	23.6	0.25	56.9	1.2	34.1	12.2	23.3	0.10	42.3	0.7	16.5	5.1
	33	20.8	0.03	63.5	0.1	5.1	0.0	20.9	0.08	48.9	0.6	11.7	4.3
12	34	24.9	0.16	55.2	1.0	25.6	7.3	24.3	0.14	40.8	0.3	6.4	6.9
	35	24.1	0.15	57.2	0.9	20.7	7.7	24.6	0.14	43.3	0.8	19.5	6.5
	36	22.8	0.32	59.5	-0.4	7.6	15.6	20.8	0.04	52.1	0.5	11.0	0.0
Mean		**23.7**	**0.17**	**57.5**	**0.9**	**31.2**	**7.9**	**23.6**	**0.12**	**43.3**	**0.6**	**15.0**	**5.5**

12, where there are points 34, 35, 36 at heights of 1.1 m, 0.6 m and 0.1 m respectively. The results of the comfort indicators are summarized in Table 3.

As before in points 1–12, the conditions of thermal comfort were closer to ideal in the winter instead of summer. The mean PMV value for all points was 0.9 and 0.6 for summer and winter, respectively. The highest value of this indicator occurred for the summer in the axes 3, 7 and 9, where at a height of 1.1 m PMV index exceeded the value of 2. In these places, the percentage of dissatisfied people was very high and oscillated around the value of 80%. In the case of winter, the PMV values were more even and only at three points did they reach the value of 1.0 or 1.1: on axis 1 at a height of 0.6 m and on axis 3 at a height of 0.1 m and on axis 4 at a height of 0.6 m. Therefore, for summer and winter, the most unfavorable conditions occurred on the same side of the coach, facing southeast. The value of PMV in this part of the coach was determined by the average radiant temperature, for which the mean value was 35.3°C for summer and 26.6°C for winter.

Analyzing the differences in the values of PMV and PPD at different heights, it was found that in the case of winter, the higher PMV values were for the height of 0.1 m and 0.6 m, and at the height of 1.1 m there were comfort conditions ($-0.5 \leq PMV \leq 0.5$) at each of the points 1–36. For the summer, the most comfortable conditions were at a height of 0.1 m. At this level, the PMV value was also within the recommended limits at every monitoring point. The highest values of PMV, in turn, concerned the height of 1.1 m. For both summer and winter, the same relationships between values and heights occurred for the PPD index. On the other hand, the risk of draft was highest in the summer at the height of 0.1 or 0.6 m. Except that, in general, low values of the DR index were obtained: an average of 7.9% for summer and 5.5% for winter.

The local vertical range of the extreme interior temperature for passengers is shown in Fig. 8. It was decided to select passengers around axis 9 (Fig. 7) with the highest PPD values at its monitoring points. The vertical gradient was determined separately for the passengers on the right and left side as the difference between the maximum and minimum temperature values at points P1-P4 (left side) and P5-P8 (right side). In this case, the height of the points corresponded to the height of the head (P1, P5: 1.1 m), shoulders (P2, P6: 1.0 m), knees (P3, P7: 0.6 m) and feet (P4, P8: 0.1 m) of passengers. In summer it was higher than the limited value of 3 K. For passengers on the left side, it was equal to 4.5 K, however, for passengers seated on the right side, it was within the acceptable range. In winter, it was below 3 K, thus it met the requirements of comfort both on the left and right side.

