Numerical simulation of aerodynamics in the cabin of transport

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Abstract. One of the most important tasks of transport engineering is the task of providing optimal and comfortable conditions for drivers and passengers, in particular: temperature, pressure, dust content, humidity, air speed, noise, illumination, etc. Temperature, humidity and vibration influences are also of great importance for the functioning of electronic devices and transport equipment. In this paper, a numerical-mathematical model is presented for studying the fields of temperatures, velocities, pressures, densities and humidity of air in the cabin space with simultaneous blowing by an external flow and internal ventilation. As a mathematical model, the Navier-Stokes equations in the RANS formulation are used, together with the diffusion equation, which allows calculating the mass fractions of water vapor in the domain. As a numerical implementation, the finite volume method is used together with turbulence models in the Ansys Fluent software product. A non-stationary setting is considered, the stabilization time of flows is found; the stabilized fields are compared with the results obtained for the stationary setting. A method for obtaining numerical values of heat transfer coefficients for walls consisting of several layers of materials is also given. As a numerical experiment, a cabin of a special technological transport with a large glass area and multi-layered walls, with one driver without passengers, is considered. The obtained results are compared with the results obtained earlier without taking into account the diffusion equation and relative air humidity modeling.

1 Introduction

The Heating, Ventilation, and Air Conditioning (HVAC) systems are vital parts of modern vehicles that play a vital role in improving passengers’ safety and comfort. These systems help maintain optimal operating conditions for the equipment by regulating noise, temperature, pressure [1,2], dust content [3], humidity, air velocity, and other similar aspects. Deviation from the normal values of these characteristics can have a serious impact on the performance and health of the driver and passengers, as well as equipment. At the same time,

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the optimal operating parameters of the system must be maintained with minimal energy consumption.

Currently, vehicle cabins - even those specifically tailored for use in challenging conditions - are made up of several layers of materials designed for insulation, protection, decoration, and other purposes. As interior materials become more intricate, thermal loads increase, and demands for thermal comfort grow, it is necessary to enhance modeling techniques for the thermal state of these vehicles. The chief objective is to establish a comfortable microclimate inside the cabin that will create ideal circumstances for human work, ventilation, air conditioning, and electronic equipment.

Previous studies have investigated the factors that contribute to a comfortable microenvironment inside the vehicle cabin, as well as the theoretical foundation for modeling this environment using heat flow equations [4-6]. To model heat transfer through the cabin walls, heat flow equations are utilized, taking into account convective exchange with the environment. The parameters used in this modeling process, including wall thermal conductivity and heat transfer coefficients, have been extensively discussed in earlier studies [7-11].

Typically, when it comes to forced convection, solving the Navier-Stokes equations analytically is not feasible, so in order to determine heat transfer coefficients, one must rely on experimental findings derived from the Nusselt criterion, which is based on similarity theory and dimensionless complexes [12-15]. The Nusselt number is what determines the heat transfer coefficient values, and this is influenced by the type of medium flow present in the near-wall regions. Mathematical formulas such as the Frank, Jurgens, and Raman equations are commonly used in order to calculate these coefficients [16,17].

Numerical solutions of hydrodynamic equations can be obtained by utilizing computational fluid dynamics methods which consider intricate boundary conditions. The prevailing research concentrates on carrying out virtual tests to enhance Nusselt values. Several publications, such as [9, 18-20], exhibit the outcomes of numerical investigations and the heat transfer coefficients of diverse heat exchangers, evaporators, coolers, and surfaces.

In the development of sophisticated modeling methods for the thermal environment in vehicle interiors, it is essential to ascertain the convective heat transfer coefficient between internal and external airflows to ensure maximum comfort of occupants and electronic devices. This requires considering the various layers of materials used in cabin construction and increasing heat loads [5, 21]. Additionally, studies have compared heat transfer coefficients with theories that do not account for air humidity and conducted virus particle movement simulations [8; 20]. In [8], the characteristics of the air fields in the vehicle cabin were obtained, as well as the heat transfer coefficients and heat fluxes through the wall without taking into account air humidity; a comparison was made of the results obtained numerically with applied theories based on averaged balance equations. In 24, the movement of Covid virus particles in cabin air currents was considered. Some patterns of particle trajectories are obtained depending on their size, as well as depending on air humidity.

