Article

Solving the Coupled Aerodynamic and Thermal Problem for Modeling the Air Distribution Devices with Perforated Plates

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Abstract: The article is focused on the comprehensive analysis of the aerodynamics of air distribution devices with the combined heat and mass exchange, with the aim to improve the following hydro- and thermodynamic parameters of ventilation systems: flow rate, air velocity, hydraulic losses, and temperature. The inadequacy of the previously obtained characteristics has confirmed the need for more rational designs of air distribution systems. Consequently, the use of perforated plates was proposed to increase hydraulic losses for reducing the average velocity and ensuring a uniform distribution of the velocity field on the outlet of the device. The effectiveness of one of the five possible designs usage is confirmed by the results of numerical simulation. The coefficient of hydraulic losses decreased by 2.5–3.0 times, as well as the uniformity of the outlet velocity field for the air flow being provided. Based on the three-factor factorial experiment, the linear mathematical model was obtained for determining the dependence of the average velocity on the flow rate, plate’s area, and diameter of holes. This model was significantly improved using the multiparameter quasi-linear regression analysis. As a result, the nonlinear mathematical models were obtained, allowing the analytical determination of the hydraulic losses and average velocity of the air flow. Additionally, the dependencies for determining the relative error of measuring the average velocity were obtained.

Keywords: aerodynamics; heat transfer; mass transfer; numerical simulation; multi-factor factorial experiment; hydraulic losses; velocity field

1. Introduction

Improving the quality of the air environment in residential and industrial premises is an urgent problem, which can be solved based on increasing the energy efficiency of the operation of ventilation and conditioning systems. Solving this problem requires, in advance, precise initial information about the dependence of the flow structure and thermodynamic parameters of the environment in ventilated premises on the operating characteristics of the air distribution system. Because the actual ventilation flows are mainly turbulent, and the geometry of the premises is rather complicated, such information data can be obtained by numerical solving the Reynolds averaged system of the Navier–Stokes partial differential equations, and the continuity equation. In conjunction with both of the corresponding equations of mass and heat transfer and boundary conditions reflecting the specific features of these ventilation systems, the above-mentioned model allows obtaining an accurate data about the distribution of velocity and temperature fields in a wide range of changes of the designs and operating modes for the projected system. Ensuring the proper microclimate parameters of the
premise is important for comfortable and productive work. For example, depending on the type of work for different categories of premises, there are strong requirements for lighting, temperature, humidity, and air velocity. Thus, in this article, the ventilation systems with air distribution devices are considered. The objective of these devices is ensuring the steady microclimate parameters and air purity at the level corresponding to the technical requirements.

Air distribution devices are important elements of ventilation and air conditioning systems. They are designed for air supply into the premises, as well as for the proper distribution of the air flow. Despite the wide variety of air distribution systems, one of the most important requirements for ensuring comfortable conditions in a given technological system is the uniformity of the air flow distribution at the outlet. Moreover, according to the requirements of the international labor organization, this flow must not exceed the velocity of 1 m/s. The air distribution devices of inflow ventilation of the “air shower” type are designed to supply large volumes of fresh air (up to $10^4$ m$^3$/h) with the air velocity in a range of 0.1–0.3 m/s for the conditioning systems in industrial and sports buildings with high requirements of comfort. They are most popular both in industrial enterprises with excessive release of heat energy and places with significant concentrations of hazardous substances. Their operation is based on creating the effect of “air oasis” in the workplace. Air distribution devices of an inflow ventilation system are usually manufactured out of sheet steel. In this case, steps and diffusers include a perforated plate, housing, lower-end panel, and upper-end panel with a jointed pipe.

Along with inflow ventilation, the local tidal ventilation systems are increasingly used. These systems create directional air flow and limited space with a certain microclimate and operating parameters. Consequently, the purpose of these systems is to maintain special conditions in a closed space bounded by the air flow zone. These conditions need to be met with the existing standards.

**Literature Review**

There are a number of studies devoted to researching the effect of perforated plates on the hydrodynamics of gas and liquid flows. Particularly, the articles [1–4] deal with the coupled thermal, structural, and vibration analysis. Traditional approaches for measuring parameters of the gas flow in air distribution devices are investigated in the articles [5–8]. These studies are mainly experimental and based on empirical approaches, which do not reflect the hydrodynamic features. As a result, the obtained analytical dependencies are properly agreed on for the proposed designs under consideration only. However, the following research should be analyzed more precisely, due to their significant role in making the research methodology.

