ARCHITECTURE CIVIL ENGINEERING ENVIRONMENT
The Silesian University of Technology No. 4/2019

doi: 10.21307/ACEE-2019-058

MODELLING OF THE AIRFLOW IN THE PASSENGER COACH

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

[a] MSc Eng.; Faculty of Energy and Environmental Engineering, The Silesian University of Technology, Konarskiego 18a, 44-100 Gliwice, Poland
E-mail address: *izabela.m.sarna@gmail.com*

[b] PhD; Faculty of Energy and Environmental Engineering, The Silesian University of Technology, Konarskiego 18a, 44–100 Gliwice, Poland

Received: 7.08.2019; Revised: 30.09.2019; Accepted: 2.10.2019

Abstract

In this paper currents requirements of HVAC designing (Heating, Ventilation and Air Conditioning) in railway vehicles have been presented. The data were based on railway standards [1, 2].
The aim of this study was to carry out the numerical calculation of airflow combined with heat exchange in a passenger coach. ANSYS CFX 12.1 software was used to carry out the simulation. Two cases of boundary conditions were considered, the first obtained from design calculations common for ordinary buildings and information included in standards and the second only based on the information included in standards. After analysing of the results, it was found that the distribution of air velocity in a coach was similar in both cases, average air velocity was 0.79 m/s. However, the distribution of air temperature was different. For case 1 the average indoor air temperature was 25.07°C and for case 2 was 23.53°C. The method of determining the heat solar gains had a great impact on the results. A further possibility of a model improvement was indicated for example human models will be introduced in coaches, in order to verify the conditions of their thermal comfort, and air recirculation.

Keywords: **Airflow; Coach; CFD; Heat exchange; Ventilation.**

NOMENCLATURE

t_{em} – mean exterior temperature, °C

t_i – indoor temperature, °C

t_e – exterior temperature, °C

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

$\dot{m}_{S1}$ – mass flow rate of air supply by lower air diffusers, kg/s

$\dot{m}_{S2}$ – mass flow rate of air supply by window air diffusers, kg/s

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

t_{s1} – sol-air temperature (lower air diffusers), °C

t_{s2} – sol-air temperature (window air diffusers), °C

t_s – sol-air temperature, °C

g – solar factor through windows, %

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

RH – relative humidity, %

w – velocity, m/s

w_{max} – maximum velocity, m/s

w_{min} – minimal velocity, m/s

1. INTRODUCTION

Every day people spend a lot of time in public transport, for example buses, trams, trains or subways. To make their journey comfortable, appropriate thermal conditions must be provided by HVAC (Heating, Ventilation, Air Conditioning) systems. Thermal comfort is achieved when passengers perceive the air temperature, humidity, air movement, and heat radiation of their surroundings as ideal and would not prefer warmer or colder air or a different humidity level [3].

Passengers often decide to travel by railway vehicles. For example, during the 2017 year over 7.5 million passengers travelled by train in Europe and went almost 600 000 million kilometres [4]. Thus, the correctness of design and work of the HVAC system is one of the most important issues in passenger coaches. More and more modern trains have been produced for a few years, however, the work of ventilation and air-conditioning systems is dissatisfactory. In summer passengers often complain about the method of air distribution. In older passenger coaches, too high indoor temperature occurs. Moreover, in some zones of the compartment the air velocity distributions are uneven, as a result, draught and discomfort zones are created. In modern, air-conditioned railway vehicles passengers are also often dissatisfied, due to too low temperature of the supplied air, especially at the head level. Therefore, HVAC systems should be improved and optimized to achieve better thermal comfort for their users.

Computational Fluid Dynamic (CFD) software is used to predict the air distribution in different ventilated rooms, for example in offices, sports halls, swimming pools, ice rinks and even in vehicles. For the railway industry, thermal comfort in passenger vehicles was described in few studies, for example [5, 6, 7, 8, 9, 10]. Zhang [5] simulated and compared three different air distribution systems for sleeping spaces in a transport vehicle. They were displacement, personalized and mixing ventilation systems. After the research, it was found that the best air quality for sleeping passengers was personalized ventilation. Aliahmadipour [6] studied the airflow in a compartment of a passenger coach in summer conditions. The research was conducted for two cases in presence of seated/slept manikins and without. The first results showed the non-symmetrical air distribution, so the thermal comfort was unfulfilled in the part of the compartment. Due to performing some simple modifications of the HVAC system, the thermal conditions improved. Thermal comfort was also investigated in double-decker train cabins with passengers in summer and winter conditions [7]. Schmeling and Bosbach [10] studied the influence of heat release on ventilation efficiency and thermal comfort with the use of thermal manikins. Different ventilation systems in passengers' rail vehicles were observed in the literature. Aliahmadipour [6] used a mixing system, where the fresh air was delivered by the main air duct located under the floor of the coach. The displacement ventilation was used by Schmeling and Bosbach [10] and Konstantinov [7]. The air inlet was located in the ceiling. Numerical simulations were made for different vehicles by researchers, for example, Suárez [8] studied air distribution in trams and Bosbach [9] compared six different ventilation concepts (mixing ventilation, cabin displacement ventilation, ceiling based displacement as well as combined ceiling and floor-based displacement ventilation for the aircraft cabin. Due to the application of CFD, it was possible to simulate the distribution of air parameters in vehicles and also better optimization of the ventilation for subway side-platforms in Tianjin Metro. It would help to build a more dependable, effective and ventilation system [11]. Moreover, simulations CFD can be used to predict coach fire. Chen [12] simulated and analyzed the impact of the ventilation factor on the burning characteristic of coach fire.

