https://doi.org/10.33271/nvngu/2024-2/113

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BASICS OF CALCULATION OF A TWO-CIRCUIT AIR PURIFICATION SYSTEM FOR POLYDISPERSE DUST

Purpose. To increase the level of environmental safety of enterprises by improving the quality of air purification from polydisperse dust in two-circuit closed systems, in particular, to obtain the basic design relations for the engineering calculation of such systems.

Methodology. The aim of the study was realised by mathematical and numerical modelling of hydrodynamic processes in the elements of a closed double-circuit purification system.

Findings. A methodology for hydraulic calculation of a closed two-circuit cleaning system was proposed by drawing up a pressure balance of individual circuits, and a dependence for the complex coefficient of hydraulic losses of the collection-return apparatus was determined.

Originality. The hydraulic calculation of two-circuit closed cleaning systems is proposed to be carried out by compiling the pressure balance of individual circuits. To calculate a specific element of the system – the collection-return apparatus – the concept of a complex hydraulic loss coefficient is introduced, which takes into account both local pressure losses and losses along the length, and also indirectly reflects the effect of flow swirl on the hydraulic resistance of this element. For the complex coefficient of hydraulic losses, the quantitative results necessary for engineering calculations were obtained by numerical modelling of hydrodynamic processes of the swirling flow in an annular pressure channel.

Practical value. The obtained results make it possible to design two-circuit closed cleaning systems for different production conditions, which, in turn, makes it possible to replace typical and inefficient direct-flow systems with a system in which the efficiency of polydisperse dust capture is significantly increased due to separate cleaning.

Keywords: *environmental safety, polydisperse dust, cleaning system, hydraulic losses, pressure balance*

Introduction. The issue of improving the level of environmental safety of urbanised areas of Ukraine remains quite relevant. This is mainly due to the fact that a significant number of relatively small enterprises are located within and in close proximity to territorial communities, for example, construction or chemical materials production facilities, woodworking and furniture workshops, beneficiation enterprises, bulk materials handling and packaging workshops, etc. Technological processes in these facilities are accompanied by the release of a certain amount of polydisperse dust into the air of the premises. Sometimes the physical or chemical characteristics of the dust mass are such that it is absolutely necessary to limit its release into the environment as much as possible; sometimes the dust mass is a useful substance and it is also economically unfeasible to discharge it into the environment. The Law of Ukraine "On Environmental Protection" stipulates that enterprises are obliged to ensure the environmental safety of people and comply with the standards of harmful impacts on the environment. To this end, enterprises must be equipped with facilities, equipment and devices for the treatment of emissions and discharges. As a rule, such facilities use direct-flow systems for emissions treatment, with the simplest inertial devices, cyclones, performing the function of trapping. Since the efficiency of inertial catchers is low, the release of dust mass into the atmosphere can be quite significant. On the one hand, such systems do not meet modern environmental safety standards, and on the other hand, small enterprises do not upgrade their cleaning equipment due to financial circumstances and the Ukrainian environmental legislation, which is quite loyal to producers. It is expected that the terms (amounts) of payment for emissions will change as Ukraine's regulatory framework approaches the requirements of the European Union.

Literature review. The environmental safety of atmospheric air depends largely on the efficiency of air purification systems at industrial enterprises. In Ukraine, dry and wet inertial catchers, electrostatic precipitators, bag filters, and in some cases even gravity catchers are common. These devices are used in conventional direct-flow systems. More sophisticated ones, such as hybrid ones for thermal power plants, are still under development and it is impossible to predict the timing of their implementation [1, 2]. In the metallurgical industry, the cleaning systems are more complex than in the thermal power industry, but essentially, they remain direct-flow systems as well [3, 4].

The main disadvantage of inertial dust collectors is the low efficiency of cleaning the flow from the fine dust component of the dust mass. Electrostatic precipitators almost do not have this disadvantage, but their cost of cleaning is very high. In addition, the use of electrostatic precipitators is also limited by certain requirements for the electromagnetic properties of dust and requires appropriate personnel qualifications [5, 6].

The latest research in the field of increasing the efficiency of inertial devices is aimed at improving the capture of fine fractions. Since the constructive methods of solving this problem for the most common catchers have been practically exhausted, some studies are devoted to finding ways to influence the fractional composition of dust, for example, by acoustic coagulation [7, 8], in particular, using ultrasound [9, 10]. This method gives good results, but requires additional equipment, the cost of which can be several times higher than the cost of the cleaning equipment itself [11]. In addition, this equipment requires the appropriate qualifications of the operating personnel, which will increase the current costs of maintaining the treatment plant and, ultimately, the cost of treatment.

