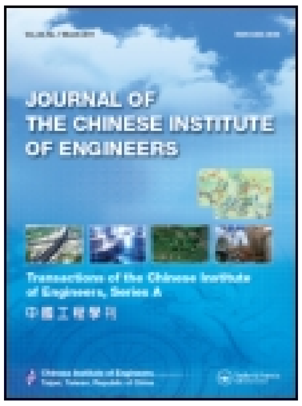


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# Numerical and experimental studies to predict properties of gas discharged from a number of nozzles on a blow pipe

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# NUMERICAL AND EXPERIMENTAL STUDIES TO PREDICT PROPERTIES OF GAS DISCHARGED FROM A NUMBER OF NOZZLES ON A BLOW PIPE

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***Key Words:*** pulse-jet bag filters, cleaning, compressible flow.




## ABSTRACT

This study numerically and experimentally investigates the properties of pressure, temperature and mass flow rate of air discharged from a number of nozzles drilled on a blow pipe. In addition to deriving a set of equations to express the physical phenomena, this study also uses measurement equipment to measure the indispensable data used in the above equations. Consequently, the theoretical and experimental works complement each other. According to the results, the mass flow rate of the air pulse discharged from a nozzle depends on the position and diameter of the nozzle on the blow pipe. Furthermore, the direction of the discharged air pulse is not perpendicular to the blow pipe and has an inclined angle which also depends on the position of the nozzle on the blow pipe. Employing the model proposed herein, a uniform mass flow rate discharged from each nozzle can be obtained by adjusting the sizes of the nozzles on the blow pipe.


## I. INTRODUCTION

Pulse-jet bag filters have found extensive industrial applications, particularly in separating fine dust from a dust laden gas stream. The mechanisms of cleaning systems have been experimentally studied as well (Bouilliez, 1986; De Ravin *et al.*, 1988; Hajek and Peukert, 1996; Morris, 1984; Saad, 1984; Sievert and Loffler, 1987). According to those results, some important parameters, e.g. the volume of the reservoir, pressure of the air in the reservoir, valve flow coefficient of the diaphragm valve, the size of the blow pipe as well as the size and number of the nozzles on the blow pipe, heavily influence cleaning performance.

Figure 1 illustrates a pulse-jet cleaning system, consisting primarily of an air reservoir, connecting pipe, diaphragm valve, blow pipe and a number of nozzles drilled in the blow pipe. As the cleaning process is initiated, the compressed air is immediately discharged from the reservoir, via the diaphragm valve, which is controlled by a solenoid valve, into the blow pipe. The pressurized blow pipe then discharges the high pressure air through the numerous nozzles drilled in the blow pipe into the corresponding bag filters, which are generally made of flexible fiber. The discharged air pulse not only inflates the bag filter abruptly, but also penetrates from the inner to outer surface of the bag filter. By doing so, the dust deposited on the outer surface of the bag filter is effectively removed. Based on the discharge processes mentioned above, this study

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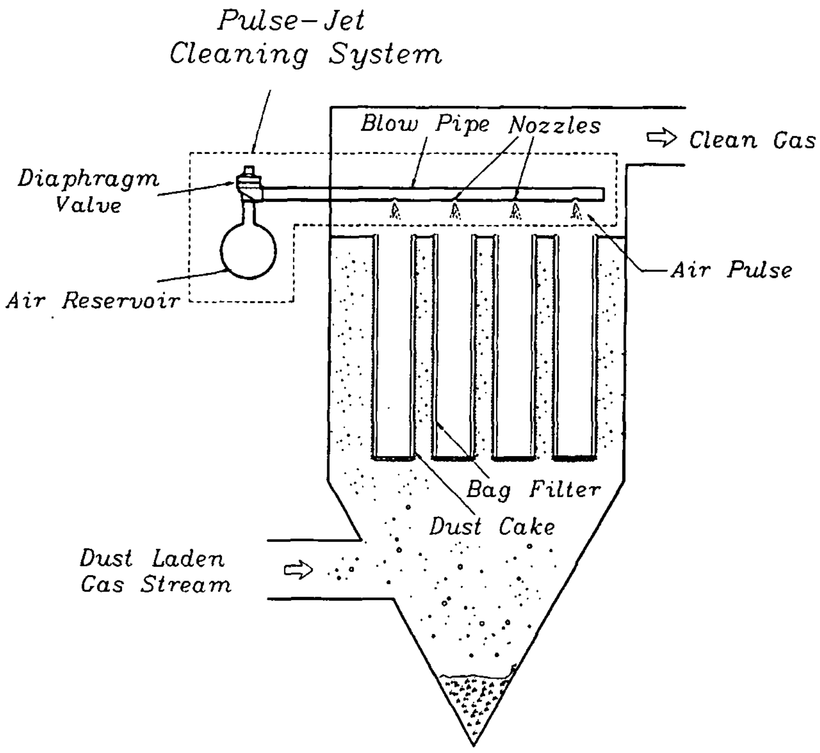
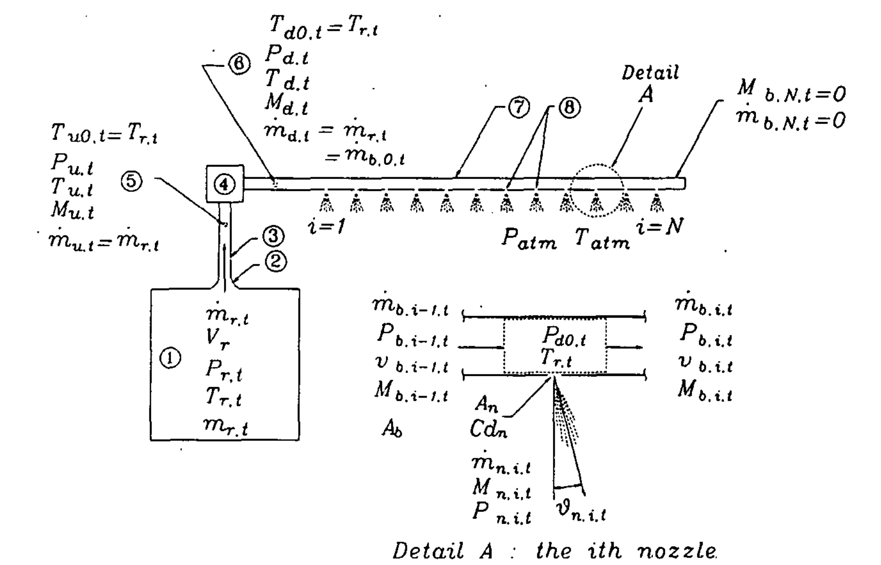