5. CONCLUSIONS

The paper presents numerical calculations of airflow and heat exchange in the open saloon passenger coach for summer and winter conditions. The air parameters and thermal comfort indicators in different monitoring points according to standards in the coach were obtained by CFD simulations and were used to assess the thermal comfort of sitting passengers. It was found that:

1. The thermal comfort conditions of the passengers in the coach were not fully satisfactory, especially in summer. The air distribution did not provide all the comfort conditions specified in railway standards [10,11]. Moreover, in the summer period, the indoor conditions were the most uncomfortable, assessing them in terms of the values of PMV and PPD indicators defined by P. O. Fanger [7]. The highest PMV value exceeded 2 on the sunny side of the coach, which means a feeling of warmth for passengers in this area.
2. The required mean internal temperature was not achieved. It was similar in summer and winter, however, it was too high in winter t_{mi} = 24.2°C and too low in summer t_{mi} = 24.4°C.
3. The obtained global vertical range of the extreme interior air temperatures exceed 3 K in summer and winter condition. The local vertical gradient was higher than the limited value for summer conditions. It was caused by the stratification of temperature. To prevent this phenomenon, it is suggested to change the ventilation air distribution method so that cooling air is first directed towards the top of the coach, where the warmer air stream accumulates.
4. The obtained horizontal range of the extreme interior air temperatures for summer was higher than 2 K. In winter, the horizontal range of the extreme interior air temperatures and the relative humidity met the requirements.
5. In the area of seated passengers, the air speed did not exceed the permissible values for the summer period, and in the winter period only for small areas. Low values of the DR index were also obtained. The highest values were in the summer in the lower parts of the coach.
6. The insolation of the external partitions, which affects the high radiation temperature, has a significant impact on the feeling of thermal comfort in summer. This study did not take into account the roller blinds with which passenger coaches are equipped.
7. The comfort assessment procedure following the railway standards [10, 11] and the standards applicable to buildings [8, 9] are different, however, in both cases, it gives similar results. Based on each method, the occurrence of areas of discomfort was indicated.
8. In order to improve the thermal comfort, the following changes should be considered: air distribution in the coach, modify the design of diffusers and guide the air deflectors in the supply elements, as well as protection against direct sunlight.

ACKNOWLEDGMENTS

The work was supported by the Polish Ministry of Science and Higher Education within the research subsidy.

REFERENCES

[1] Ferdyn-Grygierek J., Sarna I., Grygierek K. (2021). Effects of Climate Change on Thermal Comfort and Energy Demand in a Single-Family House in Poland, *Bulidings*, 11, 12, 1–17.

[2] Grygierek K., Sarna I. (2020). Impact of Passive Cooling on Thermal Comfort in a Single-Family Building for Current and Future Climate Conditions, *Energies*, 13, 5332.

[3] Lipczyńska A., Kaczmarczyk J., Melikov A. K. (2015). Thermal environment and air quality in office with personalized ventilation combined with chilled ceiling, *Building and Environment* 92,603–614.

[4] Kaczmarczyk J., Ferdyn-Grygierek J. (2020). Thermal comfort and energy use with local heaters, *Energies*, *13*(11), 1–14.

[5] Kaczmarczyk J., Lipczyńska A., Kateusz P. (2017). Indoor environment quality evaluation in dwellings: a Polish case study, *Architecture Civil Engineering Environment, 10*(4), 163–171.

[6] Nastase I., Danca P., Bode F., Croitoru C., Fechete L., Sandu M. Coşoiue I, C. (2022). A regard on the thermal comfort theories from the standpoint of Electric Vehicle design — Review and perspectives, *Energy Reports*, 8, 10501–10517.

[7] Fanger P. Thermal comfort. Analysis and Applications in Environmental Engineering, McGraw-Hill, USA, 1972.

[8] EN ISO 7730: 2005 Ergonomics of the thermal environment – Analytical determination and interpretation of thermal comfort using calculation of the PMV and PPD indices and local thermal comfort.

[9] ANSI/ASHRAE Standard 55-2010 Thermal Environmental Conditions for Human Occupancy.

[10] EN 13129:2016: Railway applications – Air conditioning for main line rolling stock – Comfort parameters and type tests.

[11] UIC leaflet 553 Heating, ventilation and air-conditioning in coaches – Standard tests.