Another way to capture fine dust is to develop and use combined cleaning systems. Thus, in [12, 13], a two-stage design is proposed, the first element of which is a coagulator with ultrasonic disc emitters, and the second is a trap in the form of an apparatus with counter swirling flows. In [14], a two- or three-stage system is used, which includes, in particular, the already mentioned cyclones and electrostatic precipitators, but in essence still remains straight-through.

In order to solve the problem of capturing fine fractions of the dust mass contained in the air taken from industrial premises, the authors of this article once proposed a two-circuit closed cleaning system (DCCS) [15, 16]. In this system, the dust-and-air flow is divided into two separate streams, one of which contains fine dust and the other coarse dust. Their subsequent separate cleaning results in an unusually high overall efficiency (up to 95 %) for dry inertial catchers. The principle of the system is that the dusty air stream captured by the collection-return apparatus (CRA) *1* is mixed with the cleaned stream of the circulation circuit at the first stage (Fig. 1). The mixing function is performed by the central ejector *2*. At the second stage, this dust-air mass enters the separation apparatus *3*, which is a 180° turn with a small slit along the outer wall [17]. The air containing mainly coarse fractions enters this slot – the main circuit flow. Depending on the thickness of the gap, the main flow accounts for 4–7 % of the volume flow rate of all air in the system, the rest is the circulation circuit flow. It carries mainly fine dust and is directed to the circulation circuit catcher *5*. A cyclone is usually used as a catcher. Therefore, the cleaning efficiency in the circulation circuit will be low (depending on the properties of the dust, it is 20 to 40 %). However, since the uncollected mass is not released into the atmosphere, but is returned to the cleaning process through the mixing and separation process, this low efficiency is not a problem. In other words, the air is free of fine dust by repeating the cleaning process many times. The main circuit flow enters the collector *4*, and the dust mass not captured in it, which is up to 5 % of the incoming dust mass, is sent to the collection-return apparatus. This small dust mass, together with a new portion of the captured dusty air, enters the mixer *2*. The process is repeated.

A double-circuit closed cleaning system cannot use the collection umbrella typically found in direct-flow aspiration systems, as it is not capable of returning the cleaned air to the main circuit. Therefore, there was a need to develop a separate collection-return element. The simplest of the possible solutions was the device shown in Fig. 2. In this device, the dust-air mixture is drawn from the room through a cylindrical pipe due to the vacuum created by the ejector. The main circuit flow is returned to the collection area of the room through an annular channel. Since there is a need for the free jet of the cleaned stream to be as compact as possible, it was decided to give it spin by means of a tangential supply, based on previous studies.

The first experience with the system proved the effectiveness of the DCCO. The scheme did not require significant capital investment, but provided a significant improvement in dust capture due to the separate cleaning scheme.

Unsolved aspects of the problem. As a pilot project, the DCCO was used to clean the air extracted from the building materials crushing plant at the request of a company specialising in the dismantling of old buildings. The system proved to be highly efficient, cost-effective and reliable, and did not require any additional specialised skills from the operating staff. However, the system was designed individually for the specific conditions of this production facility, and a universal methodology should be developed for its widespread implementation. Such a methodology should include two main components: 1) calculation of the efficiency of the system's elements and its overall efficiency; 2) rules for engineering calculation of the elements' design and selection of fans for the main and circulation circuits. The first component of the methodology was developed. The results of its calculations were well confirmed by the first experience of using the DSSO. Additional research was required to develop the second component.

Purpose. The aim of the work is to develop the basics of engineering (design) calculation of the DSSO, which would allow one to reasonably select injection equipment for the given volume flow rates of the main and circulating flow and the pressure required to provide them.

Methods. The study was carried out by drawing up an energy balance of the system's circuits and numerical modelling of collection-return apparatus in Solid Works Flow Simulation.