Fig. 1. An Illustration of a Pulse-Jet Cleaning System.




Descriptions:

① Air Reservoir
② Exit of the Air Reservoir
③ Connecting Pipe
④ Diaphragm Valve
⑤ Upstream of the diaphragm valve
⑥ Downstream of the diaphragm valve
⑦ Blow Pipe
⑧ Nozzles

Fig. 2. Physical Model.

examines the related properties which significantly affect cleaning efficiency, including pressure, temperature, mass flow rate and direction of the air pulse jetting into the bag filters. As is generally known, the discharge processes are transient processes and no available instrument can be used to directly and accurately measure the above properties of mass flow rate and temperature of the air pulse. Therefore, the feasibility of developing an appropriate model to analyze the above problem has seldom been explored.

Our recent study proposed a two-reservoir model (Fu and Ger, 1998). That model can predict related properties including pressure, temperature and mass flow rate of air discharged from a reservoir into a blow pipe. In light of the above developments, this study numerically and experimentally investigates the mechanisms of air flowing through a blow pipe and jetting into the atmosphere from a number of nozzles in the blow pipe. The theoretical analysis consists of several equations to express the physical phenomena. Correspondingly, the experimental work consists of a set of equipment to measure the indispensable data used in the above equations and the results from trials. The numerical results are compared as well. That comparison reveals that the mass flow rates of the air pulses discharged from nozzles are not uniform and the directions of the air pulses are not perpendicular to the blow pipe. Moreover, the position of the nozzle on the blow pipe and the size of the nozzle markedly affect the mass flow rate and the direction of an air pulse discharged from a nozzle. Furthermore, applying the model proposed herein allows us to obtain a uniform flow rate from nozzles, which is useful for industrial applications.

## II. MODELING

Figure 2 schematically depicts the physical model. Based on our previous results (Fu and Ger, 1997, 1998), the air properties at the exit of the air reservoir ($P_{r,t}$, $T_{r,t}$ and $\dot{m}_{r,t}$), upstream of the diaphragm valve ($P_{u,t}$, $T_{u,t}$ and $M_{u,t}$) and downstream of the diaphragm valve ($P_{d,t}$, $T_{d,t}$ and $M_{d,t}$) can be accurately calculated. In turn, the properties mentioned above are known in advance relative to the present study. Herein, the mechanisms of the air discharged from the blow pipe through the nozzles into atmospheric environment are investigated in detail.

According to Fig. 2, air having the properties of $P_{d,t}$, $T_{d,t}$, $M_{d,t}$ and $\dot{m}_{r,t}$ flows into the blow pipe from the left side and jets into the atmosphere through nozzles along the blow pipe. The right side of the blow pipe is closed. Since the air has momentum in the blow pipe, the discharged air pulse into the atmosphere is no longer perpendicular to the axial direction of the blow pipe and has an angle of inclination to the line of the nozzle.

"Detail A" in Fig. 2 illustrates the physical model of the air pulse discharged from the $i$th nozzle. Subscripts $b$ and $n$ denote the conditions of the blow pipe and nozzle, respectively. The mass flow rate, pressure, velocity and Mach number of the air in the blow pipe upstream of the $i$th nozzle are $\dot{m}_{b,i-1,t}$, $P_{b,i-1,t}$, $v_{b,i-1,t}$ and $M_{b,i-1,t}$, respectively. In addition, the mass flow rate, pressure, velocity and Mach number of the air in the blow pipe downstream of the $i$the nozzle are $\dot{m}_{b,i,t}$, $P_{b,i,t}$, $v_{b,i,t}$ and $M_{b,i,t}$, respectively. Moreover, the mass flow rate, pressure, Mach number and angle of inclination of the air pulse at the exit of the $i$th nozzle are $\dot{m}_{n,i,t}$, $P_{n,i,t}$, $M_{n,i,t}$ and $\vartheta_{n,i,t}$, respectively. The discharge coefficient of the nozzle is $Cd_n$. The stagnation properties of the air in the $i$th control volume are $T_{r,t}$ and $P_{d0,t}$, respectively.

Since the volume of the blow pipe is about 2.4% of the air reservoir and the process is assumed as adiabatic, the stagnation temperature of the air flow along the blow pipe and the mass flow rate discharged from the nozzles are estimated to be higher than those ($T_{r,t}$ and $\dot{m}_{r,t}$) at the exit of the reservoir simultaneously by 1.0% and 3.0%, respectively. As assumed herein, the stagnation temperature of the flow along the blow pipe and the total mass flow rate discharged from nozzles are the same as those at the exit of the air reservoir. Under this circumstance, the process can also be assumed to be quasi-steady. Furthermore, the pressure drop induced by the surface of the blow pipe compared with the pressure drop induced by the friction effects of the diaphragm valve is negligible. The stagnation pressure of the air along the blow pipe is regarded as the same as the stagnation pressure $P_{d0,t}$ downstream of the diaphragm valve.