2 Materials and Methods

2.1 Basic equations

The aim of this study is to create a computational technique that can compute the velocity, temperature, and humidity fields, as well as the convective heat transfer coefficients within the vehicle cabin. This will be achieved by utilizing turbulence models and the Navier-Stokes equations, as well as the energy equation. The influence of taking into account the
The equation of diffusion and humidity on the fields of temperatures and velocities is also studied.

The Navier-Stokes equations (1) - (3) and the continuity equation (4) are employed to depict the actions of a compressible viscous gas. The Euler approach and the RANS model of equations are used to describe the medium. All fields are considered averaged:

\[
\frac{\partial u}{\partial t} + u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} + w \frac{\partial u}{\partial z} = \frac{X}{\rho} - \frac{1}{\rho} \frac{\partial p}{\partial x} + \eta \left( \frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2} \right) + \frac{1}{3} \eta \left( \frac{\partial}{\partial x} \left( \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} \right) \right),
\]

\[
\frac{\partial v}{\partial t} + u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} + w \frac{\partial v}{\partial z} = \frac{Y}{\rho} - \frac{1}{\rho} \frac{\partial p}{\partial y} + \eta \left( \frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2} \right) + \frac{1}{3} \eta \left( \frac{\partial}{\partial y} \left( \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} \right) \right),
\]

\[
\frac{\partial w}{\partial t} + u \frac{\partial w}{\partial x} + v \frac{\partial w}{\partial y} + w \frac{\partial w}{\partial z} = \frac{Z}{\rho} - \frac{1}{\rho} \frac{\partial p}{\partial z} + \eta \left( \frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2} \right) + \frac{1}{3} \eta \left( \frac{\partial}{\partial z} \left( \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} \right) \right).
\]

\[
\frac{\partial \rho}{\partial t} + \rho \frac{\partial u}{\partial x} + \rho \frac{\partial v}{\partial y} + \rho \frac{\partial w}{\partial z} = 0.
\]

Here \(u(x, y, z, t), v(x, y, z, t), w(x, y, z, t)\), are the medium velocities. \(p(x, y, z, t)\) – pressure, \(\rho(x, y, z, t)\) – density. \(\eta = \mu/\rho\), where \(\eta\) and \(\mu\) are the constant viscosities.

To take into account the mass amount of water vapor in the air, the energy equation (5), the equation of diffusion and mass transfer (6) and (7) are used [9; 21]:

\[
\frac{\partial T}{\partial t} + \vec{u} \cdot \nabla T = \frac{\lambda}{c_p \rho} \Delta T - \nabla \cdot \sum_{i=1}^{n} \vec{J}_i + \phi,
\]

\[
\frac{\partial \rho Y_i}{\partial t} + \nabla \cdot (\rho \vec{u} Y_i) = \nabla \cdot \vec{J}_i, \quad i = 1..N - 1,
\]

\[
\sum_{i=1}^{n} Y_i = 1.
\]

Here \(T = T(x, y, z, t)\), is the temperature, \(\vec{J}_i\) – diffuse flow of the \(i\)-th substance; \(\lambda\) and \(c_p\) are the thermal conductivity and specific heat capacity, are assumed to be constant, and \(\phi\) are the internal heat sources. \(Y_i\) – mass local fraction.

The Mendeleev-Clapeyron relation is used as a gas model: (8):

\[
p = \frac{\rho T}{M}.
\]

Boundary conditions for equations (1) - (8) are given in [5; 16]. For the velocities on the walls, the condition of adhesion (equal to zero) of the velocities (9) is set, and the value of the initial velocities (10) is set at the inputs.

\[
\vec{u}|_{\Omega} = 0,
\]

\[
\vec{u}|_{t=0} = \vec{u}_0.
\]
The total outlet pressure is equal to atmospheric pressure.

For the fields of temperatures and concentrations, either the values of the quantities or their fluxes are specified (12) - (13):

\[ T|_\Omega = T^*, \quad -\lambda \frac{\partial T}{\partial n}|_\Omega = q^* = -\alpha(T - T_{out}). \] (11)

\[ Y_i|_\Omega = Y_i^*, \quad -\frac{\partial Y_i}{\partial n}|_\Omega = J_i^*. \] (12)

For a non-stationary problem, the initial values of all fields are additionally set. As a result, we obtain a complete boundary value problem with equations (1) - (8) and boundary conditions (9) - (13).