Research findings. ***Pressure balance of gas flows of the cleaning system.*** The air ducts of the DCCS form an annular pipeline with a branch – the main channel (MC). The presence of an ejector causes a variable flow rate in the circulation circuit (CC). This makes it impossible to use the methods of calculation of complex pipeline networks, in particular, the most common method of

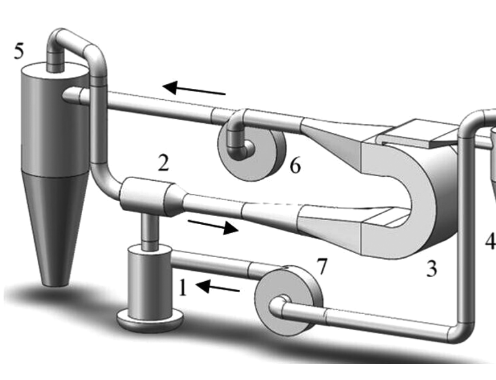

Fig. 1. Scheme of a two-circuit closed cleaning system:

1 – collection-return apparatus; 2 – central ejector; 3 – separation apparatus; 4 – cyclone of the main circuit; 5 – cyclone of the circulation circuit; 6 – fan of the circulation circuit; 7 – fan of the main circuit [15]

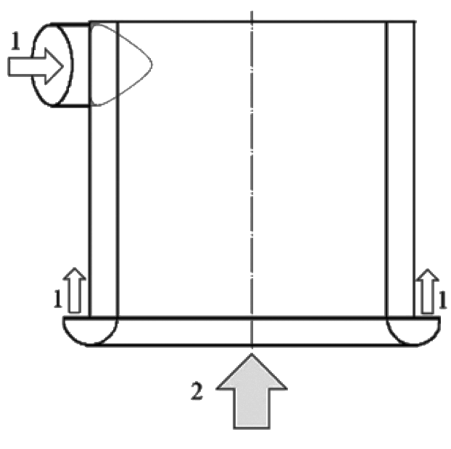

Fig. 2. Scheme of the CRA:

1 – purified air; 2 – polluted air

characteristics, which are familiar to hydraulics. Therefore, it is proposed to solve the problem by means of a pressure balance. The pressure developed by the circulation circuit fan (CCF) P_c overcomes the hydraulic resistance of the elements of this circuit: the catcher, ejector, separator and duct. Then

$$P_c = \Delta P_{c5} + \Delta P_2 + \Delta P_{c3} + \Delta P_{c8}.$$

Pressure loss in the circulation circuit catcher is

$$\Delta P_{c5} = \zeta_{c5} \frac{\rho u_c^2}{2},$$

where ζ_{c5} is the coefficient of local losses of the catcher; u_c is the characteristic velocity, m/s; ρ is the density of gas (air), kg/m^3.

In the scheme of the DCCS, the ejector performs quite standard functions for such devices – creating a vacuum in the central pipe of the air handling unit and mixing the dusty stream taken from the room with the circulation circuit flow. Therefore, a conventional ejector can be used, the calculation of pressure losses in which ΔP_2 is performed according to generally accepted methods for the selected type of device.

Pressure loss in the circulating flow separator is

$$\Delta P_{c3} = \zeta_{c3} \frac{\rho v_c^2}{2},$$

where ζ_{c3} is the coefficient of local losses in the separation apparatus for the circulating flow; v_c is the characteristic velocity, m/s.

The pressure loss to overcome the resistance of the circulation circuit duct, taking into account that the volume flow rate before and after the separation apparatus is different,

$$\Delta P_{c8} = \Delta P_{c81} + \Delta P_{c82};$$

$$\Delta P_{c81} = \left(\lambda_{c1} \frac{l_{c1}}{d_{c1}} + \sum_{i=1}^{n} \zeta_{c1i} \right) \frac{8\rho Q_{c1}^2}{\pi^2 d_{c1}^4},$$

where λ_{c1} is the hydraulic coefficient of friction; d_{c1} and l_{c1} are the diameter and length of the duct, m; ζ_{c1i} is the local loss coefficient of the i^{th} local resistance; Q_{c1} – is the volume air flow rate in the circulation circuit, to the separation apparatus, m^3/s

$$\Delta P_{c82} = \left(\lambda_{c2} \frac{l_{c2}}{d_{c2}} + \sum_{k=1}^{n} \zeta_{c2k} \right) \frac{8\rho Q_{c2}^2}{\pi^2 d_{c2}^4},$$

where λ_{c2} is the Darcy coefficient; d_{c2} and l_{c2} are the diameter and length of the duct, m; ζ_{c2k} is the local loss coefficient of the k^{th} local resistance; Q_{c2} is the volume flow rate of air in the circulation circuit after the separator, m^3/s.