The following assumptions are made to facilitate analysis:

1. The cross-sectional area of the nozzle is significantly smaller than that of the blow pipe and the flow in the blow pipe is regarded as a one-dimensional flow;
2. The air discharged from the air reservoir to the atmosphere takes a relatively short time and the process is regarded as adiabatic;
3. The pressure drop induced by the surface of the blow pipe is negligible; and
4. All of the nozzles in the blow pipe, of the same diameter have the same discharge coefficient $Cd_n$ which is equal to the ratio of the mass flow rate discharged from the nozzle to the mass flow rate of an isentropic nozzle of the same size.

Based on the above assumptions, the continuity equation and the momentum equation in the control volume of the blow pipe shown in Fig. 2 can be expressed as the following equations, respectively.

$$\dot{m}_{b,i-1,t} = \dot{m}_{b,i,t} + \dot{m}_{n,i,t} \tag{1}$$

$$P_{b,i-1,t}A_b + \dot{m}_{b,i-1,t}v_{b,i-1,t} = P_{b,i,t}A_b + \dot{m}_{b,i,t}v_{b,i,t} + \dot{m}_{n,i,t}v_{n,i,t}\sin(\theta_{n,i,t}) \tag{2}$$

Where subscript $i$ denotes the position of the nozzle on the blow pipe. The smaller value of $i$ implies a closer position of the nozzle to the diaphragm valve. For $i$=1, $\dot{m}_{b,0,t}$ equals $\dot{m}_{r,t}$. For $i$=$N$, $\dot{m}_{b,N,t}$ equals zero. According to Saad (1993), the mass flow rate $\dot{m}_{n,i,t}$ of the air discharged from the $i$th nozzle can be modified as:

$$\dot{m}_{n,i,t} = \frac{P_{d0,t}}{\sqrt{T_{r,t}}}\sqrt{\frac{\gamma}{R}}\,Cd_nA_n\cos\theta_{n,i,t}\frac{M_{n,i,t}}{(1+\frac{\gamma-1}{2}M_{n,i,t}^{\ 2})^{\frac{\gamma+1}{2(\gamma-1)}}} \tag{3}$$

where $P_{d0,t}$ and $T_{r,t}$ can be determined from a previous study (Fu and Ger, 1998). In addition, the Mach number $M_{n,i,t}$ of an isentropic flow can be determined by solving the following equation.

$$\frac{P_{d0,t}}{P_{atm}} = (1+\frac{\gamma-1}{2}M_{n,i,t})^{\frac{\gamma+1}{\gamma-1}} \tag{4}$$

The mass flow rate $\dot{m}_{b,i,t}$, static pressure $P_{b,i,t}$ and the velocity $v_{b,i,t}$ can be expressed in terms of $M_{b,i,t}$ as Eqs. (5), (6) and (7), respectively (Saad, 1993).

$$\dot{m}_{b,i,t} = \frac{P_{d0,t}}{\sqrt{T_{r,t}}}\sqrt{\frac{\gamma}{R}}\,A\frac{M_{b,i,t}}{(1+\frac{\gamma-2}{2}M_{b,i,t})^{\frac{\gamma+1}{2(\gamma-1)}}} \tag{5}$$

$$P_{b,i,t} = \frac{P_{d0,t}}{(1+\frac{\gamma-1}{2}M_{b,i,t})^{\frac{\gamma+1}{\gamma-1}}} \tag{6}$$

$$v_{b,i,t} = \sqrt{\gamma R\frac{T_{r,t}}{(1+\frac{\gamma-1}{2}M_{b,i,t})}} \tag{7}$$

Therefore, Eqs. (1) and (2) can be rewritten in terms of Mach numbers as:

$$\frac{M_{b,i-1,t}^{\ 2}}{(1+\frac{\gamma-1}{2}M_{b,i-1,t}^{\ 2})^{\frac{\gamma}{\gamma-1}}} = \frac{M_{b,i,t}^{\ 2}}{(1+\frac{\gamma-1}{2}M_{b,i,t}^{\ 2})^{\frac{\gamma}{\gamma-1}}} + \frac{A_nCd_n\cos(\theta_{n,i,t})}{A_b}\frac{M_{n,i,t}^{\ 2}}{(1+\frac{\gamma-1}{2}M_{n,i,t}^{\ 2})^{\frac{\gamma}{\gamma-1}}} \tag{8}$$

and

$$\frac{(1+\gamma)M_{b,i-1,t}^{\ 2}}{(1+\frac{\gamma-1}{2}M_{b,i-1,t}^{\ 2})^{\frac{\gamma}{\gamma-1}}} = \frac{(1+\gamma)M_{b,i,t}^{\ 2}}{(1+\frac{\gamma-1}{2}M_{b,i,t}^{\ 2})^{\frac{\gamma}{\gamma-1}}} + \frac{A_nCd_n\cos(\theta_{n,i,t})\sin(\theta_{n,i,t})}{A_b}\frac{\gamma M_{n,i,t}^{\ 2}}{(1+\frac{\gamma-1}{2}M_{n,i,t}^{\ 2})^{\frac{\gamma}{\gamma-1}}} \tag{9}$$

For the first nozzle downstream of the diaphragm valve, i.e. $i$=1, the mass flow rate of $\dot{m}_{b,0,t}$ equals the mass flow rate of $\dot{m}_{r,t}$, and the Mach number of $M_{b,0,t}$ can be determined by solving Eq. (5). Therefore, for a given value of $Cd_n$, the unknowns of $M_{b,1,t}$ and $\theta_{b,1,t}$ in Eqs. (8) and (9) can be determined by solving these two equations. The obtained Mach number of $M_{b,1,t}$ is sequentially used to determine the values of $M_{b,2,t}$ and $\theta_{b,2,t}$. By utilizing the same method, all of the Mach numbers $M_{b,i,t}$ and angles of inclination $\theta_{b,i,t}$ from $i$=1 to $N$ can be obtained. However, for $i$=$N$, the mass flow rate $\dot{m}_{b,N,t}$ at the end of the blow pipe equals zero, and the Mach number $M_{b,N,t}$ also equals zero. To satisfy the above condition, the value of $Cd_n$ is determined.

