ISSN: 2350-0328
International Journal of AdvancedResearch in Science, Engineering and Technology
Vol. 12, Issue 1, January 2025

Characteristics of Oncoming Dust Collectors Swirling Streams

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ABSTRACT: The article discusses the issues of identifying the characteristics of dust collectors with counter swirling flows. The work investigated the hydrodynamic situation that is created by the interaction of counter swirling flows and the radial distribution of static pressure values determined by the height of the apparatus and the distribution of static pressure in the exhaust pipe in order to create a complete physical picture. The distribution of axial velocity along the radius of the primary and secondary inlets and the radius of the exhaust pipe and the distribution of static pressure were determined using a five-channel ball probe. A feature of swirling flows is a significant static pressure gradient caused by the presence of a tangential component of gas velocity. The calculation of static pressure and speed at any point of the apparatus was carried out using known equations. The distribution along the radius of the static pressure apparatus was experimentally obtained for different ratios of gas flow rates across the inlets.

KEYWORDS: dust collector, hydrodynamics, speed, swirling flows, static pressure.

I. INTRODUCTION

Currently, much attention is being paid to the environmental safety of industrial plants, including cleaning the air emissions of enterprises from dust. Industrial enterprises emit millions of tons of dust into the atmosphere per year, resulting from the combustion of natural fuels, cement production, the metallurgical industry, and other industries [1]. One of the promising types of equipment for cleaning dust emissions are inertial dust collectors, which are used at the primary stages of air purification from dust, which can significantly reduce air pollution, as well as reduce the load on subsequent gas purification devices and reduce their wear.

Inertial dust collectors according to the principle of dust removal belong to dry mechanical dust collectors [2]. In inertial devices, the flow of dust particles suspended in the gas stream undergoes a sharp change in the direction of movement, while the dust particles do not move behind the stream due to high inertia, which causes inertial forces that tend to eject particles from the stream. As a result, the dust particles are pressed against the walls of the dust collector and then poured into the hopper. The advantages of inertial dust collectors are their small size and low metal consumption due to the relatively high rate of entry of dust gas flow (15-25 m/s) into the cleaning devices [2]. Among inertial dust collectors, jet-inertial dust collectors with counter swirling streams, which have high dust cleaning efficiency and low resistance to movement of the dust and gas stream, are the most widespread.

Modeling of dust collectors with oncoming swirling streams requires a comprehensive study of the hydrodynamic environment that is created by the interaction of oncoming swirling streams. The axial velocity distributions along the cross-sectional radius of the primary and secondary input streams and along the radius of the exhaust pipe were studied in order to create a complete physical picture, since very few such studies have been conducted for dust collectors with oncoming swirling streams [3]. A large number of works have been devoted to the distribution of velocity fields over the section of the apparatus [3,4,7].

II. THE EXPERIMENTAL RESEARCH METHODOLOGY IS AS FOLLOWS

All axial velocity measurements were carried out in the gas phase, as it was shown in [4] that the velocity fields of air flows practically do not deform when the solid phase is introduced.

A distinctive feature of swirling flows (in particular, counter-swirling flows) is the presence of a radial static pressure gradient associated with the presence of a tangential component of the gas velocity. The work determined the radius distribution of static pressure values determined by the height of the device, and the distribution of static pressure

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in the exhaust pipe. All of the above hydrodynamic characteristics have a significant impact on the main technical and economic characteristics of a dust collector with counter-swirling flows (capture efficiency and pressure loss) [5,6].
The axial velocity distribution along the radius of the primary and secondary inlets and the radius of the exhaust pipe and the static pressure distribution were determined using a five-channel ball probe with a sensor element diameter of 9 mm [7].During measurements, the angle σ was determined by the coefficient

$$K_\sigma = \frac{h_3 - h_1}{h_2 - h_4} = \frac{K_3 - K_1}{K_2 - K_4} \quad (1)$$

According to equation (1), the value of K_σ was determined and the calibration graph showed the angle σ [7], which determined the value of the coefficients K2 ; (K2-K4); (K3-K1) .