Volumetric flow rate is

$$Q_{c1} = Q_{c2} + Q_m.$$

Losses in the central pipe of the filling station can be neglected due to their smallness.

The pressure developed by the main circuit fan (MCF) P_m overcomes the hydraulic resistance of the elements of this circuit: the catcher, separator, annular channel of the collection-return apparatus and the air duct. Then

$$P_m = \Delta P_{m4} + \Delta P_{m3} + \Delta P_{m1} + \Delta P_{m9}.$$

Pressure loss in the main circuit catcher is

$$\Delta P_{m4} = \zeta_{m4} \frac{\rho u_m^2}{2},$$

where ζ_{m4} loss factor of the main circuit catcher; u_m is the characteristic velocity, m/s.

Main flow pressure loss in the main circuit separator is

$$\Delta P_{m3} = \zeta_{m3} \frac{\rho v_m^2}{2},$$

where ζ_{m3} is the loss factor of the separation apparatus for the main flow; v_m characteristic velocity (average velocity in the slot), m/s).

Pressure loss to overcome main circuit duct resistance is

$$\Delta P_{m9} = \left(\lambda_m \frac{l_m}{d_m} + \sum_{j=1}^{n} \zeta_{mj} \right) \frac{8\rho Q_m^2}{\pi^2 d_m^4},$$

where λ_m is the hydraulic coefficient of friction; d_m and l_m diameter and length of the channel; ζ_{mj} is the loss factor of the j^{th} local resistance; Q_m – air volume flow rate in the main circuit, m^3/s.

The flow rate balance of the system is ensured if the volume flow rate of air in the main circuit is equal to the flow rate of dusty air sucked by the ejector through the central pipe of the CRA Q_{cp}. The value of this flow rate should be set by the conditions of a particular production facility at the design stage of the DCCS. By analogy with the air exchange of premises, the concept of circulation rate $K(1/h)$ – $Q_{cp} = KV$ (V – volume of the production facility, m^3) could be used to justify the flow rate. However, since the DCCS is a recirculation-type aspiration purification system for which there are no ready-made recommendations on the values of K, Q_{cp} should be determined in a different way – from the condition $Q_{cp} = M_d/C$, where M_d is the second mass of dust emitted from production facilities, g/s; C is the average dust concentration in the room, g/m^3. C should not exceed the maximum permissible concentration for dust in the working area regulated by the General Sanitary and Hygienic Requirements for Working Area Air GOST 12.1.005-88. In addition, the condition $M_d \leq M_{cat}$, where M_{cat} is the second mass of dust collected in the main and circulation circuits of the cleaning system, g/s, must be met in the DCCS. M_{cat} is determined in accordance with the methodology for calculating system performance indicators [15].

Pressure loss in the annular channel of the collection-return apparatus is

$$\Delta P_{m1} = \zeta_{m1} \frac{L}{H} \frac{\rho v_{m1}^2}{2},$$

where L is the length of the CRA, m; H – the width of the annular channel CRA, m; v_{m1} – average speed in the annular channel, m/s.

ΔP_{m1} consist of losses along the length and in local resistances (entrance to the channel from the tangential branch pipe and flow turn at the outlet of the channel). In this case; ζ_{m1} is a complex coefficient of hydraulic losses in the annular channel, which takes into account both the length losses and local losses, including the effect of the airflow twisting effect on the value of the total losses. Due to the complexity of the process hydrodynamics, it is impossible to consider the two types of losses separately.

The values such as pipeline lengths and diameters and volumetric flow rates, and therefore average velocities, which are included in the formulas above, are individual values for each particular DCCS. They are specified in the design specification. The hydraulic friction coefficients are determined by well-known relations (for example, the Schifrinson formula) depending on the equivalent roughness of the pipe material, and the local loss coefficients are determined by reference data. Therefore, it is easy to see that the only value for which there is no standard calculation method is the complex coefficient ζ_{m1}.

Since pressure channels with an annular cross-section are often used in heat exchangers, most of the previously conducted studies of gas flow under such conditions were mainly focused on the calculation of the heat transfer process [18, 19]. The nature of the dependence of the pressure loss on the Reynolds number under conditions when the inner wall rotates was studied in [20]. The results showed the dependence of the loss value on the relative thickness of the channel, but the experiments were carried out at low (close to the critical) Reynolds numbers. The swirling flow was also studied at low Re in [21], but the issue of hydraulic resistance was not considered. Therefore, there was a need to conduct a separate study on the hydrodynamic features of the swirling flow in the annular channel in order to quantify the complex coefficient ζ_{m1}.