As for the static pressure $P_{n,i,t}$ and static temperature $T_{n,i,t}$, they can be obtained by the following equations:

$$P_{n,i,t} = \frac{P_{d0,t}}{(1+\frac{\gamma-1}{2}M_{b,i,t})^{\frac{\gamma}{\gamma-1}}} \tag{10}$$

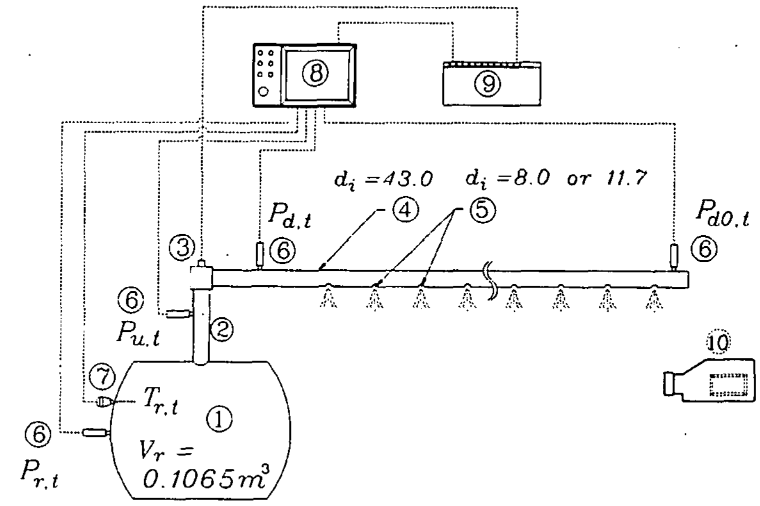
$$T_{n,i,t} = \frac{T_{r,t}}{1+\frac{\gamma-1}{2}M_{n,i,t}} \tag{11}$$

The calculation procedures for solving the above equations are summarized as follows:

(1) Calculate properties of $P_{d0,t}$, $T_{r,t}$ and $\dot{m}_{r,t}$ at the inlet of the blow pipe from previous paper (Fu and Ger, 1998);
(2) Calculate the Mach number $M_{n,i,t}$ of the air pulse at the exits of the nozzles from Eq. (4);
(3) Assume an initial value of $Cd_n$;
(4) Calculate the value of $M_{b,0,t}$ from Eq. (5);
(5) From $i$=1 to $N$, substitute the values of $M_{b,i-1,t}$ and $M_{n,i,t}$ into Eqs. (8) and (9) and then obtain the values of $M_{b,i,t}$ and $\theta_{n,i,t}$;
(6) Adjust the discharge coefficient $Cd_n$ and repeat step (4) until the condition of $|M_{b,N,t}|\leq 10^{-6}$ is satisfied; and
(7) Calculate the mass flow rate of $\dot{m}_{n,i,t}$ from Eq. (3).

## III. EXPERIMENTAL APPARATUS AND PROCEDURE

Experiments are performed to examine the feasibility of the numerical results of the proposed model. Fig. 3 illustrates the experimental apparatus. An air reservoir 1 ① with a volume of 0.1065 $m^3$ is used. Pressure $P_{r,t}$ and temperature $T_{r,t}$ of the air in the reservoir are measured by the pressure transmitter ⑥ and the thermocouple ⑦, respectively. Next, a connecting pipe ② with the diameter of 43.0 mm and length of 450 mm is used to connect the air reservoir and the diaphragm valve. The diaphragm valve ③ has a nominal diameter of 1.5 inches. Two pressure transmitters are separately installed on both sides of the diaphragm valve to measure the pressure variations of $P_{u,t}$ and $P_{d,t}$ during the discharge process. A blow pipe ④ of which the length is 1750 mm and the diameter is 43.0 mm is connected downstream of the diaphragm valve. In the blow pipe, twelve nozzles ⑤ with the diameter of $d_n$ and pitch of 100 mm are drilled perpendicular to the blow pipe. Herein, experiments using two diameters, $d_n$=8.0 mm and 11.7 mm, of nozzles are performed. Another pressure transmitter ⑥ is installed at the end of the blow pipe to measure the stagnation pressure $P_{d0,t}$ of the air in the blow pipe. Next, a programmable logic controller (PLC) ⑨ is used to generate the trigger for opening the diaphragm valve and starting the data acquisition unit ⑧. The sampling rate of the data acquisition is 500 Hz. Finally, a video camera ⑩ is used to take pictures of the air pulse during the discharge processes.




Descriptions:

① : Air Reservoir
② : Connecting Pipe (L=450)
③ : Diaphragm Valve
④ : Blow Pipe (L=1750)
⑤ : Nozzles (12 holes)
⑥ : Pressure Transmitters
⑦ : Termocouple
⑧ : Data Acquisition Unit
⑨ : Programmable Logic Controller
⑩ : Video Camera

Unit: mm

Fig. 3. Experimental apparatus.