Generally, the design of HVAC systems in the passenger coach may be a source of difficulties and doubts. This issue is a great challenge for the constructors. The aim of this paper was to prepare a numerical model of passenger coach and carry out the numerical calculation of the airflow. The simulations were made for two cases of boundary conditions: 1) obtained by using design calculations common for ordinary buildings, 2) obtained by using information included in standards [1, 2]. This was to answer the question of many constructors, which approach in design will be better to obtain the required parameters of thermal comfort. Prior to this work, the boundary conditions have been determined based on the results of measurement [11],[6],[10] or assumed values in computational simulations [8]. The information included in standards was not taken into account.

2. VENTILATION AND AIR-CONDITIONING PASSENGER COACH – CURRENT REQUIREMENTS

Current requirements and guidelines for the design HVAC systems in railway vehicles are found in European standards PN-EN 13129 [1] and UIC leaflet 553 [3]. They determine thermal comfort parameters by extension, the capacity of the air conditioning systems under defined conditions and describe the testing program and the measurement procedures for estimating the HVAC system [3]. The air distribution systems issues were not included in these documents.

The main requirements are described in this chapter. The climatic zone, which is taken into account in the selection of ventilation, heating or air conditioning devices for railway vehicles, is different than the one commonly known in building engineering. First of all, one of three zones, separately for summer and winter,

Table 1.
Parameters of outdoor air – summer conditions

Climatic zone	Maximum temperature, relative humidity, equivalent solar load
I	40°C, 40%, 800W/m^2
II	35°C, 50%, 700 W/m^2
III	28°C, 45%, 600W/m^2

Table 2.
Parameters of outdoor air – winter conditions

Climatic zone	Minimal temperature
I	-10°C
II	-20°C
III	-40°C

Table 3.
The minimum total volume flow of fresh air for railway vehicles with air conditioning device

Exterior temperature (t_{em})	Minimum fresh air rate equivalent to +20°C and 50% rel. hum, normal atmospheric pressure
t_{em} < -15°C	10m^3/h /passenger
-15°C ≤ t_{em} ≤ -5°C	15m^3/h /passenger
-5°C ≤ t_{em} ≤ +26°C	20m^3/h /passenger
t_{em} > 26°C	15m^3/h /passenger

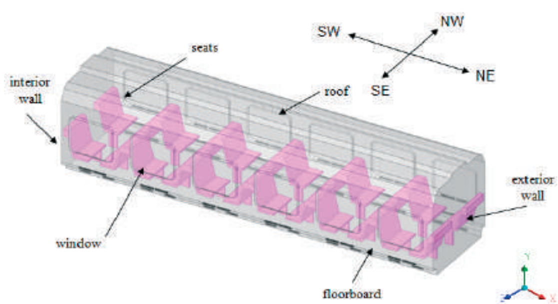

Figure 1.
The numerical coach model – geometry

is assigned for each country, in which vehicle is homologated. Central European countries, including Poland, are placed in II zone for both summer and winter conditions. Parameters of outdoor air are assigned for each zone, for example. for II zone, in summer, outdoor air temperature should not exceed t_e=35°C, RH=50% of relative humidity and a solar load equals to 700 W/m^2. Other information was shown in Tab. 1 and Tab. 2. The heat transfer coefficient is also determined on the basis of zones and its values are different for single and double deck vehicles. For the single deck it equals to 2.0 W/(m^2·K), 1.6 W/(m^2·K), and 1.2 W/(m^2·K), for I, II and III climatic zone, respectively. For double deck the coefficient is 2.5 W/(m^2·K), 2.2 W/(m^2·K) and 2.0 W/(m^2·K) for I, II and III climatic zone.