Numerical modelling of gas flow in the collection-return apparatus. The hydrodynamic features of the swirling flow in the annular channel were studied using numerical modelling in the Solid Works Flow Simulation application package.

Modelling criteria. At the preliminary stage, the dimensionless modelling criteria were defined for this task. This was done with the help of the π-theorem, according to which the functional relationship between p dimensional quantities can be represented as a relationship between k dimensionless complexes (π numbers) composed of these dimensional quantities. The number of complexes is $k = p - n$. Here, n is the number of basic units of measurement (n = 3; mass (M) is measured in kilograms, kg; geometric dimension (L) is measured in metres, m; time (T) is measured in seconds, s).

The dimensions that can affect the flow are H (m) – the width of the annular channel; D_1 (m) – the inner diameter; L (m) – the length of the channel; $v = v_{m1}$ (m/s) – the gas velocity in the annular channel; ρ (kg/m^3) – the air density; μ (Pa · s) – the coefficient of dynamic air viscosity; δ (m) – the roughness of the inner walls of the pipe. Seven dimensional quantities lead to the need to obtain four dimensionless complexes π.

Given that dim (H; L; δ) = L, dim $v = LT^{-1}$, dim $\rho = ML^{-3}$, dim $\mu = ML^{-1}T^{-1}$, using the Rayleigh method, we obtain

$$\pi_1 = Hv\rho D_1; \quad \pi_2 = Hv\rho\delta;$$

$$\pi_3 = Hv\rho L; \quad \pi_4 = Hv\rho\mu,$$

where dim $\pi_1 = L^{x1}(LT^{-1})^{y1}(ML^{-3})^{z1}L$, whence

$$\left.\begin{aligned} x_1 + y_1 - 3z_1 + 1 = 0 \\ -y_1 = 0 \\ z_1 = 0 \end{aligned}\right\},$$

$x_1 = -1$, that is

$$\pi_1 = \frac{D_1}{H}.$$

Similarly, we get

$$\pi_2 = \frac{\delta}{H}; \quad \pi_3 = \frac{L}{H},$$

where dim $\pi_4 = L^{x4}(LT^{-1})^{y4}(ML^{-3})^{z4}(ML^{-1}T^{-1})$, whence

$$\left.\begin{aligned} x_4 + y_4 - 3z_4 - 1 = 0 \\ -y_4 - 1 = 0 \\ z_4 + 1 = 0 \end{aligned}\right\},$$

and

$$\pi_4 = \frac{\mu}{Hv\tilde{n}} = \frac{1}{\mathrm{Re}},$$

where Re is the Reynolds number.

Boundary conditions and design relations. The scheme and geometric dimensions of the experimental apparatus are shown in Fig. 3. The length L = 5.5 m and width H = 100 mm of the annular channel were unchanged. The roughness δ = = 0.15 mm, which corresponds to the value for a surface made of galvanised iron, also did not change. Thus, $\pi_2 = 1.5 \cdot 10^{-3}$ and $\pi_3 = 55$. The criterion π_1 was set to four values by changing the inner diameter D_1 and the corresponding change in the outer diameter D_2, and the criterion π_4 was set by changing the average velocity in the annular channel. The simulation was performed for air with ρ = 1.2 kg/m^3 and $\mu = 1.81 \cdot 10^{-5}$ Pa · s.

In cross section 1-1 of the inlet, the volume flow rate $Q_1 = Q_m$ (m^3/s) was set (Fig. 3). By changing it, the value of the velocity in the annular channel v was varied. In cross section 2-2, the pressure P_2 = 101.3 kPa (atmospheric pressure) was set. All solid surfaces were subject to the effect of sticking, i. e., the velocity had a zero value.