Procedures of the experimental work are as follows:

(a) Set a certain duration $\Delta t$ of the electric pulse to be generated by the PLC. Herein, the duration $\Delta t$ equals 500 msec;
(b) Pressurize the air reservoir to the desired initial pressure $P_{r,t}$=0 and temperature $T_{r,t}$=0 ;
(c) Execute the PLC program to open the diaphragm valve and start the data acquisition unit. Meanwhile, turn on the video camera to take pictures;
(d) Measure and record the pressure variations of $P_{r,t}$, $P_{u,t}$, $P_{d,t}$ and $P_{d0,t}$ of the air in the reservoir, upstream and downstream of the diaphragm valve and at the end of the blow pipe, respectively; and
(e) Change the test conditions and repeat the above

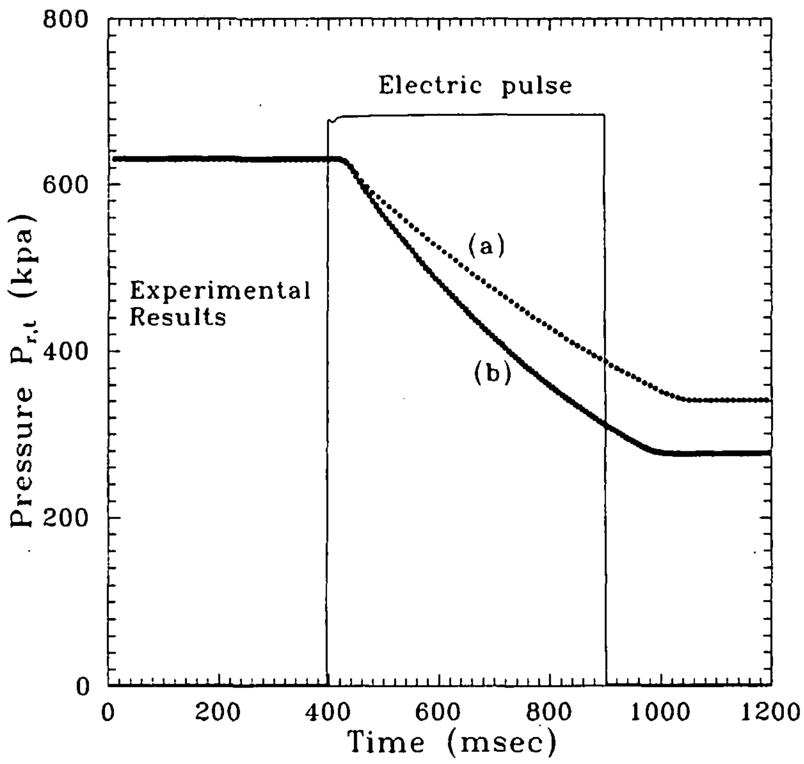



Fig. 4. The variations of the pressure of $P_{r,t}$ of the air in the reservoir; (a) $d_n$=8.0 mm, (b) $d_n$=11,7 mm.

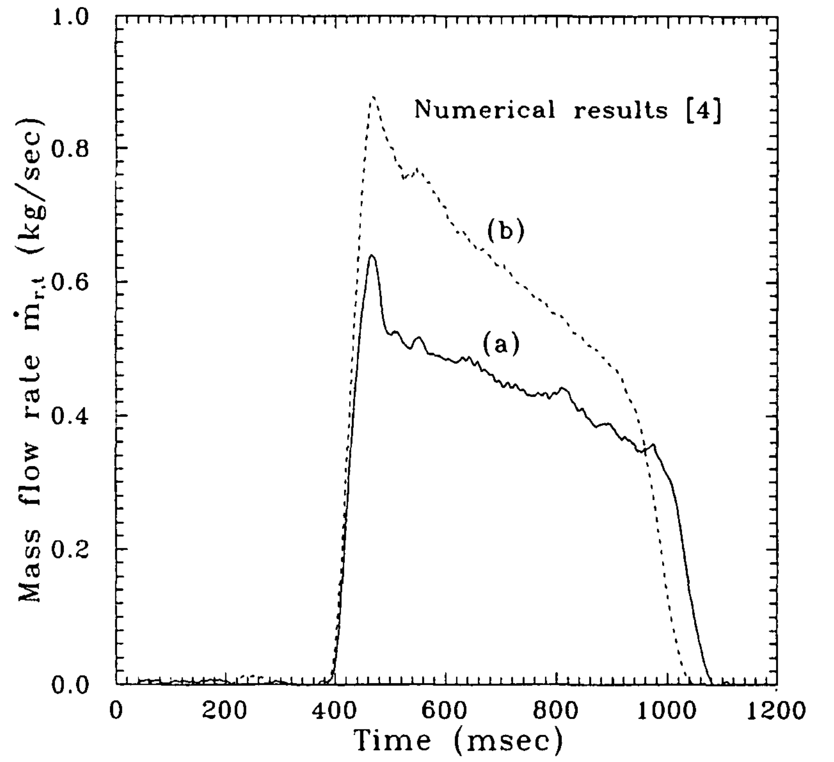



Fig. 5. The variations of the mass flow rate $\dot{m}_{r,t}$; (a) $d_n$=8.0 mm, (b) $d_n$=11,7 mm.

procedures until sufficient data are collected.

Prior to conducting the experimental work, the pressure transmitters used are calibrated by a standard pressure gauge. These pressure transmitters are made with TRANSBAR® ceramic sensing elements. The measuring range is 0 to 10 bar. The error is ±0.2% F.S. The typical response time is less than 3 msec. These pressure transmitters are calibrated by a WIKA standard pressure gauge which has a measuring range of 0 to 10 kg/cm$^2$, scale divisions of 0.05 kg/cm$^2$ and accuracy ±0.5% F.S.D. Calibration results indicate that the discrepancies between the readings of the standard pressure gauge and those of the pressure transmitters are ±0.5%.

## IV. RESULTS AND DISCUSSION

Figure 4 illustrates the measured pressure variations of $P_{r,t}$ of the air in the reservoir during the discharge process. Curves (a) and (b) are the results for $d_n$=8.0 mm and $d_n$=11.7 mm, respectively. As the diaphragm valve is opened, the discharged air decreases the pressure in the reservoir and the value of $P_{r,t}$ monotonously decreases with time. Under this circumstance, to directly measure the variations of the temperature and the mass flow rate of the air in the blow pipe is extremely difficult. Therefore, the variations of the pressure shown in Fig. 4 are used to calculate the properties of air entering the blow pipe by the methods described in our earlier studies (Fu and Ger, 1997, 1998).

