Contribution to improvement of cleaning efficiency of pulse-jet fabric filters

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Abstract

Uniform distribution of compressed air in a blow pipe is an important consideration in the efficient use of compressed air for pulse-jet cleaning of fabric filters. It influences the consumption of compressed air, operational costs for filtration, and particle penetration through the filter. This paper reviews recent developments in this area. A formula for compressed air consumption per pressure pulse is derived, based on the pressure drop in the air vessel and polytropic change in the air state in the vessel during the pulse with the coefficient near to adiabatic change. A criterion for uniform distribution of compressed air in the blow tube is derived, based on the constant flow momentum of jets from nozzles in the blow pipe. By analysing the measured pressures along different blow tubes, a procedure for designing the diameter of the holes in the blow pipe is proposed, using only the inner diameter of the blow pipe and the number and nominal diameter of the cylindrical holes. A simplified mathematical model for the distribution of compressed air in the blow tube is described. Very good agreement between calculated and measured values has been achieved.

1 Introduction

Fabric filters are widely used in various industrial processes as an efficient means for separating suspended particles from flue gases, and to prevent them from escaping into the atmosphere. From the point of view of air pollution control, fabric filtration is regarded as the best available separation technology.
Most industrial fabric filters are bag filters with pulse-jet cleaning, known as pulse-jet fabric filters. In these filters, a short burst of compressed air, controlled by a solenoid valve, is released by the diaphragm valve. The air then flows into a blow pipe, which has orifices as outlet nozzles above each filter bag in the row. These orifices direct the jets of air axially into the upper bag opening, where a venturi mixing nozzle induces clean gas, thereby enhancing the effect of the air burst and directing the air flow along the length of the bag. The forces arising from the fabric acceleration/deceleration and the reverse air flow play a role in releasing the dust cake and cleaning the fabric.

Important considerations in the proper use of a filtering area are the uniform distribution of the dirty gas in the filter inlet into the individual bags, and the uniform distribution of compressed air for pulse-jet cleaning. If these considerations are not met, there will be a local decrease in fabric and dust cake permeability and redistribution of the dirty gas to a part of the filter with higher permeability. This results in less efficient use of the filtering area and an increase in the filter pressure drop. This increase in the pressure drop will reduce the lifetime of the filter bags, increase the energy consumption for fan operation, increase the compressed air consumption, and thus increase the operational costs for particle separation. More frequent pulse-jet cleaning also causes an increase in particle penetration through the filter [1]. Incorrect filter design directly or indirectly has a negative effect on the environment, namely on air pollution.

This paper deals with the uniform distribution of compressed air for pulse-jet cleaning and the determination of compressed air consumption. Little attention has been given to these phenomena in the literature.

2 Determining compressed air consumption from the pressure drop in a compressed air vessel

A specific compressed air consumption for pulse-jet cleaning is one of the most important quantities in evaluating the operation of pulse-jet filters.

Assuming that the supply of compressed air into the vessel is closed, the absolute pressure in the vessel drops during the opening of the diaphragm valve from the initial value $P_{ves,0}$ to the terminal value $P_{ves,1}$. On the experimental filter built in the laboratory of the Department of Environmental Engineering of the Czech Technical University in Prague it was found that the change in the air state during the outflow of compressed air from the vessel can be considered as a polytropic change with the polytropic coefficient $n = 1.37$. This change of air state is due to the short opening time of the diaphragm valve close to the adiabatic change with the coefficient $\kappa = 1.40$.

The terminal air temperature in the vessel $T_{ves,1}(K)$ can thus be determined from the equation for polytropic change

$$\left(\frac{P_{ves,1}}{P_{ves,0}}\right)^{\frac{n-1}{n}} = \frac{T_{ves,1}}{T_{ves,0}}$$

(1)
Air Pollution X

The consumption of compressed air for one cleaning pulse can be determined as the difference of the initial and terminal air mass in the vessel, and leads to an equation in the form

$$\Delta M = \frac{V_{ves}}{r T_{lab}} \left(1 - P_{ves,0}^{\frac{1}{n}} P_{ves,1}^{\frac{1}{n}}\right)$$  \hspace{1cm} (2)

where $V_{ves}$ (m$^3$) is the volume of the air vessel and $r$ is the individual gas constant of the air, $r = 287.11$ J/kgK.

The terminal thermodynamic temperature in the vessel $T_{ves,1}$ reaches low values. For instance, if the solenoid valve opening time is set to a value of $\Delta t_{el} = 50$ ms and the pressure in the vessel changes from $P_{ves,0} = 7.10^5$ Pa to $P_{ves,1} = 5.10^5$ Pa, the terminal temperature is $T_{ves,1} = 268$ K, thus $t_{ves,1} = -5$ °C (for a laboratory temperature of 20 °C). With increasing time $\Delta t_{el}$, the terminal temperature $T_{ves,1}$ decreases.