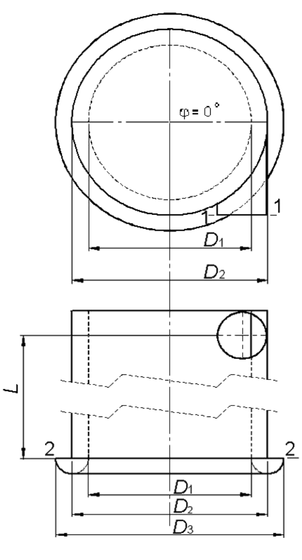

Fig. 3. Geometric parameters of the CRA

Based on the results of the numerical modelling, the pressure P_1 was determined in section 1-1. The pressure difference (P_1–P_2) was used to calculate the complex hydraulic loss coefficient

$$\zeta_{m1} = \frac{2H(P_1 - P_2) + \rho H(v_0^2 - v_{m2}^2)}{\rho v_{m1}^2 L}.$$

Speed in the inlet branch pipe CRA

$$v_0 = \frac{4Q_m}{\pi d_0^2},$$

where d_0 is the diameter of the inlet branch pipe, m.

Speed in the annular channel is

$$v_{m1} = \frac{4Q_m}{\pi(D_2^2 - D_1^2)}.$$

The speed at the exit of the CRA is

$$v_{m2} = \frac{4Q_m}{\pi(D_3^2 - D_2^2)}.$$

The Reynolds number is

$$\mathrm{Re} = \frac{v_{m1}\rho H}{\mu}.$$

Modelling results and their analysis. On the basis of the calculations, graphs of the dependence of ζ_{m1} on the Reynolds number were constructed for four values of π_1 (Fig. 4). The graphs showed that the numerical experiments were carried out mainly in the region of ζ_{m1} autosimilarity with Re. At the same time, the larger π_1, the higher the complex hydraulic loss coefficient. Fig. 5 shows the dependence of ζ_{m1}on π_1 for the automodel zone of hydraulic resistance.

The simulation results showed that no vortex zones occurred in the flow segment from the inlet to the outlet of the apparatus in the experimental intervals of the Reynolds number and π_1 numbers. Thus, the energy lost by the flow is spent on overcoming additional tangential stresses caused by the kinematic restructuring of the flow, including those associated with its twisting.

The nature of the change in the distribution of the axial component of the flow velocity uz in the annular channel can be traced in Fig. 6 (L_i is the value of the distance of the cross-section under consideration from the axis of the inlet of the CRA, as shown in Fig. 3). It can be visually seen that the tangential flow inlet causes a much greater uneven distribution of uz across the annular channel cross-section than if it were caused only by the action of viscous friction forces. Moreover,

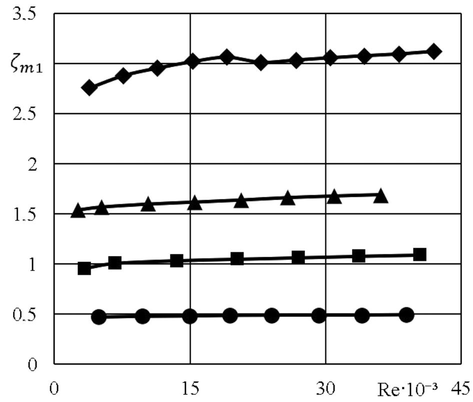

Fig. 4. Dependence of ζ_{m1} on the Reynolds number:
● – D_1 = 800 (π_1 = 8); ■ – D_1 = 1200 (π_1 = 12); ▲ – D_1 = 1600 (π_1 = 16); ♦ – D_1 = 2200 (π_1 = 22)

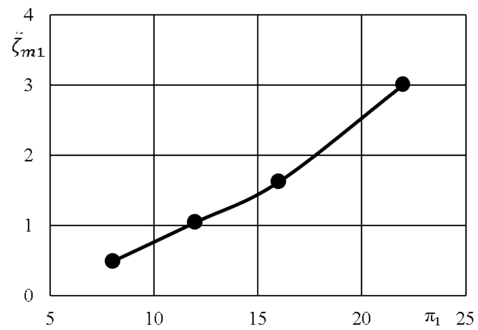

Fig. 5. Dependence of ζ_{m1} on π_1 for the automodel zone of hydraulic resistance

the distribution of the axial velocity is largely non-axisymmetric, although this effect becomes weaker with distance from the inlet.

The intensity of flow twisting can be quantified by the ratio of the average circumferential velocity in a particular cross-sectional section v_φ to the average flow rate v_{m1}. For this purpose, four mutually perpendicular sections were selected in the sections with L_i/H = 5.0–31.0, the orientation of which was set at an angle φ = 0, 90, 180 and 270°. The angle was counted counterclockwise starting from the first flow out of the nozzle. The average circumferential velocity in the bore was determined by the formula

$$v_\varphi = \frac{1}{H}\int_0^H u_\varphi(h)dh,$$

where $u_\varphi(h)$ is the distribution of the circumferential velocity in the formation based on the results of numerical modelling; h is the current coordinate in the formation, m.

