Pressure losses in apparatuses with swirling flows

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Abstract. We consider the method of calculating the pressure in apparatuses with swirling flows, depending on the different materials. The influence on the calculation of pressure losses in the two-phase swirling flows of various parameters for the organization of fuel combustion processes, mass and heat transfer, dust collection. Dependence of pressure drop on the amount of energy consumed and the choice of blowers for apparatuses with swirling flows. The causes of pressure drops in two-phase swirling flows are given and listed. The loss of gas kinetic energy in apparatuses with twisted and counter twisted flows of gas suspensions exceeds losses from all other causes, that only this loss should be taken into account. The peculiarities of design and pressure losses in vortex pressureless apparatuses are considered. The results of investigating the dependence of pressure losses in the apparatus on the air flow rate through the vortex chamber and the results of investigating the effect of the ratio of flow rates through the apparatus channels are presented.

1 Introduction

The pressure loss in any dust-collecting apparatus is a value that characterizes the energy consumption during the separation process.

Knowing dependence of pressure drop from the main design and operating parameters, it is possible to choose the correct blowing equipment, the size of the supply air ducts, the operating mode without the drift apparatus, allowing to conduct the dust collection process at minimum energy consumption. In addition, pressure loss is the most important characteristic (along with the collection efficiency), serving to compare different types of dust collectors. The main causes of pressure losses in the centrifugal dust collector are described in the relevant literature [1-3.]

2 Main part

The calculation of pressure losses in the two-phase swirling flows and dependence of these losses on various parameters is of interest for calculation and operation of vortex

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apparatuses, widely used for organizing the processes of fuel combustion, mass - and heat transfer, dust collection. Thus, the amount of energy consumed and the choice of blowers depend on the pressure drop.

The following reasons are known for pressure drop: pressure losses in the inlet pipe due to friction; losses due to gas expansion or compression at the inlet; losses in the unit due to friction against the walls; losses of kinetic energy in the unit; losses at the inlet to the outlet pipe; hydrostatic head between the inlet and outlet pipes; energy recovery in the outlet pipe [1-7].

Similar theories considering a number of causes of pressure drop are given in [1-3 etc.].

Some researchers [2, 3, 8] believe that loss of kinetic energy by gas in apparatuses with twisted and counter twisted flows of gas suspensions exceeds losses from all other causes so much, that only this loss should be taken into account. Then the pressure drop depending on the inlet velocity wi and dimensionless coefficient of hydrodynamic resistance **ξ**, which in this case expresses the value of losses, is written in the form:

$$\Delta \mathrm{P} = \xi \frac{\rho w_i^2}{2g} \tag{1}$$

The hydrodynamic drag coefficient is a function of velocity [1, 7]. The most universal empirical dependencies for calculating this coefficient are available in [1, 4, 9].

In the presence of a swirl device, there are significant pressure losses in the apparatus [4, 5, 9]; in this case, the degree of influence of the pitch of the swirl blade S of the swirler on hydrodynamic resistance is estimated as ΔP ~ S-0.35 [2].

Small resistance of swirl chamber and apparatus VZP, as well as high tangential gas velocity is achieved by application of dispersed gas input along the chamber circumference [1, 5, 7, 9, 10].

Thus, in the apparatuses with swirling gas-suspension flows, the pressure losses largely depend on the geometrical characteristics of the apparatuses, gas density, gas velocity in the inlet and outlet pipes, and the concentration of materials in the gas flow.

Let us consider the pressure losses in vortex pressureless apparatuses [6]. Design features of vortex dryer with no-drain (the presence of three inlet and one outlet branch), does not allow to carry out direct measurement of pressure in each branch of the apparatus. However, based on these measurements, it is possible to calculate the total (equivalent) pressure drop in the apparatus.

Let's write down the equation of energy balance of flows in the apparatus to determine the total pressure drop in the vortex pressureless apparatus in the form:

$$\left(\mathrm{P}_1 + \frac{\rho_{\text{г}1} V_1^2}{2}\right) L_1 + \left(\mathrm{P}_2 + \frac{\rho_{\text{г}2} V_2^2}{2}\right) L_2 + \left(\mathrm{P}_3 + \frac{\rho_{\text{г}3} V_3^2}{2}\right) L_3 =$$

$$= \left(\mathrm{P}_4 + \frac{\rho_{\text{г}4} V_4^2}{2}\right) L_4 + \Delta \mathrm{P}_0 L_0 \tag{2}$$

Here, P1, P2, P3, P4 - pressure in the middle, lower, upper and outlet inlets, respectively; ρr1V1, ρr2V2, ρr3V3, ρr4V4 - gas density and speed in the inlet and outlet pipes of the apparatus; **Δ**P0 - pressure loss in the apparatus; L_0 - total gas flow through the apparatus.