3 Criterion for uniform distribution of compressed air in the blow pipe

The most important part of a pulse-jet filter cleaning device is the venturi mixing nozzle built in the upper bag opening, where the energy of the primary pulse jet and the energy of the induced gas are converted to pressure. The venturi nozzle in fact plays the role of a mixing ejector.

The theory of mixing ejectors for steady flows comes from applying the impulse theorem. The decisive influence on the function of the mixing ejector thus comes from the momentum of the primary jet flow of compressed air. Although pulse-jet cleaning is evidently an unsteady process, we can assume that the flow momentum of the jets from the orifices of the blow pipe is the decisive criterion for uniform distribution of compressed air in the blow pipe.

For most of the time of the pressure pulse in the blow pipe, the flow from the orifices is critical and the flow momentum $H_i$ (kg.m/s$^2$) of the flow in the $i$-th orifice is

$$H_i = M_{max,i} w_{crit,i}$$  \hspace{1cm} (3)

where $w_{crit,i}$ (m/s) is the critical velocity and $M_{max,i}$ (kg/s) is the maximum mass flow through the orifice. If the conditions for critical flow are fulfilled, $(P/P_0)_i \leq 0.528$, the maximum mass flow $M_{max,i}$ (kg/s) flows through the $i$-th orifice. For $M_{max,i}$ we can write

$$M_{max,i} = S_i \mu_i \Psi_{max} P_{0,i} \sqrt{\frac{2}{r T_{0,i}}}$$  \hspace{1cm} (4)
where $S_i (m^2)$ is the orifice cross section, $\mu_i (1)$ the coefficient of flow contraction ($\mu_i = 0,740$ for a cylindrical hole), $\Psi_{\text{max},i} (1)$ the flow coefficient (for air $\Psi_{\text{max},i} = 0,484$), $P_{0,i} (\text{Pa})$ the stagnation pressure and $T_{0,i} (\text{K})$ the stagnation temperature.

The critical velocity $w_{\text{crit},i}$ is given by the equation

$$w_{\text{crit},i} = \sqrt{\frac{2}{\kappa} \frac{r}{\kappa+1} T_{0,i}} \quad (5)$$

where $\kappa (1)$ is the isoentropic exponent, $\kappa = 1,40$ for air.

Experiments on the laboratory filter have shown that, due to the geometric configuration of the blow pipe, where the cylindrical holes in the tube wall serve as outlet orifices, the static pressure in the tube in place of the orifice can be considered as the stagnation pressure $P_{0,i}$.

Assuming that there is only one stagnation temperature $T_0$ in the blow pipe, $w_{\text{crit},i}$ is according to eqn (5) constant for all orifices. The condition of constant flow momentum $H_i = \text{const}$ can then be simplified to the condition $M_{\text{max}} = \text{const}$. From eqn (4) it thus follows that

$$S_i \mu_i P_{0,i} = \text{const} \quad (6)$$

This equation is considered as the criterion for evaluating the uniform distribution of pressurised air in the blow pipe for pulse-jet cleaning of the bags in one row.

With respect to the time dependence of the static pressure in the blow pipe, the time average value of the static pressure for the period of critical flow in the $i$-th orifice is used as the stagnation pressure $P_{0,i}$ in eqn (6).

For other calculations, the stagnation temperature $T_0$ in the blow pipe is determined as the mean value of the initial and terminal air temperature in the vessel, thus

$$T_0 = \frac{T_{\text{ver},0} + T_{\text{ver},1}}{2} \quad (7)$$

The distribution of the compressed air in the blow pipe was investigated in the cleaning device of the experimental filter – Figure 1. Compressed air flows from the air vessel through the 6/4" diaphragm valve into the 6/4" blow pipe with an inner diameter of $d_T = 42$ mm, where 10 cylindrical holes are drilled, marked as R (reference), 1, 2, ..., 9. The pitch of the holes corresponds to the pitch of a real industrial bag filter with 10 bags in one row.

The overpressures inside the air vessel and in the blow pipe are measured by means of DMP 331 pressure sensors (f. BD Sensors) with a range of 0-10.10\(^5\) Pa (0-10 bar). In the blow pipe the static overpressures are measured in the wall just opposite the cylindrical holes. All measured data, including the solenoid valve opening time $\Delta t_{\text{val}}$, are recorded and processed in the measuring and control unit, based on the PC and corresponding A/D and D/A programmable...
Figure 1: Scheme of the cleaning device of the experimental filter.