The calculation of static pressure and velocity at any point was carried out using the following equations:

$$h_{\text{ст}} - h_a = \frac{K_m * m}{1+0.001(t-15)} * \frac{760(273+t)}{B*273} * \frac{\rho_{\text{ж}} * \gamma_r}{\rho_b * 1000} * (h_2 - K_2 * \frac{h_2 - h_4}{K_2 - K_4}) \quad (2)$$

$$V = \sqrt{K_m} * \sqrt{m} * \frac{1}{\sqrt{1+0.001(t-15)}} * \sqrt{\frac{760(273+t)}{B*273}} * \sqrt{\frac{\gamma_r}{\rho_r} * \left(\frac{h_3 - h_1}{K_3 - K_1} - \frac{h_2 - h_4}{K_2 - K_4}\right)} * \sqrt{\frac{\rho_{\text{ж}}}{\rho_b * 1000}} \quad (3)$$

where $K_m = K_1 * K_2 * K_3 * K_4$ – coefficient of micromanometers;

The Vz component was determined by dependence:

$$V_z = -V * \cos\sigma * \sin\varphi \quad (4)$$

Calculation by equations (1), (2), (3) it was conducted using a computer.
The calibration of the probe was carried out by a reference Prandtl tube in a wind tunnel with a strict orientation of the flow in space [7]. Using a Prandtl tube, hst and hsk were determined.

III. SIMULATION&RESULTS

As noted above (paragraph 1.2), a characteristic feature of swirling flows is a significant static pressure gradient caused by the presence of a tangential component of the gas velocity Vφ.
In Fig. 1., fig.2 and Fig.3. The obtained distribution over the radius of the static pressure apparatus is shown. Static pressure measurements were carried out at six height levels (H = 0; 0,04; 0,08; 0,12; 0,16; 0,2 m).
As can be seen from the figures, the static pressure profile depends on the ratio of flow rates through the channels of the device. (Q_2/Q in experiments, it was 0.6; 0.7; 0.8). With increasing ratio Q_2/Q the pressure on the axis increases, and on the periphery (at the wall) decreases. With increasing Q_2/Q alignment of profiles is observed R_{st}
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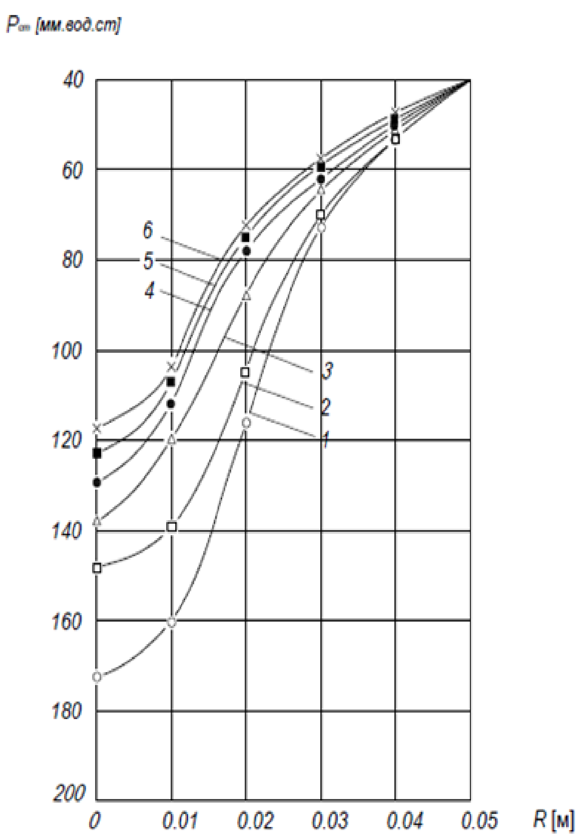