Opening the brackets and grouping the terms of equation [3], we obtain a relation for determining **Δ**P0 in a case, when the coolant temperature in the channels of the apparatus is the same (ρr1 = ρr2 = ρr3 = ρr4)

$$\Delta p_0 = \frac{p_1 L_1 + p_2 L_2 + p_3 L_3}{L_0} + \frac{\rho_г}{2L_0}\left(\frac{L_1^3}{S_1} + \frac{L_2^3}{S_2} + \frac{L_3^3}{S_3} + \frac{L_4^3}{S_4}\right) - p_4 \quad (3)$$

where S1, S2, S3, S4 are the areas of inlet and outlet spigots of the unit.

From expression (3) we obtain the equation for determining the hydraulic resistance coefficient of the apparatus $\boldsymbol{\xi}$:

$$\xi = \xi_1^{(1-К)} - \xi_2^{К} - \xi_3^{К/2} + \left[\frac{(1-К)^3}{a_1^2} + \frac{K^3}{a_2^2} - \frac{1}{b^2}\right] \quad (4)$$

Where

$$\xi = \frac{2\Delta P}{\rho_г V^2}; \quad \xi_1 = \frac{2\Delta P_{1ст}}{\rho_г V_1^2}; \quad \xi_2 = \frac{2\Delta P_{2ст}}{\rho_г V_2^2}; \quad \xi_3 = \frac{2\Delta P_{3ст}}{\rho_г V_3^2};$$

kcoefficient of hydraulic resistance of the primary, secondary and main inlets of the apparatus.

a1 = s1/s; a2 = s2/s; b = s3/s;

s is the cross-sectional area of the apparatus;
k - coefficient of the ratio of air flow rates by inputs.

3 Methods

The experiments to determine the pressure losses in the vortex dryer of no-bottom type - apparatus with counter-flows with a vortex chamber were conducted on the experimental setup SVZP-VK 200/340 [6].

The experiments were carried out on the cold model in idle mode - for pure gas phase and in the presence of material particles according to the following procedure.

The required air flow rate through the apparatus channels was set by regulating the dampers on the inlet branch pipes according to the readings of measuring diaphragms. After the outlet of the drying apparatus on a stationary mode, we measured the pressure drop in the apparatus channels with the help of impulse tubes connected to micromanometers MMN-240.

Static pressure under experimental conditions in the channels of primary and secondary flows was controlled by micromanometers MMN-240.

During the experiments, the desired operating mode of the apparatus was set by means of the slide valve, and after the mode stabilization the necessary measurements were made.

The experiments to determine the pressure loss were determined in the gas phase, since in many works [3] the influence of dustiness on the pressure drop was not observed, or an insignificant influence of dustiness on the pressure loss in the apparatus was observed [3, 4]. The total pressure loss in the apparatus ΔP was found according to dependence (3).

Measurements were carried out at steady-state readings of the micromanometers. The air temperature during the experiment was 297 K. Using the obtained values and the formula [4], the total pressure drop in the vortex dryer without entrainment was calculated.

4 Results

Fig. 1 shows the results of the study of the dependence of pressure loss in the apparatus on the air flow rate through the vortex chamber. During the experiment, the air flow rate through the upper and lower channels did not change.

Fig. 1 shows that in the investigated air flow rate variation interval (up to 9-10-2 m3 /s), the hydraulic resistance of the apparatus increases with increasing air flow rate L1. Accordingly, the specific energy consumption increases with increasing air volume, which is related to the pressure loss by a directly proportional relationship:

$$\Delta E = \frac{\Delta P}{3600} \text{kW}\cdot\text{h/ 1000m}^3 \quad (5)$$

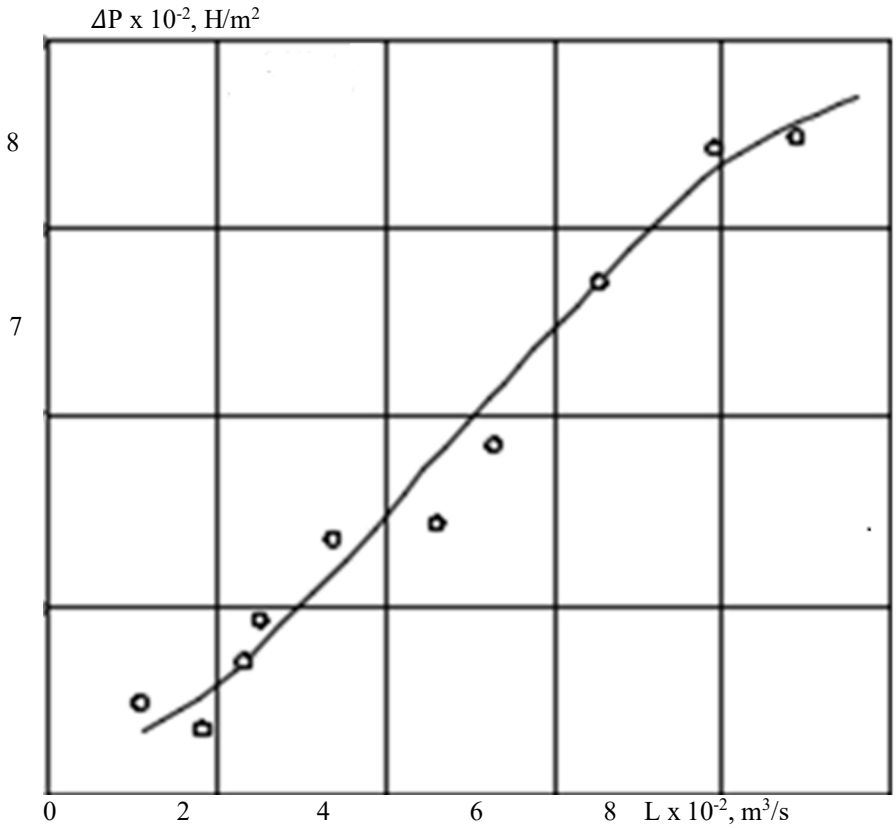

Fig. 1.Dependence of pressure loss on coolant flow rate.

$$L_1 = 0{,}012 \text{ м}^3/\text{c}; \quad L_2 = 0{,}025 \text{ m}^3/\text{s}.$$

In the presence of material in the apparatus, the pressure drop is less at the same values of air flow through the vortex chamber and dryer as a whole as in the apparatus not loaded with material, and the difference in pressure drop between empty and loaded material in the apparatus is the greater the material concentration. Thus, at a material concentration of 0.12 kg/kg the pressure loss in the apparatus decreases on average by 5%, and at a material concentration of 0.47 kg/kg - by 11%. In the experiments, the model material was cellulose acetate (ρm = 1330 kg/m^3). Reducing the pressure drop in the vortex without a carry-over

apparatus when feeding the material as compared with an empty apparatus can be explained by a decrease in the tangential component of the gas velocity due to dissipation of the energy of the gas flow weight when the gas flow and material particles friction against the walls of the apparatus. This is confirmed by the results of experimental studies of the velocity field of the gas phase in the apparatus.

Reducing the pressure loss in the vortexless apparatus without entrainment of material has a favorable effect on the operation of the apparatus in industrial conditions.

Fig. 2 shows the results of studies of the effect of the ratio of flow rates through the apparatus channels $\mu = (L_1 + L_3)/L_2$ on pressure losses in the apparatus, at different total air flow rates.