2.2 Numerical implementation

The averaging of fields according to formulas (13) is applied for the numerical solution of hydrodynamics equations:

\[ f = \bar{f} + f', \] (13)

here \( f' \) — pulsation, \( \bar{f} = \frac{1}{T} \int_{t-T/2}^{t+T/2} f(\tau) \, d\tau. \)

Further in the text, the band above the averaged characteristics is not indicated.

The solution of the boundary value problem is carried out numerically in Ansys CFD using the k-\( \omega \) turbulence model [2; 6; 17]. In this case, the choice of the turbulence model was determined by its universality.

Figure 1a depicts a rectangular technological transport cabin situated in a wind tunnel. The cabin has an inlet that allows air to enter at a predetermined speed, temperature, and humidity. Meanwhile, the outlets maintain a zero relative pressure. To simplify the problem, all findings are presented for the average section of the cabin using a flat approach. To simplify the calculations, the internal equipment, as well as heat dissipation from the equipment, were not taken into account.

\[ \text{Fig. 1. Cabin view: a) Cabin inside (mesh), b) Cabin inside and outside (mesh).} \]
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Table 1 presents the characteristics of the domain grid.

Table 1. Domain grid characteristics

<table>
<thead>
<tr>
<th>Parameter name</th>
<th>Parameter value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of cells</td>
<td>33625</td>
</tr>
<tr>
<td>Number of wall layers</td>
<td>6</td>
</tr>
<tr>
<td>Minimum cell area, m²</td>
<td>1.1 e-3</td>
</tr>
<tr>
<td>Maximum cell area, m²</td>
<td>3.7 e-3</td>
</tr>
</tbody>
</table>

2.3 Task parameters

The cabin wall consists of layers of different materials with characteristics in Table 2:

Table 2. Cabin wall characteristics

<table>
<thead>
<tr>
<th>Layer material</th>
<th>Coefficient of thermal conductivity, $\lambda_i$, W/(m·K)</th>
<th>Layer thickness, $\delta_i$, mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Metal</td>
<td>58</td>
<td>3</td>
</tr>
<tr>
<td>Polyurethane</td>
<td>0.32</td>
<td>30</td>
</tr>
<tr>
<td>Bituminous mastic</td>
<td>0.27</td>
<td>5</td>
</tr>
</tbody>
</table>

Compiled by the authors.

For the values from Table 2, the thermal conductivity of the walls is 8.9 W / (m·K) with a thickness of 38 mm.

The floor is thermally insulated, the heat flow is zero. The thermal conductivity coefficient for laminated glass is taken equal to 0.96 W/(m·K), calculated by the formulas (14) – (15):

$$R = \sum \frac{\delta_i}{\lambda_i}, \quad (14)$$

$$\alpha = \frac{1}{R}, \quad (15)$$

In Table 3 shows the initial values of the parameters.

The omission of the reduction in the proportion of moisture in chilly air resulting from the condensation on the air conditioning evaporator and the respiration of the operator was not taken into account.
Table 3. Values of the parameters

<table>
<thead>
<tr>
<th>Num</th>
<th>Parameter name</th>
<th>Parameter value</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Free stream speed, m/s</td>
<td>6</td>
</tr>
<tr>
<td>2</td>
<td>Cabin airflow temperature, °C</td>
<td>14</td>
</tr>
<tr>
<td>3</td>
<td>Air flow rate in cabins, m/s</td>
<td>1</td>
</tr>
<tr>
<td>4</td>
<td>Incoming flow temperature, °C</td>
<td>30</td>
</tr>
<tr>
<td>5</td>
<td>Average temperature on the human surface, °C</td>
<td>25</td>
</tr>
<tr>
<td>6</td>
<td>Mass fraction of water vapor for all domains</td>
<td>0.0091</td>
</tr>
</tbody>
</table>

Compiled by the authors.

Using the formulas for relative humidity (RH), as well as the Arden Buck formula [3], the values of relative humidity can be found. For the above reference values, the relative humidity is 34.7% for 30°C and 92.2% for 14°C.

### 3 Results

In the initial phase, an unsteady issue was addressed, where the starting temperature both inside the cabin and in the outer domain was 30°C. The water vapor mass fraction was set at 0.0091 (equivalent to 34.7% RH).

The Figure 2 shows a graph of residuals for a non-stationary problem.