The minimal amount of fresh air per one seating or bed should be in the range of 10-20 m^3/h, depending on the outdoor air temperature (Tab. 3). According to UIC 553 leaflet [2], it is recommended to increase the airflow up to 5 m^3/h for each place for a smoker. In order to obtain thermal comfort conditions, the adequate parameters should be provided. The indoor air temperature is determined on the basis of the adjustment curve, depending on the outdoor air temperature. Velocity and relative air humidity are read on the basis of limiting curves as a function of the indoor air temperature. In air-conditioned passenger coach velocity should be in the range of 0.07–0.60 m/s, air temperature 22–27°C and relative humidity about 52–60%. Excluding comfort zones, airspeed must not be lower than 0.05 m/s. More information about comfort parameters can be found in standards and UIC leaflet [1, 2].

3. METHODS

3.1. Description of the numerical coach model.

The numerical model of passenger coach was based on real data. It consisted of a restaurant part, open compartment and sanitary. The open compartment was only included in the scope of research. This part of the passenger coach was named as “the coach” in the paper (Fig. 1). The overall dimensions of the coach were 10.6×2.6×2.5 m and the construction of the vehicle was symmetrical. The walls of the compartment were exterior except for the partition wall. There were six identical, double glazed windows on longitudinal walls. The original model was highly detailed and calculation time would be very long. Therefore, the model was simplified in software ANSYS CFX. Unnecessary elements, such as construction elements, were deleted or modified. Additionally, the seats for passengers were designed.

3.2. Grid data

An unstructured grid was used in every variant of calculation. It was mostly built with tetrahedral elements (7273308). The total number of elements was 8591099. The element size was in the range of 15–50 mm (Fig. 2). Local refinement of mesh was included around inlets and outlets, where maximum edge length was 2 mm. The inflated boundary was used for every element of the coach, except diffusers. It consisted of five layers of prismatic elements with a maximum thickness of 200 mm (Fig. 3).

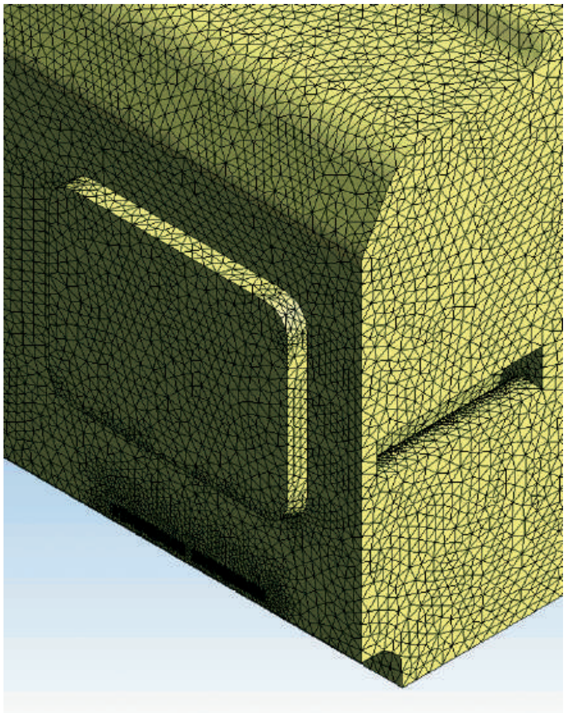

Figure 2.
The discretization grid on the side of the exterior partitions of the coach

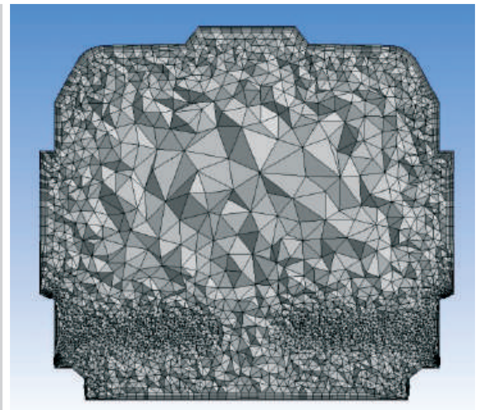

Figure 3.
The discretization grid with inflated boundary – the cross-section

3.3. Boundary conditions and design assumptions

The boundary conditions were based on available information contained in Polish standard [1], the UIC leaflet 553 [5] and design calculation for summer conditions. They were determined in two ways: 1) by using design calculation and based on standards or 2) based only on standards [1]. The results were compared.