Several blow pipes with three different nominal orifices with a diameter of $d_{\text{nom}} = 12, 10$ and $8 \text{ mm}$ were tested in this way. As an example, the time dependence of overpressures $p$ (bar) in place of four orifices of the "type A" blow pipe are shown in Figure 2. In the "type A" blow pipe the first two orifices (R and No. 1) have a diameter of $13 \text{ mm}$, the next six orifices (No. 2 - 7) $12 \text{ mm}$ (nominal diameter $d_{\text{nom}}$), and the last two orifices (No. 8 and 9) $11 \text{ mm}$.

Figure 2: Time dependence of overpressures in place of orifices No. 1, 3, 4 and 7 of a "type A" blow pipe.
The figure shows clearly the unsteady character of the flow and the tendency for the pressure to increase along the length of the blow pipe. The measured data of different blow pipes with ten orifices were analysed, and the procedure for designing the diameter of the orifices, meeting the condition in eqn (6), was derived.

If \( k_s \) (1) is the shape coefficient of the blow pipe

\[
k_s = \frac{10d_{nom}^2}{d_r^2}
\]

then in the range of \( k_s \in (0,3; 1) \) we can write for the ratio of the cross sections of the last and the first orifice \( S_9 / S_R \),

\[
\frac{S_9}{S_R} = 0,969 - 0,1k_s
\]

From the ratio of \( S_9 / S_R \), expressed by means of nominal diameter \( d_{nom} \) and difference \( \Delta d \), it follows that

\[
\Delta d = d_{nom} \frac{1 - \left( \frac{S_9}{S_R} \right)^{\frac{1}{2}}}{1 + \left( \frac{S_9}{S_R} \right)^{\frac{1}{2}}}
\]

and the individual diameters of the orifices are then

\[
\begin{align*}
    d_R &= d_1 = d_{nom} + \Delta d \\
    d_2 &= d_3 = \cdots = d_7 = d_{nom} \\
    d_8 &= d_9 = d_{nom} - \Delta d
\end{align*}
\]

For the used pipe flow with \( d_r = 42 \text{ mm} \) and \( d_{nom} = 12 \text{ mm} \) there is, for example, \( k_s = 0,8163 \), \( S_9 / S_R = 0,887 \) and \( \Delta d = 0,36 \text{ mm} \equiv 0,4 \text{ mm} \) and the ideal blow pipe with uniform distribution of compressed air, according to the condition in eqn (6), should be drilled as follows: \( d_R = d_1 = 12,4 \text{ mm} \), \( d_2 + d_7 = 12 \text{ mm} \) and \( d_8 = d_9 = 11,6 \text{ mm} \).

4 Mathematical model of the flow in the blow pipe

A simplified solution of the airflow through the blow pipe was described in [2], where only peak pulse pressures along the blow pipe were solved. The model
described here makes use of the model in [2] and solves the airflow in the blow pipe during the whole cleaning pulse, apart from the transitions at the beginning and at the end of the pulse [3]. The total air mass flowing out of the blow pipe is also calculated.

The model results from Bernoulli’s equation and the continuity equation. A scheme of the blow pipe with the compressed air vessel, including the points where the air state conditions are calculated, is shown in Figure 3.

![Figure 3: Scheme of the blow pipe with the compressed air vessel and the points where the air state conditions are calculated](image)

The expansion during the air flow from the vessel into the pipe is polytropic with the coefficient $n = 1.37$, near to adiabatic expansion. At points $R', 1', 2', 9'$, which are just behind the outflow nozzles in the blow pipe, barometric pressure $p_b$ (Pa) is assumed.

Integration of the general form of Bernoulli’s equation yields the equation, valid for all nozzles in the blow pipe,

$$
\frac{n}{n-1} \frac{P_{ves}}{\rho_{ves}} \left[ \left( \frac{P_b}{P_{ves}} \right)^{\frac{n-1}{n}} - 1 \right] + \left( 1 + \zeta_o \right) \frac{w_i^2}{2} + \zeta_v \frac{w_i^2}{2} + \sum_{i=1}^{i=9} \zeta_p \frac{w_i^2}{2} = 0 ,
$$

(12)

In this equation $w$ (m/s) is gas velocity, $P_{ves}$ (Pa) and $\rho_{ves}$ (kg/m$^3$) are absolute pressure and density of air in the vessel. Coefficient of losses $\zeta_o$ (1) expresses the losses when air flows through the diaphragm valve and the first part of the pipe until the reference point $R$. Coefficient $\zeta_v$ (1) represents the pressure loss due to flow division into the nozzle, flow through the cylindrical nozzle including expansion at the outflow from the nozzle. Coefficient $\zeta_p$ (1) represents the losses due to flow division in the straight direction and the friction in the pipe between adjacent nozzles.