The article [9] presents the numerical approach for the analysis of convection heat transfer from perforated plates. Based on the Reynolds averaged Navier–Stokes equations with the consequent application of the large-eddy simulation method, the numerical results were obtained. Particularly, it was found that the design with three openings had the best thermo-hydraulic performance. This research has proved that using grids in air-distribution devices is correct. However, this research does not provide a reliable mathematical model for designing related devices. Consequently, this approach should be significantly supplemented by the reliable mathematical model.

The article [10] deals with the analytical and numerical approaches of the shock tube problem in a channel with a pseudo-perforated wall. Based on the approximation procedure for a shock wave, the periodically located barriers were analyzed. As a result, the approximate analytical solution was obtained for the case of the rectangular channel with an array of grids in the wall. It was found that the leading gas-dynamic discontinuity is caused by relaxation zones. Finally, the approximate analytical solution for obtaining the flow parameters is proposed. However, despite these values corresponding with the numerical simulation results, the mathematical model is based on the simplified equations of flow motion in terms of pressure and velocities.

Ways for modeling the hydrodynamic forces on grids of perforated plates are presented in the article [11] for the case of combined steady, low frequency, and high-frequency motion. As a result, it was proved that the porous block model is capable of capturing the global large-scale wake structures,
which are responsible for the reduction in fluid flow velocity and associated forces on a structure. It allows numerical simulations for perforated elements to be carried out using porous block models. However, there are no analytical dependencies suggested for estimating the resistance coefficients of hydraulic losses agreed both for the analytical and computational models, as was realized in the paper [12]. Consequently, the related regression procedures should be implemented to overcome this weakness.

In the research work [13], semi-fluidized bed hydrodynamics have been studied experimentally. As a result, based on the empirical and semi-empirical models, the proper perforated plate was designed to have a negligible pressure drop across it. Additionally, experimental results proved the obtained hydrodynamics parameters. Another application of the effect of grid plates is presented in the research [14]. Particularly, the effect of a high porosity packing on hydrodynamics and mass transfer in bubble columns was studied experimentally. However, the proposed empirical models are not based on related physical models reflecting the hydromechanical process. Consequently, any changed designs need to be additionally investigated due to the absence of these models.

The importance of the research is highlighted by recent achievements in the field of management of heating, ventilation, and air cooling systems [15].

Due to the information mentioned above, the aim of the research is to improve air distribution devices for the inflow ventilation system. To achieve this goal, the following objectives have been set and solved:

- numerical simulation of the air flow in the air distribution devices for determining the uniformity of the velocity field and hydraulic losses in the device;
- an analysis of the obtained aerodynamic, heat, and mass exchange parameters of the operating process in the flow section of the air distribution devices;
- an improvement of the design of the air distribution devices considering the flow features by choosing the design with the most uniform velocity field and the lowest hydraulic losses;
- determination of the rational design parameters for providing a comfortable microclimate in premises;
- development of the mathematical models of aerodynamics for the air distribution devices in the case of the inflow ventilation systems with various designs and analysis of the efficiency of their operation;
- evaluation of the parameters of the proposed mathematical model by the experimental data using the regression analysis.

The research object is operating processes in a flow section of air distribution devices for the inflow ventilation system. The research subjects are hydro- and thermodynamic parameters of the air flow.

2. Materials and Methods

For determination of the rational design parameters of the air distribution device, the following five design models are proposed:

(a) without directing elements inside the panel (Figure 1a);
(b) with a directing element as a perforated cone for improving the uniformity of the velocity field in the entire area of the panel (Figure 1b);
(c) with two directing elements as perforated cones of different diameters (Figure 1c);
(d) with a conical outlet with the angle 45° (Figure 1d);
(e) with a conical outlet and two directing elements as perforated cones of different diameters for improving the uniformity of the velocity field in the entire area of the panel (Figure 1e).
A numerical experiment on the modeling of coupled aerodynamics and thermal analysis through the system of inflow ventilation was carried out using the “ANSYS CFX” software [16,17]. For this purpose, three-dimensional models of different designs were created using the “SolidWorks” software.