In both variants, it was assumed that the coach was in the II climatic zone, in which outdoor temperature was 35°C. Inside the vehicle, the temperature was set to 27°C [1]. The mass flow rate of air supply was based on real data of the HVAC unit, and for the coach, it was $\dot{m}_S$ = 0.605 kg/s. Ventilation ducts were placed under the floor of the coach. The air inlets were located in two places: under windows and seats. Air outlets were placed below window air diffusers (Fig. 4). The air distribution was designed as follows: 20% amount of the air ($\dot{m}_{S1}$) was supplied into the coach by lower diffusers, it was mixed with the indoor air. Then the amount of indoor air ($\dot{m}_E$) was sucked in by outlets. It was mixed with the remaining amount of supply air (80% amount of air) and flowed out ($\dot{m}_{S2}$) though upper grille vents. This phenomenon of induction was accounted for in the simulations. According to the heat balance, air supply temperature was calculated.

Boundary conditions were determined in a different way. For case 1 it was obtained based on design calculation. The solar gains for transparent partitions were calculated by means of Carrier method and for opaque partition they were determined for steady conditions. Heat accumulation was left out due to the very low thickness of exterior walls. The results of these computations were converted into the sol-air temperature. The heat transfer coefficients were

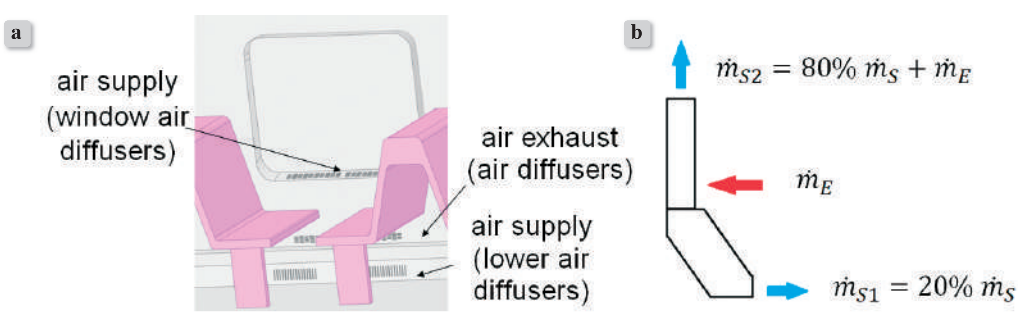

Figure 4.
a) Localization of diffusers, b) scheme of air distribution in the coach, $\dot{m}_S$ – mass flow rate of air supply $\dot{m}_{S1}$ – mass flow rate of air supply by lower air diffusers, $\dot{m}_{S2}$ – mass flow rate of air supply by window air diffusers, $\dot{m}_E$ – mass flow rate of air exhaust

Table 4.
Boundary conditions for case 1

The element of model and kind of boundary conditions	Value of case 1
South-east wall; "Wall" with heat transfer coefficient U and sol-air temperature	$U = 2\ \frac{\mathrm{W}}{\mathrm{m^2 \cdot K}},\ t_s = 61.8°\mathrm{C}$
Interior wall; "Wall"	Adiabatic wall
North-west wall; "Wall" with heat transfer coefficient U and sol-air temperature	$U = 2\ \frac{\mathrm{W}}{\mathrm{m^2 \cdot K}},\ t_s = 38.3°\mathrm{C}$
North-east wall; "Wall" with heat transfer coefficient U and sol-air temperature	$U = 2\ \frac{\mathrm{W}}{\mathrm{m^2 \cdot K}},\ t_s = 38.88°\mathrm{C}$
South-east windows; "Wall" with heat transfer coefficient U and sol-air temperature	$U = 2\ \frac{\mathrm{W}}{\mathrm{m^2 \cdot K}},\ t_s = 157.99°\mathrm{C}$
North-west windows; "Wall" with heat transfer coefficient U and sol-air temperature	$U = 2\ \frac{\mathrm{W}}{\mathrm{m^2 \cdot K}},\ t_s = 48.71°\mathrm{C}$
Roof; "Wall" with heat transfer coefficient U and sol-air temperature	$U = 1.2\ \frac{\mathrm{W}}{\mathrm{m^2 \cdot K}},\ t_s = 53.1°\mathrm{C}$
Floorboard; "Wall" with heat transfer coefficient U and exterior temperature	$U = 1.2\ \frac{\mathrm{W}}{\mathrm{m^2 \cdot K}},\ t_e = 35°\mathrm{C}$
Lower diffusers; "Inlet" with mass flow rate of supply air and temperature of ventilation supply air	$\dot{m}_{S1} = 0.121$ kg/s, $t_{S1} = 19.4°\mathrm{C}$
Upper diffusers; "Inlet" with mass flow rate of supply air and temperature of ventilation supply air	$\dot{m}_{S2} = 0.516$ kg/s, $t_{S2} = 19.9°\mathrm{C}$
Exhaust diffusers; "Outlet" with mass flow rate of exhaust air	$\dot{m}_E = 0.032$ kg/s