Fig. 1 Static pressure distribution over the section of the V apparatus (K=0.61)

1. ○ - H = 0.2 м, 2. □ - H = 0.16 м,
3. △ - H = 0.12 м, 4. ● - H = 0.08 м,
5. ■ - H = 0.4 м, 6. × - H = 0

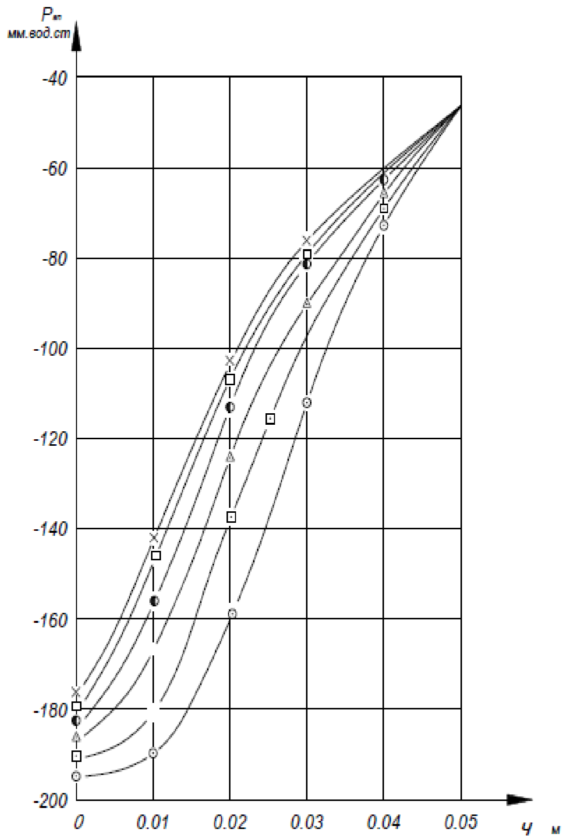

Fig. 2 Static pressure distribution over the section of the apparatus (K=0.7)

1. ◎ - H = 0.2 ; 2. ⊡ - H = 0.16 ;
3. △ - H = 0.12 ; 4. ◐ - H = 0.08 ;
5. ⊡ - H = 0.04 ; 6. × - H = 0

To a first approximation, the distribution of Pst can be described by solving the radial equilibrium equation [9,13,14,15]

$$\frac{\partial P}{\partial r} = \rho_r \frac{{V_\varphi}^2}{r} \qquad (5)$$

This equation satisfactorily describes the distribution of Vz up to, r=r* at the wall, the discrepancy reaches 50%, which can be explained by the presence of turbulent pulsations, the effect of which is significantly affected by the walls of the apparatus [8,9,10].

In Fig.4. The distribution of static pressure in the exhaust pipe of the VZP-200 dust collector is obtained. As can be seen from the distribution shown, with increasing K, the static pressure value at the pipe axis increases, while at the periphery (at the pipe wall) the static pressure level is approximately the same (profile alignment occurs).

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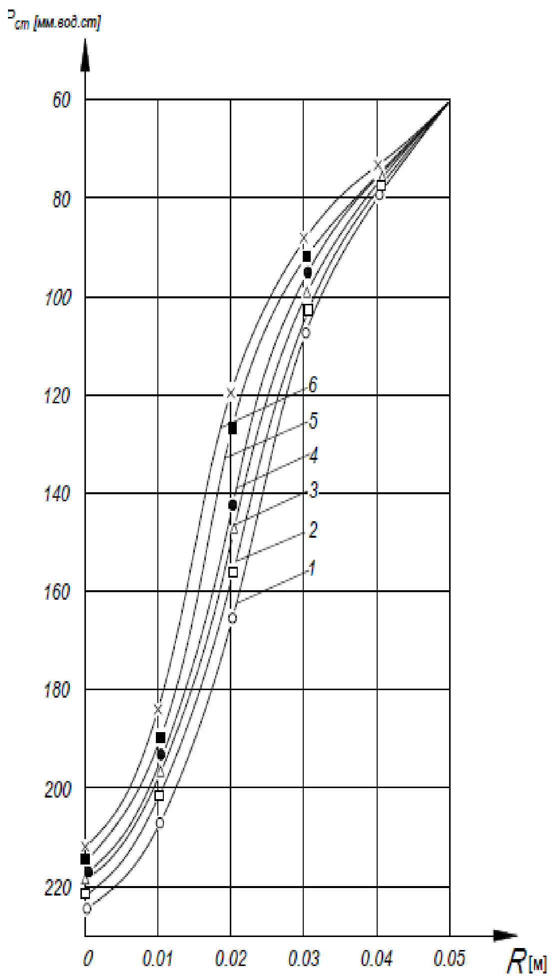