Figures 5 and 6 summarize the numerical results of the mass flow rate $\dot{m}_{r,t}$ and the stagnation pressure $P_{d0,t}$ of air entering the blow pipe, respectively.

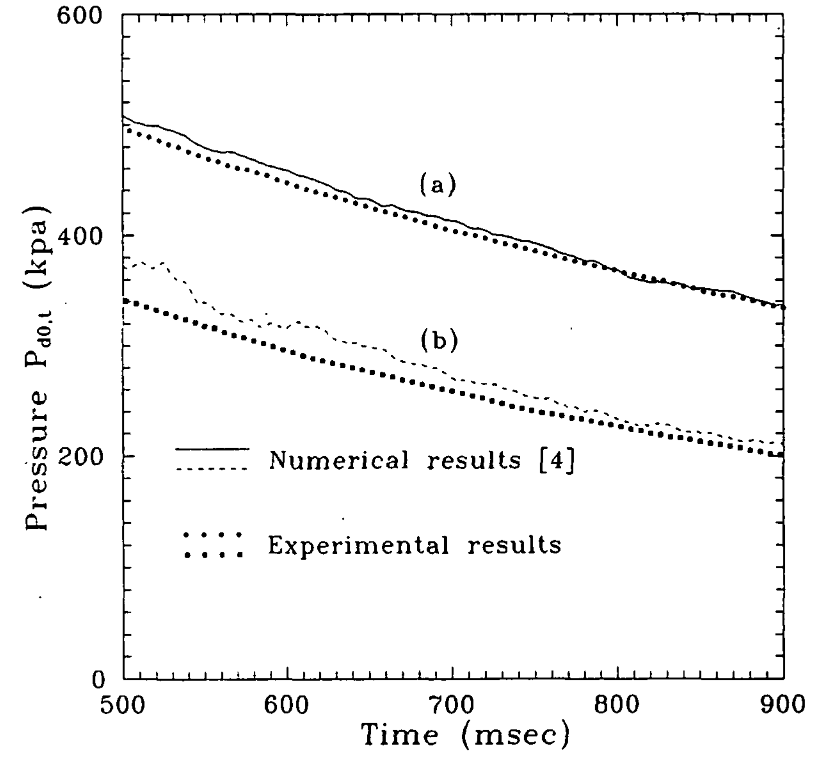



Fig. 6. The variations of the stagnation pressure $P_{d0,t}$ in the blow pipe; (a) $d_n$=8.0 mm, (b) $d_n$=11,7 mm.

Fig. 5 indicates that under the same stagnation pressure $P_{d0,t}$, the larger diameter implies a larger mass flow rate. With the increment of time, the mass flow rate decreases drastically for $d_n$=11.7 mm and slightly for $d_n$=8.0 mm. Fig. 6 displays the variations of the stagnation pressures of $P_{d0,t}$. The experimental results, which are measured at the end of the blow pipe shown in Fig. 2, agree well with the numerical ones. Therefore, applying the model proposed in (Fu and Ger, 1998) allows us to consider that the numerical results of the air properties at entrance of the blow pipe ($P_{d0,t}$, $T_{d0,t}$ and $\dot{m}_{r,t}$) are accurate and are

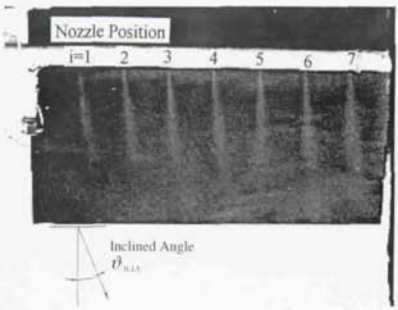


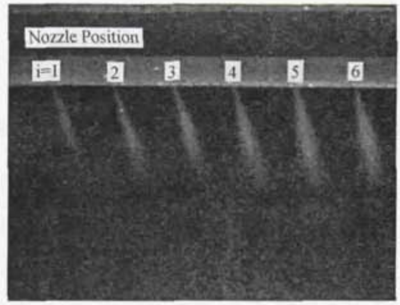

Fig. 7. Photo of the air pulse for nozzle diameter $d_n$=8.0 mm and nozzle positions $i$=1~7.

available to calculate the air properties of the air discharged from the nozzles.

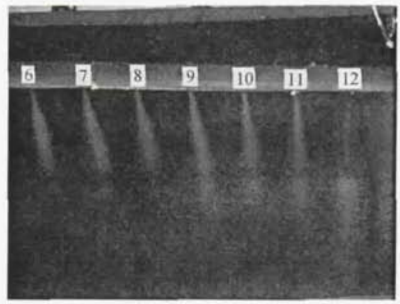
The Mach numbers $M_{n,i,t}$ at the exits of the nozzles are calculated from Eq. (4) by substituting the stagnation pressures of $P_{d0,t}$ in Fig. 6 into the equation. The calculated Mach numbers $M_{n,i,t}$ are greater than unity and reveal that the flow is choked at the exits of the nozzles during the discharge process. The fact that all the Mach numbers of $M_{n,i,t}$ are equal to unity is concluded. Under such a circumstance, the mass flow rate $\dot{m}_{n,i,t}$ is a function of the stagnation properties of $P_{d0,t}$ and $T_{r,t}$ of the air in the blow pipe and the angle of inclination, $\theta_{n,i,t}$ of the air pulse. However, the stagnation properties of the air in the blow pipe are independent of the positions of the nozzles. Therefore, in Eq. (3), the angle of inclination, $\theta_{n,i,t}$ is the only variable which causes the mass flow rates $\dot{m}_{n,i,t}$ discharged from the nozzles to differ from each other.