Table 5.
Boundary conditions for case 2

The element of model and kind of boundary conditions	Value of case 2
South-east wall; "Wall" with heat transfer coefficient U and sol-air temperature	$U = 1.6\ \frac{\mathrm{W}}{\mathrm{m^2 \cdot K}},\ t_s = 53°\mathrm{C}$
Interior wall; "Wall"	Adiabatic wall
North-west wall; "Wall" with heat transfer coefficient U and sol-air temperature	$U = 1.6\ \frac{\mathrm{W}}{\mathrm{m^2 \cdot K}},\ t_s = 36.23°\mathrm{C}$
North-east wall; "Wall" with heat transfer coefficient U and sol-air temperature	$U = 1.6\ \frac{\mathrm{W}}{\mathrm{m^2 \cdot K}},\ t_s = 36.42°\mathrm{C}$
South-east windows; "Wall" with heat transfer coefficient U and sol-air temperature	$U = 1.6\ \frac{\mathrm{W}}{\mathrm{m^2 \cdot K}},\ t_s = 297.5°\mathrm{C}$
North-west windows; "Wall" with heat transfer coefficient U and sol-air temperature	$U = 1.6\ \frac{\mathrm{W}}{\mathrm{m^2 \cdot K}},\ t_s = 64.31°\mathrm{C}$
Roof; "Wall" with heat transfer coefficient U and sol-air temperature	$U = 1.6\ \frac{\mathrm{W}}{\mathrm{m^2 \cdot K}},\ t_s = 65°\mathrm{C}$
Floorboard; "Wall" with heat transfer coefficient U and exterior temperature	$U = 1.6\ \frac{\mathrm{W}}{\mathrm{m^2 \cdot K}},\ t_e = 35°\mathrm{C}$
Lower diffusers; "Inlet" with mass flow rate of supply air and temperature of ventilation supply air	$\dot{m}_{S1} = 0.121$ kg/s, $t_{S1} = 16.27°\mathrm{C}$
Upper diffusers; "Inlet" with mass flow rate of supply air and temperature of ventilation supply air	$\dot{m}_{S2} = 0.516$ kg/s, $t_{S2} = 16.94°\mathrm{C}$
Exhaust diffusers; "Outlet" with mass flow rate of exhaust air	$\dot{m}_E = 0.032$ kg/s

based on real data from the technical catalogue [13] and engineering practices. For case 2 solar gains for windows were calculated based on the equivalent solar load of 700 W/m^2 [1] and converted from surface gains into sol-air temperature (t_s) [1]. The solar factor through windows (g=60%) was included. Due to the lack of information about the values of the solar gains for opaque walls in standards [1, 2], they were calculated on the basis of the temperature differences, which were taken according to engineering practices. In all solar gains, the attitude of partitions was taken into account and these values were decreased by computational coefficients. The heat balance did not include heat gains from human and lighting. All boundary conditions were shown in Table 4 and Table 5.

3.4. Numerical method

The ANSYS CFX 12.1 software was used to numerical simulations. The Reynolds-averaged Navier-Stokes equations were solved by the Finite Volume Method. The model of the turbulent airflow was based on the Shear Stress Transport turbulence model, which is a combination of k-ε and k-ω model. The Discrete Transfer Model was used to simulate thermal radiation in the coach. Non-slip conditions in near-wall boundary layer were taken into consideration. The calculations were carried out in steady-state, three-dimensional and non-isothermal conditions. In order to control air parameters during the calculations and to check if the solutions are convergent, the monitor points were created. They were located near air inlets and above windows. The simulation of each case lasted 8 days because of limitations in memory and processing power.

4. RESULT AND DISCUSSION

The obtained results of the simulations were studied using the ANSYS CFX-Post module. The results of the calculations were performed in a graphical form (Fig. 5–10). The planes, in which values of air parameters (velocity and temperature) changed in a wide range, were analyzed. The XY, Z=1.804 m vertical plane was a symmetry plane (Fig. 5, 8). The vertical planes YZ were located in different places. The first plane YZ, X=-11.63 m passed through the area of air inlets (Fig. 6, 10), the second plane YZ, X=-9.55 m was placed between windows (Fig.7, 9). According to the Polish standards [14] the boundaries of the thermal comfort were between 0.1÷1.7 m of the height for a standing position and 0.1÷1.1 for a sitting position. Therefore, distributions of air velocity and temperature out of this range were not taken into account in the result description. They were shown for only illustrative purposes. The range of values and colours were selected adequately so that the air distributions were shown legibly. In order to make the accurate comparison of values of air velocity and air temperature from two cases monitoring points were introduced. They were located in the vertical axis of symmetry at the height of 1.95 m (points 1–6) and in the comfort zone at the height of 0.85 m and near the seats (point 7–12).