Fig.3. Static pressure distribution over the section of the VZP – 200 apparatus (K=0.8)

1. ○ - H = 0.2 ;
2. □ - H = 0.16 ;
3. △ - H = 0.12 ;
4. ● - H = 0.08 ;
5. ■ - H = 0.04 ;
6. × - H = 0

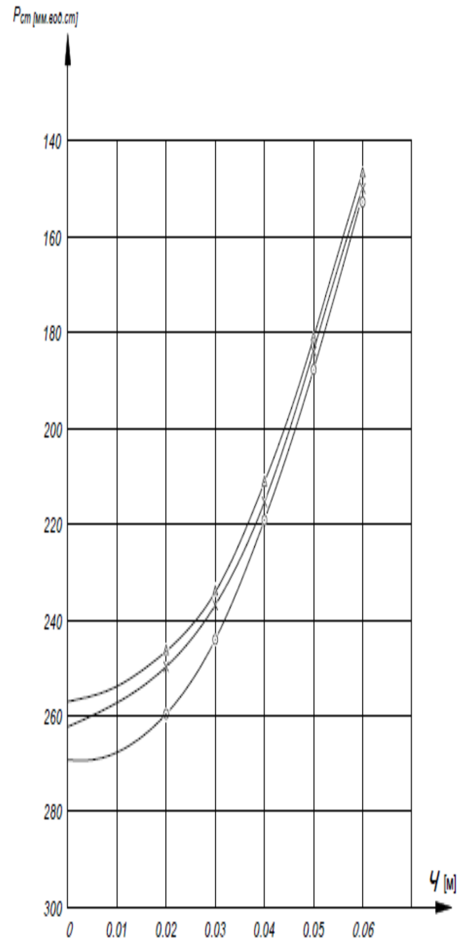

Fig. 4.Static pressure distribution over the exhaust pipe section of the device VZP – 200. Q = 0,22 м^3/с.

1. ○ - K = 0.802
2. × - K = 0.7
3. △ - K = 0.6

From the presented graphs, it can be seen that with an increase in the flow rate ratio through the channels, pressure losses increase (both in the separation volume itself and in the exhaust pipe). This once again confirms the results that were obtained in the study of pressure losses in the VZP apparatus.
The results obtained are also related to the efficiency of trapping and qualitatively confirm the experimental data obtained on the efficiency of trapping, depending on the ratio of channel costs.

IV. CONCLUSION

Based on the data obtained, the following conclusions can be drawn:
1. The static pressure fields in the VZP apparatus are determined and it is established that with an increase in the flow ratio, the difference in static pressure in the apparatus is equalized.
2. The distribution of static pressure over the section of the apparatus at different flow ratios is obtained.

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International Journal of AdvancedResearch in Science, Engineering and Technology
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3. The axial velocity distribution is obtained along the radius of the section at the outlet of the primary and secondary inlets, as well as along the radius of the exhaust pipe of the apparatus.
4. It is established that with an increase in the flow rate ratio through the channels, pressure losses increase, which confirms the results obtained in the study of pressure losses in the VZP apparatus.

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