In the stated problem, the calculated space consists of the internal volumes of the air distribution device and the premise with the moving air (Figure 2). The considered air distribution device is the last element of the technological system of an inflow ventilation system. This system generally consists of the outdoor air intake, cleaning system, compressor, heat exchanger, collector, and the system of air distribution pipelines. The perforated plate from the stainless steel X10CrNiTi18-9 represents a cylindrical surface with a thickness of 4 mm and the array of stamped holes with a diameter of 8 mm.

For simplification of the numerical simulation procedure, volumes of the premise and the air distribution device were separated. The calculation mesh is built only for various panel designs. The mesh for the room has remained unchanged. The transfer of thermodynamic parameters between the estimated grids occurs on the interface surface.

![Figure 2. Design elements of the calculation model.](image-url)
The boundary conditions have to be considered for the mathematical certainty of the stated problem. They include the following features:

- geometrical conditions characterizing the shape and dimensions of the calculation area (two separate domains);
- physical conditions describing properties of the environment (density, viscosity, thermal conductivity);
- time conditions forming the features of the process in time (for the case of the unsteady process);
- boundary conditions for determining parameters of the process on boundaries of the calculated area: inlet (flow rate and constant temperature), outlet using the “indoor” model (the overpressure is 0.1 MPa), walls using the “wall” model (zero velocity and constant temperature), and “symmetry”.

During CFD modeling, the air distribution device was bounded by a cylindrical surface with a diameter of 800 mm and a height of 1500 mm. The width and height of the premises are 6 m and 4 m, correspondingly. For modeling air flow in the distribution device, an unstructured mesh was used with the total number of elements in a range of $3\times10^6$ depending on the variant of the device. For modeling the air flow in the premises, a hexahedral structured mesh with $2.6\times10^5$ elements was used. The transfer of thermodynamic parameters between two separate domains was carried out through the vertical interface surface.

To solve the coupled thermal and aerodynamic problem, a heat transfer model was used as the “thermal energy” setup, including the system of variational equations of continuity, momentum, and total energy. The model of air motion in calculated volumes is described by the continuity and Navier–Stokes equations averaged by the Reynolds number. During the calculations, a model of gravity was used, which allows consideration of the process of free convection in a closed volume under the mass (volume) forces. The $k$-$\varepsilon$ model of turbulence was used, and the convergence criterion was $10^{-4}$.

3. Results

3.1. Numerical Simulation Results

Using the “ANSYS CFX” software, numerical simulations conducted for obtaining the air velocity distribution at the outlet in height are presented in Figure 3 for the different operating modes (the flow rates are 3000, 4000, 5000, 6000, and 7000 m$^3$/h). The results of numerical simulations as the distributions of air velocity at the outlet of the air distribution device in height under the different operating modes are presented in Table 1. The values of the velocity field were measured 10 mm after the interface surface.

The analysis of Table 1 for the design model (a) shows the inefficiency of this model because the cooled air is removed from the pipe and moved down to the bottom panel, and only then begins to flow through the perforated panel into the premise, whereas it should be uniformly distributed by the panel’s height. Moreover, a sharp gap in flow at the outlet creates significant vortices that lead to an increase in hydraulic losses in the system.

When the directing elements as the perforated cones are installed inside the panel (design models (b) and (c)), the situation improves due to the formation of the vortex structures inside the air distribution device. This fact allows for obtaining the uniform air flow at the outlet (Figure 3). But significant vortices between the directing elements and the upper-end surface lead to the air entrainment inside the device. These vortices both increase the hydraulic losses in the system and block the part of the perforated surface near the upper-end wall. It should be noted that an increase in the flow rate leads to an increase in the vortex size and blocking zone (Figure 4).
Table 1. Velocity distribution by the panel height for the different designs and flow rates.

| Design Flow Rate [×10³ m³/h] | 3   | 4   | 5   | 6   | 7   |
|------------------------------|-----|-----|-----|-----|-----|
|                              |     |     |     |     |     |
| (a)                          |     |     |     |     |     |
| (b)                          |     |     |     |     |     |
| (c)                          |     |     |     |     |     |
| (d)                          |     |     |     |     |     |
| (e)                          |     |     |     |     |     |

Velocity distribution by the panel height for the different designs and flow rates.
Figure 3. Velocity distribution in the premises with the air distribution system.