The comparison of the air velocity distribution for the two cases was shown in Fig. 5–7. The values of them were similar, average air velocity for both variants was 0.79 m/s. In monitoring points (Fig. 11) the values also were similar (without point 6). The

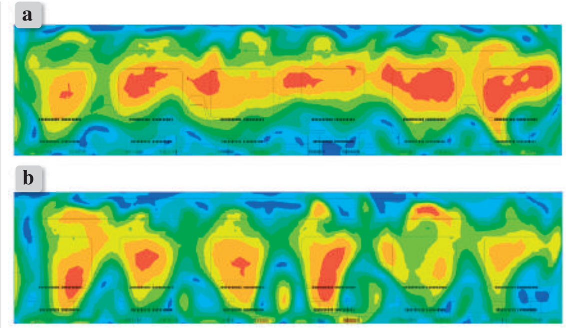

Figure 5.
The comparison of distribution of air velocity in the vertical plane XY, Z = 1.804 m: a) case 1, b) case 2

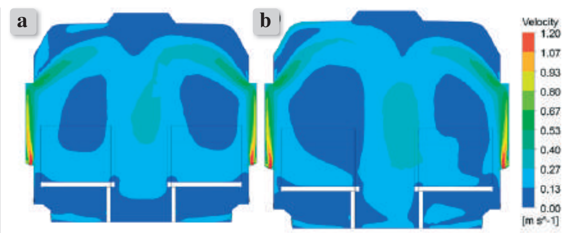

Figure 6.
The comparison of the distribution of air velocity in the vertical plane YZ, X = -11.63 m: a) case 1, b) case 2

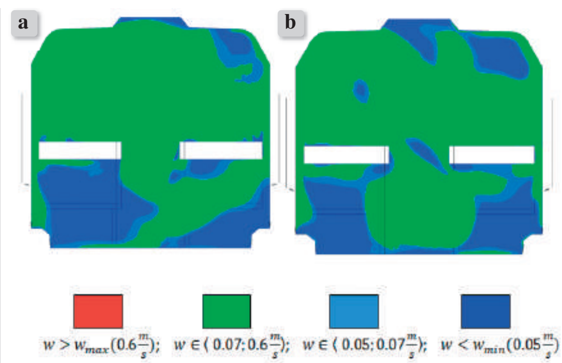

Figure 7.
The comparison of the comfort zone in the vertical plane YZ, X = -9.55 m: a) case 1, b) case 2

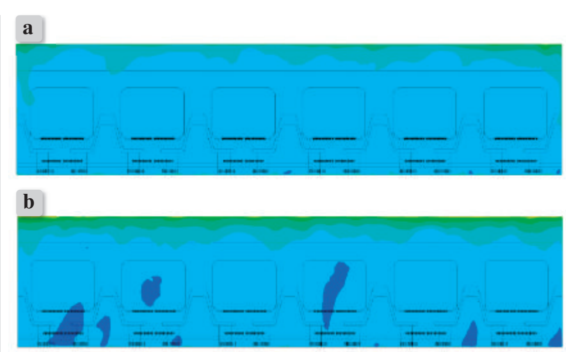

Figure 8.
The comparison of the distribution of air temperature in the vertical plane XY, Z = 1.804 m: a) case 1, b) case 2

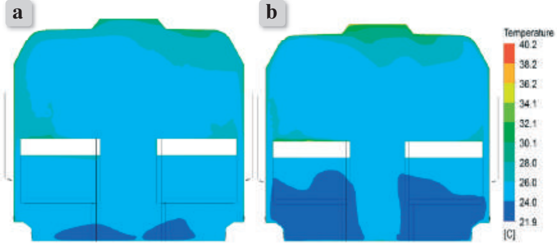

Figure 9.
The comparison of the distribution of air velocity in the vertical plane YZ, X = -9.55 m: a) case 1, b) case 2

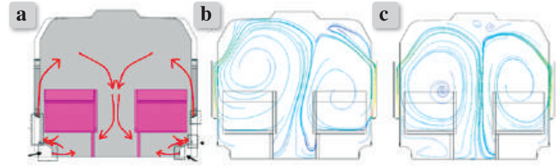

Figure 10.
The comparison of the distribution of air velocity in the vertical plane YZ, X = -11.63 m: a) theoretical distribution, b) numerical distribution for case 1, c) numerical distribution for case 2