[12] Sarna I., Palmowska A. (2019). Modelling of the airflow in the passenger coach. *Architecture Civil Engineering Environment, 12*(4),125–133.

[13] Zhang Z., Lu Y. Numerical study on air quality and thermal comfort in high speed train compartment with underfloor air supply. Proceedings of the 2nd International Conference on Industrial Aerodynamics, Qingdao, China, 18-20 October 2017.

[14] http://www.railway-technical.com/trains/rolling-stock-index-l/coach-parts.html, Available online: 07.04.2018.

[15] Aliahmadipour M., Abdolzadeh M., Lari K. (2017). Air flow simulation of HVAC system in compartment of a passenger. *Applied Thermal Engineering*, 8, 973–990.

[16] Goelz P., Orellano A. Simulation of transient thermal comfort in trains, Proceedings of the 5th International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics, Sun City, South Africa, 1-4 July 2007.

[17] Karthik K., Nasrulla M., Raj T. K., Karthick N. Krishnakanth S.(2021). Comparative CFD analysis on conditioned air flow and temperature distribution in metro train for different cities of India, *International Journal of Modern Agriculture*, 10, 10.

[18] Schmeling D., Bosbach J.(2017). On the influence of sensible heat release on displacement ventilation in a train compartment, *Building and Environment*, 125, 248–260.

[19] Meyenberg M., Schemling D., Winter S. A moving thermal manikin for the simulation of walking passengers in aircraft of trains, Proceedings of the Roomvent & Ventilation, Espoo, Finland, 2–5 June 2018.

[20] Zhang Y., Li J., Sun H., Liu J., Chen Q. (2015). Evaluation of different air distribution systems for sleeping spaces in transport vehicles, *Building and Environment*, 94, 665–675.

[21] Yang L., Li X., Tu J. Numerical study of cabin interior air environment response of high-speed trains passing each other, Proceedings of the 4th International Conference On Building Energy, Environment, Melbourne, Australia, 5–9 February 2018.

[22] Palmowska A., Walczyk K. CFD Modelling of The Airflow in The Driver's Cabin of a Modern Rail Vehicle. Proceedings of the 5th World Congress on Mechanical, Chemical, and Material Engineering (MCM'19), Lisbon, Portugal, August 15–17, 2019.

[23] Alam M., S., Salve U. R. (2020). Enhancement of thermal comfort inside the kitchen of non-airconditioned railway pantry car, International Journal of Heat and Technology, 39, 1, 275–291.

[24] Konstantinov M., Wagner C. Flow and Thermal Comfort Simulations for Double Decker Train Cabins with Passengers. Proceedings of the Third International Conference on Railway Technology: Research, Development and Maintenance, Cagliaru, Italy, 5-8 April 2016.

[25] Ghosh S., Bharadwaj S. J., Bharadwaj S. J. A design study on a proposed high speed train service in Western India A synergistic dialogue of optimal speed and cabin comfort using CFD. Proceedings of the 2017 IEEE Region 10 Symposium, Penang, Malaysia, 5-8 November 2017.

[26] Autodesk (2011) 'Autodesk Ecotect Analysis', products. Available online: http://www.cadpoint.co.uk/ecotectanalysis/(accessed 29 December 2016).

[27] Haller G. (2006). Thermal Comfort in Rail Vehicles. RTA Rail Tec Arsenal Fahrzeugversuchsanlage GmbH, Vienna.

[28] Pełech A. (2009). Wentylacja i klimatyzacja – podstawy (Ventilation and air-conditioning – fundaments), Oficyna Wydawnicza Politechniki Wrocławskiej, Wrocław.

[29] Lipska B., Palmowska A., Ciuman P., Koper P. (2015). Modelowanie numeryczne CFD w badaniach i projektowaniu rozdziału powietrza w pomieszczeniach wentylowanych (CFD numerical modelling in research and design of air distribution in ventilated rooms), *Instal*, 3, 33–43.