To reduce the negative effects of the vortices, making the inlet pipe into the air distribution device as a cone with the angle 4° (the design model (d)) was proposed. The analysis of Table 1 shows that this decision minimizes the negative impact of vortices in the space between the directing elements and the upper face wall, as well as reducing hydraulic losses of the system (Table 2).

Table 1 allows comparing the velocity fields at the outlet for various designs and operating modes of the air distribution device. Particularly, the velocity field is the least uniform for the variant (b), and the most uniform for the variant (e).

To reduce the negative effects of the vortices, making the inlet pipe into the air distribution device as a cone with the angle 4° (the design model (d)) was proposed. The analysis of Table 1 shows that
this decision minimizes the negative impact of vortices in the space between the directing elements and the upper face wall, as well as reducing hydraulic losses of the system (Table 2). Hydraulic losses are measured as the pressure difference at the inlet and the grid surface divided by the volumetric weight of the air.

Table 2. Hydraulic losses [m] for the different designs and flow rates.

| Design | Flow Rate [$\times 10^3$ m$^3$/h] |
|--------|----------------------------------|
|        | 3  | 4  | 5  | 6  | 7  |
| (a)    | 0.51 | 0.90 | 1.40 | 2.01 | 2.72 |
| (b)    | 0.55 | 0.94 | 1.47 | 2.08 | 2.81 |
| (c)    | 0.56 | 0.94 | 1.54 | 2.13 | 2.92 |
| (d)    | 0.17 | 0.37 | 0.56 | 0.78 | 1.09 |
| (e)    | 0.19 | 0.40 | 0.59 | 0.83 | 1.14 |

Considering the previously obtained results, the design model (e) was proposed. There are two directing elements placed inside the panel as the perforated cones of different diameters. The inlet of the air distribution system is manufactured as a cone with the diffuser angle 4°. Figures 4 and 5 show that the proposed design model of the air distribution device allows obtaining the uniform distribution of the air velocity in the panel’s height. As a result, this design has relatively low hydraulic losses of the system. However, additional attention should be paid to the vortices and the stagnant zone between the directing elements.

The abovementioned justification for choosing the design model (e) as the rational one is additionally proved by the analysis of temperature fields (Figure 5).

Figure 5. Temperature distribution in the premises with the air distribution system.

Additionally, Table 2 shows that an increase of the directing elements’ size in a housing of the air distribution device leads to an increase of the hydraulic losses of the system. At the same time, using the inlet diffuser decreases the hydraulic losses by 2.5–3.0 times.
3.2. Multi-Factor Factorial Experiments

The main parameter of the study is the average air velocity [18], because this parameter is standardized and determines the comfortable conditions in premises. Moreover, low outlet velocity of the air flow (despite the significant flow rate) is the main advantage of these ventilation systems. These are the factors of the experiment: flow rate, plate's area, and diameter of holes.

The levels and intervals for variation of factors, as well as the matrix of planning a three-factor experiment, are presented in Tables 3 and 4, respectively.

Table 3. Levels and intervals for variation of factors.

| Parameter                  | Diameter of Holes D [mm] | Flow Rate Q [×10³ m³/h] | Perforated Plate’s Area S [m²] |
|----------------------------|--------------------------|--------------------------|-------------------------------|
| Basic level                | 8                        | 5                        | 0.25                          |
| Variation interval         | 3                        | 2                        | 0.15                          |
| Upper level                | 11                       | 7                        | 0.4                           |
| Lower level                | 5                        | 3                        | 0.1                           |

Table 4. The matrix of planning a three-factor experiment.

| Experiment No. | Holes Diameter D [mm] | Flow Rate Q [×10³ m³/h] | Area of a Plate S [m²] | Average Velocity V [m/s] |
|----------------|-----------------------|--------------------------|------------------------|--------------------------|
| 1              | 5                     | 7                        | 0.1                    | 7.70                     |
| 2              | 11                    | 7                        | 0.1                    | 8.23                     |
| 3              | 5                     | 3                        | 0.1                    | 1.38                     |
| 4              | 11                    | 3                        | 0.4                    | 2.34                     |
| 5              | 5                     | 7                        | 0.4                    | 7.50                     |
| 6              | 11                    | 7                        | 0.4                    | 8.11                     |
| 7              | 5                     | 3                        | 0.4                    | 1.20                     |
| 8              | 11                    | 3                        | 0.4                    | 1.87                     |

It should be noted that the values’ average velocities were obtained numerically at 10 mm close to the outlet. The traditional factorial experiment realized according to the methodology, described in detail within the works [19,20], shows that the average velocity as the ratio of the flow rate to the cross-sectional area can be approximately determined by the following linear regression dependence:

\[ V = a_0 + a_1 D + a_2 Q + a_3 S, \]  

where \( D \) is diameter of holes [mm], \( Q \) is flow rate [×10³ m³/h], \( S \) is area of a perforated plate [m²], and \( V \) is average velocity [m/s].