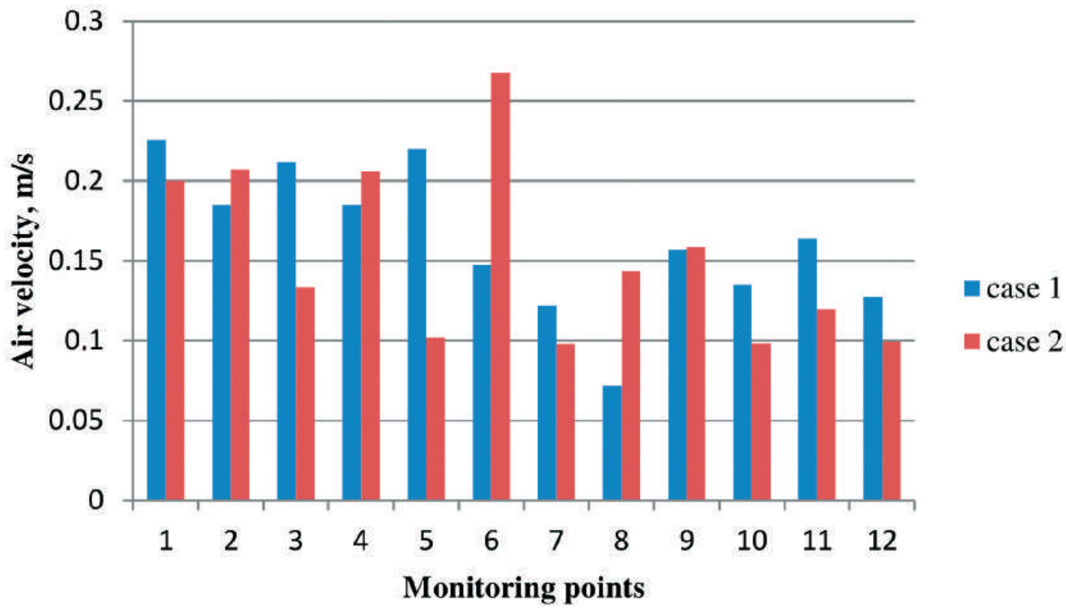

Figure 11.
The comparison of air velocity in monitoring points for case 1 and case 2

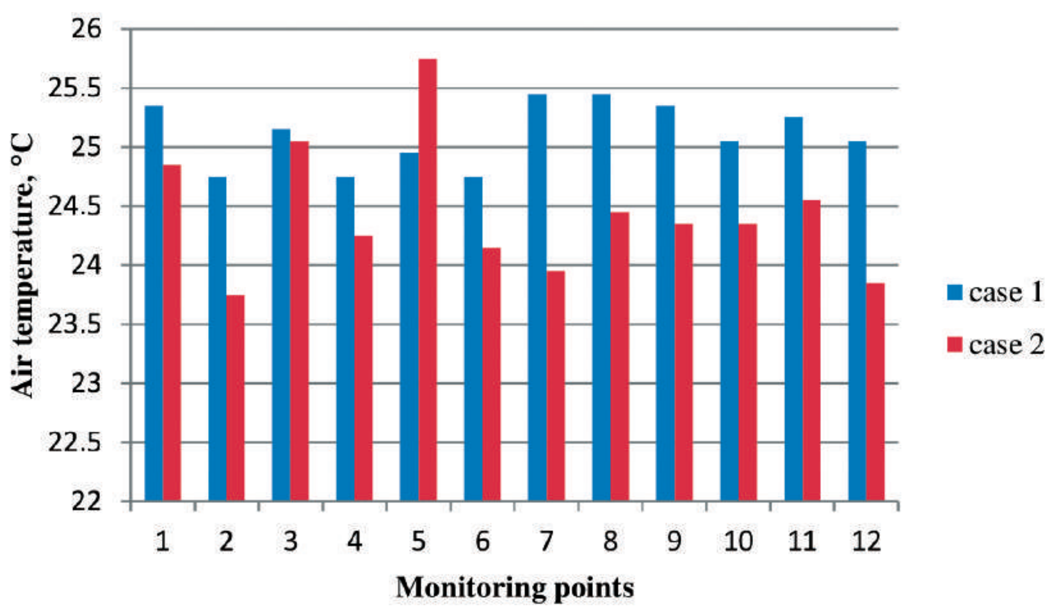

Figure 12.
The comparison of air temperature in monitoring points for case 1 and case 2

airstream flowing out from inlets mixed with the indoor air in a large degree. Due to this the dead zones (w_{max}<0.7 m/s for $t_i = 27°C$, $w_{min} > 0.05$ m/s for $t_i = 18°C$) were located only near floor between seats and near the ceiling where people seldom stay.

