Finite element simulation of airflow in a field cleaner

I A Panfilov¹, A N Soloviev¹,², A A Matrosov¹, B Ch Meskhi¹, O O Polushkin¹, D V Rudoy¹, and V I Pakhomov¹,²

¹Don State Technical University, Gagarin sq., 1, Rostov-on-Don, 344003, Russia
²Agrarian Research Center "Donskoy", Zernograd, Russia
E-mail: ³solovievarc@gmail.com

Abstract. The work is devoted to the study of the dynamics of the grain mass in a stripping field installation and a stationary installation for separating grain. At the first stage, a model of the air mass movement in the installation under consideration is constructed and, on the basis of a mathematical model that takes into account the turbulence of the air mass movement, using the finite volume method in the ANSYS package, the velocity and pressure field is calculated. The influence of the technological parameters of the installations on these fields is studied: the diameters and rotation speed of the stripping drum and the additional drum-fan, the shape of the surface of the upper deck of the installation chamber. The calculations performed allow us to choose rational geometric and kinematic parameters at which no stagnant zones arise in the chamber.

1. Introduction
The urgent task of today in the grain sector of agriculture is the task of ensuring low-energy and low-traumatic forced extraction of grain from the ear. This requires not only the improvement of existing methods and methods of grain harvesting [1-7], but also the development of new highly effective physical and mechanical methods of its harvesting and, accordingly, the design of appropriate equipment [8-12].

The solution of this problem, on the one hand, requires the development and creation of adequate analytical and numerical physical and mechanical models and models based on CAD-CAE complexes capable of describing the interactions between the elements of the stem-ear-grain system, and at the same time, on the other hand physical and mechanical models of the process of impact on the stem-ear-grain system of the working bodies of grain harvesting equipment.

In the previous work [12], the modes of action on the ear were studied, in which it is possible to extract grain from it due to the resonant movements of the ear and grain in the ear. The present work is devoted to the study of the dynamics of air mass in a field installation that implements stripping harvesting of ears with further separation of grain and chaff.

The study of the movement of the grain mass can be divided into two steps using an iterative scheme for finding rational parameters. At the first stage, a model of air mass movement in the considered installations is built and based on a mathematical model that takes into account the turbulence of the movement using the finite volume method, the velocity and pressure field is calculated. At the second stage, the movement of a fragment of the grain mass in the chambers of the installations is considered and the trajectory of its movement is studied based on the integration of the differential equations of the dynamics of the point. This paper considers the first stage of the study.
A series of calculations was carried out; the geometric and kinematic parameters of the installations were obtained.

2. Mathematical formulation of the problem of air mass movement in installations

When using the $\kappa$-$\varepsilon$-model of turbulence [17-23], the mathematical model of the medium flow has the following form:

$$\frac{\partial \rho V_x}{\partial t} + \frac{\partial \rho V_x V_x}{\partial x} + \frac{\partial \rho V_x V_y}{\partial y} = - \frac{\partial P}{\partial x} + \frac{\partial \rho}{\partial x} \left( \mu_e \frac{\partial V_x}{\partial x} + \mu_t \frac{\partial V_x}{\partial y} \right),$$

$$\frac{\partial (\rho V_x V_y)}{\partial x} + \frac{\partial (\rho V_y V_y)}{\partial y} = - \frac{\partial P}{\partial y} + \frac{\partial \rho}{\partial y} \left( \mu_e \frac{\partial V_y}{\partial x} + \mu_t \frac{\partial V_y}{\partial y} \right),$$

$$\frac{\partial (\rho V_y)}{\partial x} + \frac{\partial (\rho V_y)}{\partial y} = \frac{\partial}{\partial x} \left( \mu_e \frac{\partial V_y}{\partial x} \right) + \frac{\partial}{\partial y} \left( \mu_t \frac{\partial V_y}{\partial y} \right) + \mu_t \Phi R e,$$

where

- $\rho$ - target air density ($\rho=\text{const}$), $V_x, V_y$ - fluid velocity vector projection, $P$ - fluid pressure,
- $\mu$ - given physical (laminar viscosity) ($\mu=\text{const}$), $\mu_t$ - turbulent (vortex) viscosity, $\mu_e$ - effective viscosity,
- $K$ - kinetic energy of turbulence per unit mass of fluid, $\varepsilon$ - the rate of dissipation in heat of the kinetic energy of turbulence per unit volume of fluid,
- $C_\mu, C_{1e}, C_2, \sigma_e$ - empirical constants of the $\kappa$-$\varepsilon$-model,
- $C_\mu = 0.09, C_{1e} = 1.44, C_2 = 1.92, \sigma_e = 1.3.$