The coefficients \( a_0, a_1, a_2, \) and \( a_3 \) are determined by ensuring the minimum of the error function:

\[ R_0([A]) = \sum_{i=1}^{N} (a_0 + a_1 D_i + a_2 Q_i + a_3 S_i - V_i)^2, \]  

where \( D_i, Q_i, S_i, \) and \( V_i \) are \( i \)-th values of the experimental data, and \( N = 8 \) is the total number of the experimental dataset.

Finally, the system of four linear equations

\[ \frac{\partial R_0([A])}{\partial a_j} = 2 \sum_{i=1}^{N} \left( \frac{\partial (a_0 + a_1 D_i + a_2 Q_i + a_3 S_i - V_i)}{\partial a_j} \right) \left( a_0 + a_1 D_i + a_2 Q_i + a_3 S_i - V_i \right) = 0 \]  

for \( j = 0, 1, 2, 3 \) allows determining the unknown parameters \( a_0, a_1, a_2, \) and \( a_3 \) as elements of the following vector:
where quasi-linear regression analysis [21,22] was proposed, using the following analytical dependence:

\[
\frac{Q}{S^n D^{2(1-n)}} = C \quad (6)
\]

where \(C, n\) are unknown dimensionless parameters subjected to further evaluation.

This dependence corresponds to the physical features of the studied problem because the air velocity is directly proportional to the flow rate and inversely proportional to the cross-sectional area of the channel. Moreover, formula (6) meets all the requirements of the theory of dimensions [23]:
terms $S^n$ and $D^{2(1-n)}$ allow obtaining the correct dimension of the average velocity (m/s) regardless of the power $n$.

Due to the fundamentals of the least square method [24], the unknown parameters $C$ and $n$ are determined by the data from the matrix of planning a three-factor experiment (Table 4) using the procedure of minimizing the following error function:

$$R_V(C, n) = \sum_{i=1}^{N} \left[ \ln C + n \ln \frac{D_i^2}{S_i} - \ln \frac{V_i^2 D_i^2}{Q_i^2} \right] \rightarrow \text{min.}$$  \hspace{1cm} (7)$$

The system of equations for evaluation of the parameters $C$ and $n$ has the form:

$$\begin{cases} \frac{\partial R_V}{\partial \ln C} = 2 \sum_{i=1}^{N} \left[ \ln C + n \ln \frac{D_i^2}{S_i} - \ln \frac{V_i^2 D_i^2}{Q_i^2} \right] = 0; \\ \frac{\partial R_V}{\partial n} = 2 \sum_{i=1}^{N} \left[ \left( \ln C + n \ln \frac{D_i^2}{S_i} - \ln \frac{V_i^2 D_i^2}{Q_i^2} \right) \ln \frac{D_i^2}{S_i} \right] = 0. \end{cases}$$  \hspace{1cm} (8)$$

The use of identity transformations allows rewriting this system in the following matrix form:

$$\begin{pmatrix} N \\ \sum_{i=1}^{N} \ln \left( \frac{D_i^2}{S_i} \right) \\ \sum_{i=1}^{N} \ln^2 \left( \frac{D_i^2}{S_i} \right) \end{pmatrix} \begin{pmatrix} \ln C \\ n \end{pmatrix} = \begin{pmatrix} \sum_{i=1}^{N} \ln \left( \frac{V_i^2 D_i^2}{Q_i^2} \right) \\ \sum_{i=1}^{N} \ln \left( \frac{V_i^2 D_i^2}{Q_i^2} \right) \ln \left( \frac{D_i^2}{S_i} \right) \end{pmatrix}.$$  \hspace{1cm} (9)$$