The results of the calculations are shown in the Table. It can be seen that the intensity of the twist decreases in a wavy manner as the flow approaches the outlet of the annular channel. The waviness indicates that the tangential inlet not only imparts swirl to the flow, but also makes the flow helical. This circumstance, firstly, is of great theoretical importance for the hydrodynamics of a swirling flow in an annular channel, and, secondly, is of purely practical importance. This flow character should be taken into account when finalising the length of the annular channel. For example, if L/H equals 25, the intensity of the twist in different zones of the outlet cross-section will differ significantly (1.16; 0.89; 0.59; 0.39), which will lead to a significant violation of the axial symmetry of the free jet. This will adversely affect the completeness of the secondary capture of dust particles that were not captured in the main circuit ap-

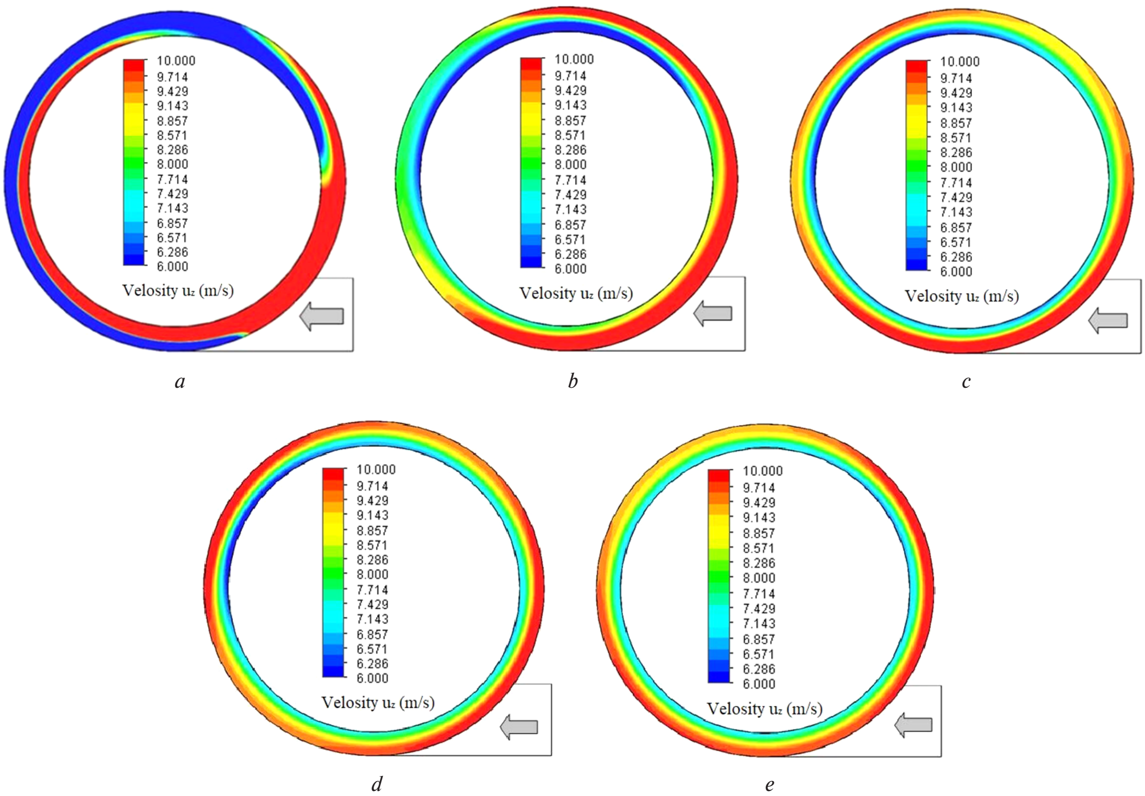

Fig. 6. Distribution of the axial velocity component u_z in the annular channel:
a – $L_i/H = 5$; *b* – $L_i/H = 10$; *c* – $L_i/H = 15$; *d* – $L_i/H = 20$; *e* – $L_i/H = 25$