Fig. 2. Residuals for a non-stationary problem

The fields were stabilized within 80 seconds. After 80 seconds, the mean temperature in the cabin was recorded at 15.7°C as shown in Figure 3. Additionally, Figure 4 displays an average humidity of 83.6% within the cabin.
Fig. 3. Average temperature in volume for non-stationary problem

Compiled by the authors.

Fig. 4. Average RH for non-stationary problem

Compiled by the authors.

After the time indicated above, all processes can be considered stabilized; a stationary setting was used.

The Figure 5 shows a graph of residuals for stationary problem.
Fig. 5. Residuals for stationary problem.

Figures 6-7 show convergence plots of the numerical method in steps of average temperatures and relative humidity values. The initial temperature values were set to 30°C, the initial humidity value was set to 34.7%.

The fields of pressures, velocities, temperatures, densities and RH are shown in the figures 8-12.

The cabin exhibited notable non-uniformity in both its internal temperature and humidity fields. The driver-facing area registered the minimum air temperature and humidity at 14°C and 92.1% each, while the walls recorded the maximum temperature of 29.3°C and humidity of 92.1%. The RH values ranged from 39.3% to 92.1%. Furthermore, a zone of air stagnation was observed in the upper right section of the cabin. The arrangement and number of deflectors, as well as the temperature and velocity of the inflowing air, can significantly impact the air flow direction, temperature, and humidity.

Hence, numerical simulation can aid in determining the optimal configurations for the deflectors to enhance air circulation within the passenger compartment.

Fig. 6. Average temperature in volume stationary problem
Fig. 7. Average RH for stationary problem

Fig. 8. a) Static pressure (relative), Pa. b) Density, kg/m3. Compiled by the authors.

Fig. 9. Vector velocity field m/s. a) inside and outside, b) inside. Compiled by the authors.
Fig. 10. Velocity m/s. a) inside and outside, b) inside. Compiled by the authors.

Fig. 11. RH, % a) inside and outside, b) inside. Compiled by the authors.

Fig. 12. Temperature, °C. a) inside and outside, b) inside. Compiled by the authors.
Figures 13-18 show the values of heat transfer coefficients (HTC) through the wall and heat flows (HF). It can be noted that the profiles of the values of the HTC determine the profiles of air velocities on the inner and outer walls.

**Fig. 13.** HTC on the left wall. Compiled by the authors.

**Fig. 14.** HF on the left wall Compiled by the authors.

**Fig. 15.** HTC on the top wall Compiled by the authors.
Fig. 16. HF on the top wall Compiled by the authors.

Fig. 17. HTC on the right wall Compiled by the authors.

Fig. 18. HF on the right wall Compiled by the authors.
4 Discussion

The air fields within the vehicle cabin were analyzed in [5], with a focus on heat transfer coefficients and heat flux through the walls. However, air humidity was not taken into account. The results were compared to existing theories and showed a 4% discrepancy in average temperature values. In [22], the analysis was expanded to include the diffusion equation and virus particle movement in the air. This study found a 3% discrepancy in average temperature values over time.

5 Conclusion

The study conducted an analysis of the air fields present within the vehicle cabin, with a specific focus on heat transfer coefficients and heat flux through the walls. The analysis included external airflow and internal ventilation, and numerical and analytical modeling were used to simulate humidity. The Navier-Stokes equations were utilized in conjunction with the diffusion equation to obtain a numerical solution, resulting in fields of velocities, temperatures, pressures, and air densities. The study also examined heat transfer coefficients and heat flows through the cabin walls, making it applicable to cabins of varying complexity, high air flow rates, and different climate equipment configurations.

The model being presented also permits the emulation of moisture levels within the passenger area, encompassing the successful calculation of when a "dew point" will materialize on the inner side of an automobile window. This enables the reduction of said occurrence by regulating the temperature, direction, and velocity of the incoming air from the transportation climate unit, as well as the moisture content.

As further studies, it is planned:
- study of accounting for air humidity on heat transfer coefficients;
- modeling of process of the increased humidity on "dew point";
- modeling of moisture condensation on the windows and internal surfaces of the cabin in an explicit form of the phase transition of moisture vapor;
- simulation of saturation with water vapor from the evaporator of the climate system;
- optimization of blowing modes and effective removal of condensate from the internal surfaces of the cab;
- taking into account the influence of internal equipment and heat release from internal equipment on the fields of temperatures and velocities;
- accounting for radiant heat transfer in the cabin, as well as heating from the radiant radiation of the sun through the glazing.

References

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