Thus, the two-parameter quasi-linear regression dependences for evaluation of the above-mentioned parameters take the forms:

$$C = \exp \left[ \sum_{i=1}^{N} \ln \left( \frac{D_i^2}{S_i} \right) - \sum_{i=1}^{N} \left( \ln \left( \frac{V_i^2 D_i^2}{Q_i^2} \right) \ln \left( \frac{D_i^2}{S_i} \right) \right) \right];$$  \hspace{1cm} (10)$$

$$n = \frac{\sum_{i=1}^{N} \ln \left( \frac{D_i^2}{S_i} \right) \cdot \sum_{i=1}^{N} \ln \left( \frac{V_i^2 D_i^2}{Q_i^2} \right) - \sum_{i=1}^{N} \ln \left( \frac{V_i^2 D_i^2}{Q_i^2} \right) \cdot \sum_{i=1}^{N} \ln \left( \frac{D_i^2}{S_i} \right)}{\left[ \sum_{i=1}^{N} \ln \left( \frac{D_i^2}{S_i} \right) \right]^2 - N \cdot \sum_{i=1}^{N} \ln^2 \left( \frac{D_i^2}{S_i} \right)}.$$  \hspace{1cm} (11)$$

Particularly, for the above-mentioned data (Table 4), the numerical calculations allow determining the values $C = 0.87$ and $n = 0.70$.

The resulting spatial surfaces reflecting the dependence of the average velocity $V$ on the diameter $D$, flow rate $Q$ and area $S$ are presented in Figure 7.

In this case, the relative error $\delta_V$ of determining the average velocity can be determined by the following formula [25,26]:

$$\delta_V = \frac{1}{V} \sqrt{\left( \frac{D \partial V}{\partial D} \right)^2 \delta_D^2 + \left( \frac{Q \partial V}{\partial Q} \right)^2 \delta_Q^2 + \left( \frac{S \partial V}{\partial S} \right)^2 \delta_S^2},$$  \hspace{1cm} (12)$$

where $\delta_D$, $\delta_Q$, and $\delta_S$ are relative errors of measuring the diameter, flow rate, and area, respectively.

Using the dependence (6), formula (12) can be rewritten in the identical simplified form:

$$\delta_V = \sqrt{4(1-n)^2 \delta_D^2 + \delta_Q^2 + n^2 \delta_S^2}.$$  \hspace{1cm} (13)
It should be also noted that in the case when the average velocity \( V \) weakly depends on the area \( S \), the parameter \( n \) is equal to zero and analytical dependence (6) is simplified:

\[
V = C_0 \frac{Q}{D^2},
\]

(14)

The use of a single-parameter quasi-linear regression procedure [27] allows determining the constant \( C_0 \):

\[
C_0 = \frac{\sum_{i=1}^{N} Q_i V_i D_i^2}{\sum_{i=1}^{N} Q_i^2},
\]

(15)

In this case, the relative error (13) takes the following form:

\[
\delta_V = \sqrt{4\delta_D^2 + \delta_Q^2}.
\]

(16)

The relative error of the diameter corresponds to the following factors: the technological peculiarities of the process of stamping holes, dimensional and machining. Particularly, for diameters \( \varnothing 8H9 \), the maximum dimensional deviation is 0.036 mm, and the relative error is about 0.5%. The relative error of the flow rate is mainly related to the measurement accuracy. As a rule, the error of measuring the air flow is in a range of 5–7%, depending on the installation accuracy.

Figure 7. The nonlinear dependence of the average velocity on the diameter, flow rate, and area.

3.3.2. Hydraulic Losses

Application of the regression analysis additionally allows evaluating the unknown parameters of the following analytical dependence for determining the hydraulic losses [28,29]:

\[
\Delta h = \frac{\Delta p}{\gamma} = \frac{V^2}{2g} = \alpha Q^2,
\]

(17)

where \( \Delta h \) is hydraulic losses [m], \( \Delta p \) is pressure difference along the air distribution device [Pa], \( \gamma \) is specific weight of the environment [N/m\(^3\)], \( \zeta \) is dimensionless coefficient of the hydraulic losses [-], \( g = 9.81 \text{ m/s}^2 \) is gravitational acceleration [m/s\(^2\)], \( \alpha = \zeta/(2gS) \) is modified coefficient of the hydraulic losses [s\(^2\)/m\(^3\)].