Table

Changing the twist parameter along the annular channel

L_i/H	v_φ/v_{m1}			
	φ = 0°	φ = 90°	φ = 180°	φ = 270°
5.0	2.52	1.95	1.20	0.62
6.0	2.43	2.19	1.58	1.13
7.0	2.25	2.19	1.79	1.50
8.0	1.78	1.80	1.92	1.64
9.0	1.20	1.32	1.98	1.66
10.0	1.04	0.94	1.75	1.64
11.0	1.01	0.81	1.35	1.60
12.0	1.13	0.90	1.03	1.42
13.0	1.45	1.14	0.90	1.20
14.0	1.64	1.30	0.82	0.90
15.0	1.65	1.38	0.88	0.76
16.0	1.52	1.48	1.00	0.82
17.0	1.23	1.47	1.07	0.93
18.0	1.03	1.32	1.16	1.04
19.0	0.76	1.05	1.20	1.15
20.0	0.78	0.82	1.08	1.18
21.0	0.91	0.67	0.87	1.17
22.0	1.02	0.75	0.62	0.92
23.0	1.10	0.79	0.52	0.65
24.0	1.17	0.83	0.56	0.49
25.0	1.16	0.89	0.59	0.39
26.0	1.00	0.99	0.66	0.40
27.0	0.81	1.03	0.73	0.46
28.0	0.65	1.02	0.77	0.52
29.0	0.58	0.95	0.80	0.58
30.0	0.45	0.88	0.75	0.60
31.0	0.49	0.70	0.68	0.59

paratus. However, this disadvantage can be significantly reduced by choosing $L/H = 22.0$ or $L/H = 29.0$.

Conclusions. Thus, it has been shown that the hydraulic calculation of a two-circuit closed air purification system can be carried out by drawing up a pressure balance of individual circuits. This makes it possible to determine the required pressure values, and therefore, according to the characteristic $P = f(Q)$, to select fans for the main and circulation circuits from existing catalogues.

Most of the values that make up this balance can be calculated in accordance with the known rules of hydraulics. To calculate the pressure loss in a filling station, the concept of a complex hydraulic loss coefficient is introduced, which integrally takes into account three factors that affect its value. By means of numerical modelling, the dependence of this coefficient on the relative width of the channel and the main regime parameter (Reynolds number) was determined. It turned out that the narrower the channel, the greater ζ_{m1} and, accordingly, the greater the pressure loss. In the range of practically probable Reynolds numbers, there is no dependence of ζ_{m1} on Re.

The analysis of the flow simulation results showed that the tangential inlet gives the flow in the pressure annular channel not only a twist, but also makes the flow helical.

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Основи розрахунку двоконтурної системи очищення повітря від полідисперсного пилу

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Мета. Підвищення рівня екологічної безпеки підприємств за рахунок поліпшення якості очищення повітря від полідисперсного пилу у двоконтурних замкнутих системах, зокрема отримання основних розрахункових співвідношень для інженерного розрахунку таких систем.

Методика. Мета дослідження реалізовувалася шляхом математичного й числового моделювання гідродинамічних процесів у елементах замкнутої двоконтурної системи очищення.

Результати. Запропонована методика гідравлічного розрахунку замкнутої двоконтурної системи очищення шляхом складання балансу тиску окремих контурів, визначена залежність для комплексного коефіцієнту гідравлічних втрат апарату збирання-повернення.

Наукова новизна. Гідравлічний розрахунок двоконтурних замкнутих систем очищення запропоновано проводити шляхом складання балансу тиску окремих контурів. Для розрахунку специфічного елементу системи – апарату збирання-повернення – уведено поняття комплексного коефіцієнту гідравлічних втрат, що враховує як місцеві втрати тиску, так і втрати за довжиною, а також опосередковано відображає вплив закрутки потоку на гідравлічний опір цього елементу. Для комплексного коефіцієнта гідравлічних втрат шляхом числового моделювання гідродинамічних процесів течії закрученого потоку у кільцевому напірному каналі отримані кількісні результати, необхідні для інженерних розрахунків.

Практична значимість. Отримані результати дозволяють проєктувати двоконтурні замкнуті системи очищення для різних виробничих умов, що, у свою чергу, дає можливість замінювати типові й неефективні прямоточні системи на систему, у якій за рахунок роздільного очищення значно підвищується ефективність уловлювання полідисперсного пилу.

Ключові слова: *екологічна безпека, полідисперсний пил, система очищення, гідравлічні втрати, баланс тиску*

The manuscript was submitted 02.11.23.