The continuity equation is modified into a form valid for all nozzles, where the difference in density between points before and after the nozzle is neglected.
The relative cross sections $s_i = \mu S_i / A$ have been introduced, where $\mu$ (1) is the coefficient of contraction, $S_i$ ($m^2$) the cross section of the individual nozzles, and $A$ ($m^2$) is the cross section of the blow pipe.

Equations (12) and (13) written for individual nozzles form a set of equations from which the air velocities $w_i$ in selected points of the pipe can be calculated. The values of absolute static pressure at selected points in the pipe $P_i$ are calculated from Bernoulli's equation and the equation of polytropic change between individual points in the pipe. The air state condition in the vessel is the initial state. For pressure $P_R$ at point R the following relation is derived

$$P_R = P_{ves} \left[ 1 - \frac{\left(1 + \zeta_v \right) w_R^2}{2 \frac{n}{n-1} \frac{P_{ves}}{P_{ves}}} \right]^{\frac{n}{n-1}}$$ \hspace{1cm} \text{(14)}$$

The pressures at the other points result from the pressure at the previous point in the pipe, and are calculated from the formula

$$P_{i+1} = P_i \left[ 1 + \frac{\left(1 + \zeta_P \right) w_{i+1}^2}{2 \frac{n}{n-1} \frac{P_{i}}{P_{i}}} \right]^{\frac{n}{n-1}}, \quad i = R, 1, 2, \ldots,$$ \hspace{1cm} \text{(15)}$$

By means of equations (12)-(15) the static pressures $P_i$ and the velocities along the distribution pipe $w_i$ are determined in one time moment, corresponding to the state condition in the vessel $(P_{ves}, \rho_{ves})$. In order to determine the air parameters in the next time moment (after a given period), the air state condition in the vessel after a given period has to be known.

The total air mass $\Delta M$, (kg) flowing out from the pipe during a given short time interval $\Delta t$ (s) can be calculated. This air mass is expressed as the sum of the air masses flowing out from the individual nozzles. The air flows through the individual nozzles are calculated as the critical outflow through a convergent nozzle – eqn (4). Since the short transitions at the beginning and at the end of the pulse are neglected, the conditions for critical outflow are valid during the whole considered pulse duration.

Based on calculated air mass $\Delta M$, the new static pressure $P_{t+\Delta t}$ in the vessel after time interval $\Delta t$ can be calculated from the modified relation for compressed air consumption during one pressure pulse, eqn (2)
From this new value for the pressure in the vessel the new velocities and static pressures along the blow pipe are determined again. This procedure is repeated to get the time dependence of the individual air parameters during the whole pressure pulse.

The total air mass flowing out of the blow pipe during the whole pressure pulse is determined as the sum of the individual air masses $\Delta M_i$ flowing out at individual elementary time intervals.

The calculated and measured time dependence of the static pressures in the vessel and in the blow pipe were compared and are in very good agreement. For the “type A” blow pipe the values of the loss coefficients $\zeta_v = 2.45$, $\zeta_o = 3.0$ and $\zeta_p = 0.08$ were used in the calculations.

Although the used mathematical model is rather simplified, it gives a good estimation of the state condition in the blow tube and enables us to estimate if the condition of uniform distribution of compressed air in the blow tube will be met.

Conclusions

The paper presents the results of research on an aspect of the uniform distribution of compressed air in the blow pipe, which is an important consideration for the proper use of the filtering area of a pulse-jet fabric filter.

The criterion for evaluating the uniform distribution of compressed air in the blow pipe is derived, based on the constant flow momentum of the jets from the nozzles in the blow pipe.

A procedure for designing the diameter of the cylindrical holes in the blow pipe is proposed. This calculation needs only information about the inner diameter of the blow pipe and the number and nominal diameter of the holes.

A simplified mathematical model of the flow in the blow pipe is proposed. The model results from Bernoulli's equation and the continuity equation, and is in good agreement with the experimental data. The model gives us a good estimation of the state condition in the blow pipe, and enables us to estimate whether the condition of uniform distribution of compressed air in the blow pipe will be met.

Assuming that the supply of compressed air into the vessel is closed, a formula for compressed air consumption per pressure pulse is derived. The calculation is based on the pressure drop in the vessel and the polytropic change with the exponent $n = 1.37$ in the air state in the vessel during the outflow of compressed air.

These results contribute to improving the cleaning efficiency of pulse-jet fabric filters and directly and indirectly have a positive effect on the environment, namely by reducing air pollution.
References