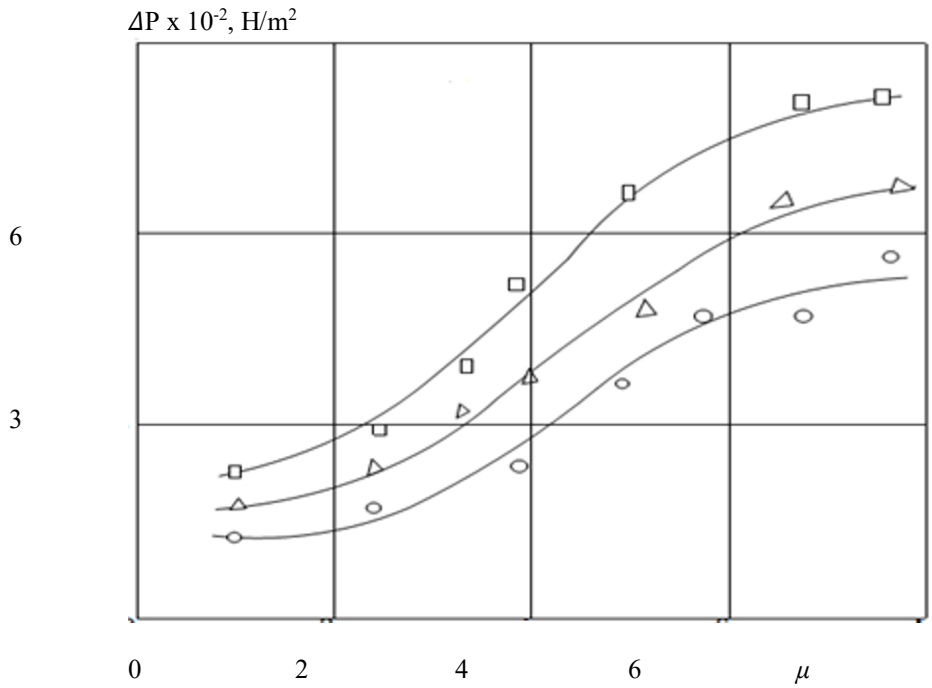

Fig. 2. Dependence of pressure loss on resistance of gas flow along the channel.

○ - 9 x 10^{-2} m^3/s; Δ - 1,2x 10^{-1} m^3/s; □ – 1,5x 10^{-1} m^3/s;

As can be seen in Fig. 2, the pressure loss increases with increasing μ, and in the initial section the pressure loss is determined mainly by the resistance arising in the upward gas flow inlet channel L_2 and the outlet nozzle of the unit.

Starting from the ratio μ equal to 1,5 -:- 2 (for different total gas flow rates) the pressure drop in the vortex chamber and downward gas flow spigot starts to dominate over other losses, and the main share of pressure loss falls on the vortex chamber. At the same time, there is an increase in the total pressure drop in the apparatus.

The coefficient of hydraulic resistance of the apparatus, calculated from the value **ΔP**, increases with changes in the total flow rate, and starting from the value of average air speed through the apparatus V= 5 m/s, it becomes a constant value. A theoretical and experimental explanation of this is given in [11-13].

5 Conclusions

The obtained results were used for calculation of experimental and industrial models of vortex apparatuses for drying cellulose acetate, cyclone furnace for lignite combustion, dust collectors with countercurrent swirling flows for dust of asphalt-concrete plants.

Obtained experimental results on pressure losses allow calculating head losses for industrial vortex apparatuses at designing (taking into account ξ values for apparatuses of industrial diameter, which from D = 400 mm is a constant value). The calculated values of ΔP make it possible to correctly calculate the supply ducts and the required power of the electric drive of the blower device.

References

1. Planovsky A.N., Mushtaev V.I., and Ulyanov V.M., Drying Dispersed Materials in the Chemical Industry. Moscow. Chemistry, 1979. P. 288.
2. Karpov A.N. Experimental study of particle velocity and resistances in pneumatic conveying by radioactive indicator method. Izv. of Higher Education Institutions, Power Engineering. 2001. №3, pp. 75-81.
3. Sazhin B.S., Gudim L.I. Dust collectors with counter-curled flows. Chemical Industry. 1985. №8, pp. 50-54.
4. Kozulin N.A., Ershov A.I. About the solid phase effect on aerodynamics and resistance of cyclone apparatuses. Thermal Power Engineering. 1992. №1, pp. 18-20.
5. Bulkin V.A. Nikolaev N.A. Study of hydrodynamics and mass transfer in direct flow upward helical motion of gas and liquid in pipes. No. 6. 1970, pp. 898.
6. Straus V. Industrial Gas Purification. Moscow, Khimia. 1981. p. 616.
7. Yufa A.I. Mukhiddinov D.N. Dryer for dispersed materials. 1048272. USSR. Bulletin. Chemistry and Chemical Technology, № 38. 1983.
8. Sazhin B.S., Kochetov O.S., Yelin A.M., Chunaev M.V. Labor protection in textile industry. Moscow, Moscow State Technical University named after A.N. Kosygin, 2004. Pp. 433.
9. Vetoshkin A. G. Processes and apparatuses of dust cleaning. Penza, 2005. P. 210
10. Sazhin B.S., Belousov A.S. Hydrodynamic characteristics of two-phase flows in vortex devices. Moscow State Technical University after A.N. Kosygin. 2007. Pp. 93-97.
11. Kizin M.G. Calculation methods and recommendations for gas cyclones. Vladimir, 1970, p. 258.
12. Frolov E.V., Shitikov E.S. On the hydrodynamic resistance of vortex dust collectors. - In: Calculation, Design and Investigation of Machines, Devices and Chemical Production Units. Moscow State University of Chemical Engineering, 1982, pp. 49-52.
13. Galich V.N. Improving the effectiveness of centrifugal dust collectors through the use of counter swirling flows. Author's abstract. Dissertation. Cand. of Sci. M.I.: MTI 1982, p. 24.