This dependence is an identical modification of the Darcy–Weisbach equation [30], widely used for solving the applied problems in the field of hydroaeromechanics.
The unknown parameter $\alpha$ can be determined by minimizing the following error function:

$$R_{\Delta h}(\alpha) = \sum_{i=1}^{N} (\alpha Q_i^2 - \Delta h_i)^2 \rightarrow \text{min},$$  

where $Q_i$, $\Delta h_i$ are $i$-th values of the experimental data from Table 4.

The equation for evaluation of the parameter $\alpha$:

$$\frac{dR_{\Delta h}}{d\alpha} = 2 \sum_{i=1}^{N} \left[ (\alpha Q_i^2 - \Delta h_i) Q_i^2 \right] = 0$$  

and after its identity transformation it takes the following form:

$$\alpha = \frac{\sum_{i=1}^{N} Q_i^2 \Delta h_i}{\sum_{i=1}^{N} Q_i^4}.$$  

The evaluated values of the parameter $\alpha$ for the different design schemes of the air distribution device are 0.056, 0.058, and 0.060 for designs (a), (b), and (c), and 0.022 and 0.23 for designs (d) and (e), respectively. The related approximating curves are presented in Figure 8.

![Figure 8](image_url)

**Figure 8.** The dependence of the hydraulic losses on the flow rate for the different design models of the air distribution system.

Consequently, the improved designs of the air distribution devices have the coefficient of hydraulic losses up to 2.5–3.0 times lower than the traditional design schemes.

4. Discussion

Numerical simulations of aerodynamic, heat, and mass transfer processes in premises with air distribution devices of the inflow ventilation system have been carried out using the “ANSYS CFX” software. As a result of the calculations, the distributions of velocity and temperature fields were obtained both for the air distribution device and in a premise. Additionally, heat flux on the design surfaces was determined, and the presence of vortices and air stagnation zones was established.

To unify the methods of comprehensive calculation of air distribution devices, methodologies for analytical determination of average velocity and hydraulic losses were also developed. Research works implemented according to the methodology of the multifactorial experiment have allowed evaluation of the obtained dependences and the related spatial surface for determining the average velocity in the air distribution device. The multi-parameter quasi-linear regression analysis was
realized for determining the average velocity and hydraulic losses. As a result, the mathematical model was proposed based on the obtained regression dependencies. Unknown coefficients of the mathematical model were evaluated based on the experimental results data using the presented analytical dependencies. Additionally, the formula for determining the relative error of calculating the average velocity was obtained.

It should be additionally noted that the proposed approach has significant prospects for further development under the conditions of its comprehensive implementation with the systems of computational intelligence, particularly to identify the parameters of the above-mentioned mathematical models and the related regression dependencies with the use of artificial neural networks, as was previously realized within the research works [31–34]. Moreover, in further research, special attention should be paid to demonstrate benefits of the proposed designs of the air distribution device in terms of the air diffusion performance index, using the approach presented in the research [35].

5. Conclusions

As a result of the research, numerical simulations for various designs of the air distribution systems have allowed obtaining of the rational design parameters, ensuring the requirements for comfortable microclimate parameters in premises. Particularly, it was justified that the improved designs (d) and (e) have the coefficient of hydraulic losses 2.5–3.0 times lower in comparison with the traditional designs. The above-mentioned information has allowed improvement for the air distribution device for the inflow ventilation system.

The reliable mathematical model of aerodynamics for the air distribution devices, in the case of the inflow ventilation systems with various designs, was developed based both on linear and quasi-linear regression procedures. The proposed approach allows determining the parameters of the model by the experimental data. Particularly, formulas (6) and (17) allow determining the average velocity and hydraulic losses in the air distribution device using the regression dependencies (11) and (20) for evaluating the unknown parameters.

The practical results of the research and the related design with the obtained rational parameters are implemented into the practice of designing air distribution devices at the machine-building enterprises of Sumy and Kharkiv, regions of Ukraine, as well as for ensuring the rational air distribution at the enterprises of the “Guala closures group” (Italy, Ukraine).

The obtained results will be useful for researchers and specialists in the fields of aerodynamics, thermal energy, and energy-efficient technologies, particularly to assess compliance with comfortable conditions in premises, to analyze their thermal state, as well as to evaluate the efficiency of different means of energy saving.

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