Figures 7 and 8 illustrate the air pulses jetting into the environment for $d_n$=8.0 mm and $d_n$=11.7 mm cases, respectively. The definition of the angle of inclination, $\theta_{n,i,t}$ in Figs. 7 and 8 is the angle between the normal line of the nozzle and the center line of the air pulse. Air discharged from the nozzle is a kind of expansion process and the temperature of the air pulse decreases abruptly, thereby condensing the vapor included in the air pulses. Consequently, slight white traces induced by the air pulses are observed beneath the nozzles.

Tables 1 and 2 list the mass flow rate $\dot{m}_{n,i,t}$ and angle of inclination, $\theta_{n,i,t}$ for nozzle diameters $d_n$=8.0 mm and $d_n$=11.7 mm, respectively. The duration of the discharge process is 0.5 seconds, i.e. from $t$=0.4 to $t$=0.9 second. The air properties of $P_{d0,t}$, $T_{d0,t}$, and $\dot{m}_{r,t}$ at $t$=0.6 second equal 458.4 kPa, 284 K and 0.489 kg/sec for $d_n$=8.0 mm and 315.5 kPa, 272 K and 0.717 kg/sec for $d_n$=11.7 mm, respectively. The




(a)




(b)

Fig. 8. Photos of the air pulse for nozzle diameter $d_n$=11.7 mm and nozzle positions : (a) $i$=1~6 , (b) $i$=7~12.

discharge coefficient $Cd_n$ for $d_n$=8.0 mm and $d_n$= 11.7 mm are 0.742 and 0.743, respectively. The discharge coefficients are close since the geometries of the nozzles closely resemble each other.

According to Tables 1 and 2, $\overline{m}_{n,i,t}$ is an average mass flow rate which is obtained from the value of $\dot{m}_{r,t}$ divided by the number of the nozzle $N$. The value of $\dot{m}_{n,i,t}$ increases with an increase of the number of $i$ and, in turn, the mass flow rate $\dot{m}_{n,i,t}$ becomes larger as the position of the nozzle more closely approaches the right end of the blow pipe. This is attributed to the fact that the momentum of the air flow gradually decreases from the left end to the right end of the blow pipe. Therefore, the angle of inclination, $\theta_{n,i,t}$ becomes smaller with an increase in the value of the number of $i$. The deviations of the experimental and numerical results of the angles of inclination are only slight. For the larger diameter ($d_n$=11.7 mm) in Table 2, the mass flow rate $\dot{m}_{n,i,t}$ and angle of inclination, $\theta_{n,i,t}$ exceed those of the smaller diameter ($d_n$=8.0 mm) in Table 1.

**Table 1. The mass flow rates $\dot{m}_{n,i,t}$ and angles of inclination, $\theta_{n,i,t}$ for nozzle diameter $d_n$=8.0 mm.**

| (1)Arrangement of nozzle | $i$=1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| (2)Mass flow rate $\dot{m}_{n,i,t}$ (kg/sec) | 0.0403 | 0.0404 | 0.0405 | 0.0406 | 0.0407 | 0.0408 | 0.0408 | 0.0409 | 0.0409 | 0.0410 | 0.0410 | 0.0410 |
| (3)Deviation $\frac{\dot{m}_{n,i,t}-\overline{\dot{m}}_{n,i,t}}{\overline{\dot{m}}_{n,i,t}}$ (%) | -1.16% | -0.85% | -0.57% | -0.33% | -0.11% | 0.08% | 0.25% | 0.38% | 0.49% | 0.57% | 0.62% | 0.65% |
| (4)Angle of inclination $\vartheta_{n,i,t}$ (deg.) | 10.76 | 9.82 | 8.88 | 7.94 | 7.00 | 6.07 | 5.13 | 4.20 | 3.26 | 2.33 | 1.40 | 0.47 |
| (5)Experimental results (Fig. 7) | 8 | 7 | 5 | 4 | 4 | 3 | 3 | | | | | |
| (6)Deviation (deg.) | -2.8 | -2.8 | -3.9 | -3.9 | -3.0 | -3.1 | -2.1 | | | | | |

**Table 2. The mass flow rates $\dot{m}_{n,i,t}$ and angles of inclination, $\theta_{n,i,t}$ for nozzle diameter $d_n$=11.7 mm.**

| (1)Arrangement of nozzle | $i$=1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| (2)Mass flow rate $\dot{m}_{n,i,t}$ (kg/sec) | 0.0560 | 0.0571 | 0.0580 | 0.0588 | 0.0595 | 0.0601 | 0.0606 | 0.0610 | 0.0613 | 0.0615 | 0.0616 | 0.0617 |
| (3)Deviation $\frac{\dot{m}_{n,i,t}-\overline{\dot{m}}_{n,i,t}}{\overline{\dot{m}}_{n,i,t}}$ (%) | -6.24% | -4.45% | -2.91% | -1.58% | -0.45% | 0.52% | 1.33% | 1.99% | 2.50% | 2.89% | 3.14% | 3.26% |
| (4)Angle of inclination $\vartheta_{n,i,t}$ (deg.) | 24.79 | 22.31 | 19.94 | 17.65 | 15.44 | 13.27 | 11.16 | 9.08 | 7.03 | 5.01 | 3.00 | 1.00 |
| (5)Experimental results (Fig. 8) | 20 | 18 | 16 | 14 | 12 | 13 | 13 | 12 | 10 | 6 | 2 | 0 |
| (6)Deviation (deg.) | -4.8 | -4.3 | -3.9 | -3.7 | -3.4 | -0.3 | 1.8 | 2.9 | 3.0 | 1.0 | -1.0 | -1.0 |