Continuity equation (1), motion (2) and $\kappa$-$\varepsilon$-model equation (3) are written in a conservative form; unknowns are functions $V_x, V_y, \rho, \kappa$ and $\varepsilon$.

The boundary conditions for the system of transport equations (1) - (3) can be:

- speed in case of speed setting at the border
- pressure. In the case of a free boundary, the pressure 0 is set relative to the external pressure.
- no-slip conditions on the inner walls, the velocities are 0.

To obtain the fields of air velocities in the body of a field machine, this problem was solved numerically by the method of finite volumes [17].

To implement the numerical-analytical algorithm, the following conditions were set boundary conditions – Figure 1:

1. In section 1-3, air is drawn in, the pressure is 0 Pa. Here and below, the pressure is specified as the difference between atmospheric pressure.
2. It section 4-6, the air flow is released, the pressure is 0 Pa.
3. In section 7, 8-13, the boundary conditions of the housing wall are set, the conditions for air adhesion - the flow velocity are equal to 0 m/s.
4. The speed of rotation of the stripping drum is set. The original speed is 640 rpm clockwise in the original coordinate system.
5. The rotation speed of the beater is set. The original speed is 2100 rpm clockwise in the original coordinate system.

The drum and beater speeds have also been parameterized. The physical parameters of the air were set:

- Density – 1.225 kg/m³,
- Viscosity – 1.79 $10^{-5}$ kg/(m·s),
- Acceleration of gravity 9.81 m/s².

Numerical-analytical algorithm was implemented using two transport equations for turbulent characteristics (k-ε model) [18].

To obtain the fields of air velocities in the housing of the final threshing device, this problem was solved numerically by the method of finite volumes [17].

To implement the numerical-analytical algorithm, the following conditions were set, boundary conditions:

1. Fan speed in rpm is set.
2. In section 1 (Figure 2), air is drawn in, the pressure is 0 Pa. Here and below, the pressure is specified as the difference between atmospheric pressure.
3. In section 3, Figure 2, the air flow is released, the pressure is 0 Pa.
4. In section 2 (Figure 2), the air flow rate is set to 20 m/s.
5. Sticking conditions are set on the other walls of the casing, the casing and the walls of the fan.

All geometrical dimensions, drum rotation speed and air flow drawing speed were parameterized to make it possible to study the dependence of the sought parameters on these characteristics.
3. Calculation of CAE in complex airflow in field installation

3.1. Field installation
Figures 2, 3 show a finite-volume partition of the internal air domain of a field machine.

![Figure 2. Finite-volume partitioning.](image)

![Figure 3. Finite-volume partitioning (zoomed).](image)

3.2. Calculation with standard parameters
To obtain the distribution fields of flow rates in the body of the field machine, a series of calculations was carried out at different geometric parameters and drum rotation speeds.

Figures 4 - 8 show the contours of the velocity fields and the velocity vectors for the initial geometric parameters (Figure 1). The rotation speeds of the drum and beater are 640 rpm and 2100 rpm, respectively. The maximum design flow velocity was 36.2 m/s. On the left side of the drum, an
increase in the flow rate is observed, due to the narrowing of the space between the drum and the housing wall.

A slight discrepancy in the display of velocities in the graphs of displaying fields and vectors is due to the peculiarities of displaying and visualizing numerical results.

**Figure 4.** Velocity contours for standard sizes.

**Figure 5.** Velocity vector contours for standard dimensions.
Figure 6. Velocity vector contours for standard dimensions (enlarged).