For case 1 the average indoor air temperature was 25.07°C. It was higher by about 1.54°C than for case 2. The value of temperature obtained near the floor was (Fig. 8, 9) too low (23–24°C). The stratification of temperature was unnoticeable, the bigger part of the coach temperature was similar (24.7–26°C). For case 2 the average indoor temperature (Fig. 9) was significantly too low $t_i = 23.53°C$, especially in the area from 0.1 to 0.6 m. It was due to cold air outflow from diffusers. According to the design assumption, the indoor temperature should be 27°C (+-1°C), and the range of mean temperature measured 1.1m above the floor should not exceed 2K. The stratification of temperature was at significant level ranging between 22÷26°C. Therefore, passengers could have felt locally uncomfortable. In monitoring points values of air temperature were much higher (without point 5) in case 1 than case 2 (Fig. 12).

The comparison of the numerical distribution of air velocity of simulations for two cases was shown in Fig. 10. It was noted that the air distribution of case 2 was more similar to the theoretical scheme. The streamlines were more symmetrical and regular. However, in both cases, there was a bigger amount of air on the left side of the coach.

5. CONCLUSIONS

By means of CFD, it was possible to take into account most of the phenomena associated with the flow of the air and heat in the passenger coach. It was found that:

1. The main differences in approach to design of HVAC system basing only on standards or common engineering calculations and standards were caused by a different way of determining solar heat gains and other values of the heat transfer coefficient. The value of solar load for case 1 was definitely less than for case 2. According to the standards, the value of the heat transfer coefficient was the same for each partition (wall, window, floorboard, and roof), while in calculation different values were assumed. The results of the research showed that these differences did not influence the air distribution in the coach but the amount cold demand. In order to provide thermal comfort conditions for case 2, the temperature of the air supply was lower than for case 1. This should be taken into account while selecting a cooling unit. The installation, which was designed based on only the standards, works in worse thermal conditions.
2. A different way of inputting boundary conditions, in particular, solar heat gains, was of importance. It influenced the value of the indoor air temperature. Lower values of this parameter were obtained in case 2. It was caused by the fact that standard values of solar heat gains are much higher than those obtained according to design calculations. The value of indoor temperature in both cases (for case 1 $t_1 = 25.07°C$ and for case 2 $t_1 = 25.53°C$) was lower than the designed temperature $t_1 = 27°C$. The amount of solar radiation had a significant impact on the value of air parameters.
3. The air velocity distribution was similar in both cases. The proposed air distribution was at satisfying level, however, there were some discomfort zones, especially in the area of legs. Therefore, air distribution system should be improved to avoid the risk of draught and feeling cold near the passengers' ankles.
4. A further possibility of model improvement is to introduce human models in passenger coach and human gains of heat into the calculations or use air recirculation.
5. The conclusions presented in the paper resulting from the comparative analysis are based only on the results of numerical studies and they require experimental validation in conditions of an actual object, which will be the subject of further research.
6. The main direction of further research is to study the thermal comfort of passengers in the improved model.

REFERENCES

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

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

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

[4] International Union of Railways, Railway Statistics 2017, Retrieved February 2 2019 from https://uic.org/IMG/pdf/uic-statistics-synopsis-2017.pdf

[5] 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.

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

[7] Konstantinov M., Wagner C. (2016). 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.

[8] Suárez C., Iranzo A., Salva J. A., Tapia E., Barea G., Guerra J. (2017). Parametric Investigation Using Computational Fluid Dynamics of the HVAC Air Distribution in a Railway Vehicle for Representative Weather and Operating Conditions. *Energies*, 10, 1074.

[9] Bosbach J., Lange S., Dehne T., Lauenroth G.(2013). Alternative ventilation concept for aircraft cabin. Springer.

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

[11] You S.-J. Yuan F.-D.(2007). CFD simulation and optimization of the ventilation for subway side-platform. *Tunnelling and Underground Space Technology*, 22, 474–482.

[12] Chen J, Yi L., (2013). Influence of Ventilation Status on Combustion Characteristics of Coach Fire, Elsevier. Procedia Engineering, 52, 42–47.

[13] Technical catalogue of Rawag(n.d.). In Schaltbau Rawag information, Retrieved 4 April, 2018 from www.rawag.pl

[14] PN-ISO 7726: Ergonomics of the thermal environment – Instruments for measuring physical quantities.