Figure 7. Velocity vector contours for standard dimensions in the X direction.
Figure 8. Velocity vector contours for standard dimensions in the Y direction.

3.3. Optimizing body geometry
An optimization calculation was carried out to assess the effect of the hull geometry. The original body geometry (see Figure 1) has been changed by moving the top down 70 mm - Figure 9.

Figure 9. Modified geometry of the original field machine body.

Figures 10 - 12 show the contours of the velocity fields and the velocity vectors for the modified housing shown in Figure 9. The rotation speeds of the drum and beater are 640 rpm and 2100 rpm, respectively. The maximum design flow velocity was 31.2 m/s.

For this geometry, a more even distribution of the air velocity fields is observed.
Figure 10. Velocity contours for modified body geometry.

Figure 11. Velocity field contours for the modified body geometry along the X-axis.
3.4. Study of the behavior of flow velocities on the diameter of the stripping drum

Table 1 and the graph (Figure 14) show the calculations of the maximum speed of the velocity fields at fixed drum and beater revolutions 640 rpm and 2100 rpm, respectively, and different values of the drum diameter.

| First wheel diameter, mm | Maximum flow rate, m/s |
|--------------------------|------------------------|
| 350                      | 23.9                   |
| 450                      | 23.90                  |
| 550                      | 24                     |
| 650                      | 31.4                   |
| 700                      | 36.2                   |

Figure 12. Velocity contours for modified body geometry along the Y-axis.

Figure 13. Dependence of the maximum airflow rate in the field machine on the drum diameter.
At drum diameters less than 550 mm, the maximum speed values are preserved, since the maximum flow rates in these cases are already set by the beater and remain unchanged.

Figures 14-18 show the contours of the velocity fields and the velocity vectors for different drum diameters. The rotation speeds of the drum and beater are 640 rpm and 2100 rpm, respectively.

**Figure 14.** Velocity field contours for drum diameter 350 mm.

**Figure 15.** Velocity field contours for a drum diameter of 450 mm.
Figure 16. Velocity field contours for drum diameter 550 mm.

Figure 17. Velocity field contours for drum diameter 650 mm.
3.5. Investigation of the behavior of flow rates on the speed of rotation of the stripping drum

Table 2 and the graph (Figure 19) show the calculations of the maximum speed of the velocity fields for a fixed drum diameter of 650 mm and different values of the drum rotation speeds. The diameter and rotation speed of the beater are equal to the initial values. [22]

Table 2. Dependence of the maximum flow rate on the drum rotation speed.

| Drum rotation speed, rpm | Maximum flow rate, m/s |
|--------------------------|------------------------|
| 500                      | 26.7                   |
| 600                      | 30.5                   |
| 700                      | 32.8                   |
| 800                      | 35.8                   |
| 900                      | 40.1                   |
| 1000                     | 49.9                   |

The value of the maximum flow rate from the speed of rotation of the drum

Figure 19. Dependence of the maximum flow rate on the drum rotation speed.
Figures 20-25 show the calculations of the velocity fields for a fixed drum diameter of 650 mm and different values of the drum rotation speeds. The diameter and rotation speed of the beater are equal to the original values.

**Figure 20.** Speed field contours for 500 rpm.

**Figure 21.** Speed field contours for 600 rpm.
Figure 22. Speed field contours for 700 rpm.

Figure 23. Speed field contours for 800 rpm.
4. Conclusion

The first stage of the iterative algorithm for solving the problem of optimizing the kinematic and geometric characteristics of the field stripper and stationary installations has been developed. The task of studying the movement of the grain mass is divided into two stages. At the first stage, a model of air mass movement in the considered installations is built and based on a mathematical model that takes into account the turbulence of the movement using the finite volume method; the field of velocities and pressures is calculated. The influence of the technological parameters of the installations on these
fields was studied: the diameters and rotation speed of the stripping drum and the additional drum-fan, the shape of the surface of the upper deck of the installation chamber. The calculations have shown that additional field experiments are required to build an adequate model of the stationary installation. A series of calculations was carried out according to the developed algorithm; the geometric and kinematic parameters of the installations were obtained.

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