**Table 3. The diameters $d_{n,i}$ and angles of inclination, $\theta_{n,i,t}$ for mass flow rates $\dot{m}_{n,i,t}$ being uniform, $d_n$= 8.0 mm.**

| (1)Arrangement of nozzle $d_n$=8.0 mm | $i$=1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| (2)Nozzle diameter (mm) $d_{n,i}$ | 8.047 | 8.034 | 8.023 | 8.013 | 8.004 | 7.997 | 7.990 | 7.985 | 7.981 | 7.977 | 7.975 | 7.974 |
| (3)Deviation $\frac{d_{n,i}-d_n}{d_n}$ (%) | | 0.43% | 0.29% | 0.16% | 0.05% | -0.04% | -0.12% | -0.19% | -0.24% | -0.28% | -0.31% | -0.32% |
| (4)Angle of inclination $\vartheta_{n,i,t}$ (deg.) | 10.88 | 9.90 | 8.93 | 7.96 | 7.01 | 6.06 | 5.12 | 4.18 | 3.25 | 2.32 | 1.39 | 0.46 |

In many industrial applications, the mass flow rates discharged from a number of nozzles on a blow pipe must be the same, i.e. a uniform mass flow rate distribution. To obtain a uniform mass flow rate distribution, adjusting the sizes of the nozzles according to their positions is required. Such an adjustment can be made by solving the governing equations mentioned earlier to obtain different nozzle diameters $d_{n,i}$ at different positions. Notably, the uniform mass flow rate is basically equal to the total mass flow rate $\dot{m}_{r,t}$ divided by the number of nozzles $N$. Tables 3 and 4 summarize the results of the adjusted

**Table 4. The diameters $d_{n,i}$ and angles of inclination, $\theta_{n,i,t}$ for the mass flow rates $\dot{m}_{n,i,t}$ being uniform, $d_n$=11.7 mm.**

| (1) Arrangement of nozzle $d_n$=11.7 mm | $i$=1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| (2) Nozzle diameter (mm) $d_{n,i}$ | 12.08 | 11.96 | 11.86 | 11.78 | 11.72 | 11.66 | 11.62 | 11.58 | 11.55 | 11.53 | 11.52 | 11.51 |
| (3) Deviation (%) $\frac{d_{n,i}-d_n}{d_n}$ | 3.24% | 2.23% | 1.40% | 0.71% | 0.14% | -0.33% | -0.71% | -1.01% | -1.25% | -1.42% | -1.54% | -1.59% |
| (4) Angle of inclination (deg.) $\vartheta_{n,i,t}$ | 114.59 | 112.35 | 110.54 | 109.05 | 107.82 | 106.81 | 106.00 | 105.34 | 104.84 | 104.47 | 104.23 | 104.11 |

nozzle diameters $d_{n,i}$ and angles of inclination, $\theta_{n,i,t}$. The properties of the air entering the blow pipe are the same as those of Tables 1 and 2. The fact that the momentum of air flow in the blow pipe is larger near the entrance of the blow pipe accounts for why a larger nozzle diameter is required for a nozzle near the entrance of the blow pipe to discharge the same mass flow rate as other nozzles. According to Tables 3 and 4, the nozzle diameter $d_{n,i}$ varies from large to small with an increase in the number of $i$. Notably, the mass flow rate discharged from each nozzle is uniform.

## V. CONCLUSION

This study numerically and experimentally investigates the air properties of the air pulse discharged from nozzles in a blow pipe. Based on the results presented herein, the following conclusions are drawn:

1. The mass flow rate of the air pulse appears to depend on the position of the nozzle. The closer the nozzle to the diaphragm valve the lower the mass flow rate ;
2. Air pulses discharged from the nozzles are not perpendicular to the blow pipe. Also, the angle of inclination of the air pulse depends on the position of the nozzle on the blow pipe. The closer the nozzle to the diaphragm valve the larger the angle of inclination; and
3. Applying the model proposed herein allows us to obtain a uniform distribution of the mass flow rate from nozzles by adjusting the sizes of the nozzles according to their positions.

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## NOMENCLATURE

| | |
|---|---|
| $A$ | cross-sectional area, m$^2$ |
| $Cd$ | Discharge coefficient, ratio of the real mass flow rate to the mass flow rate of an isentropic flow, $=\frac{(\dot{m})_{real}}{(\dot{m})_{isen}}$, dimensionless |
| $d$ | diameter, m |
| $m$ | mass of air in a reservoir , ($=PV/RT$), kg |
| $\dot{m}$ | mass flow rate, kg/s |
| $M$ | Mach number, dimensionless |
| $P_{atm}$ | atmospheric pressure, kPa |
| $P$ | absolute pressure, kPa |
| $R$ | gas constant of air,(=287.04), m$^2$/(s$^2$ K) |
| $T$ | temperature, K |
| $V$ | volume, m$^3$ |
| $v$ | velocity, m/s |
| $\vartheta$ | angle of inclination of the air pulse, degrees |
| $\gamma$ | Ratio of specific heats, ($=c_p/c_v$), dimensionless |

Subscripts

| | |
|---|---|
| $r$ | air reservoir conditions |
| $b$ | blow pipe conditions |
| $n$ | nozzle conditions |
| $u$ | conditions upstream of the diaphragm valve |
| $d$ | conditions downstream of the diaphragm valve |
| $i$ | position of nozzle |
| $t$ | time |

## REFERENCES

1. Bouilliez, L., 1986, "Importance of Physical Parameters for the Cleaning Efficiency of a Reverse Jet-Cleaned Dust Collector," *Solids Handling Conference*, pp. B11-B25.
2. De Ravin, M., Humphries, W., and Postle, R., 1988, "A Model for the Performance of a Pulse Jet Filter," *Filtration & Separation*, Vol. 25, pp. 201-207.
3. Fu, W. S., and Ger, J. S., 1997, "A Concise

Method for Determining a Valve Flow Coefficient of a Valve under Compressible Gas Flow," submitted to *Experimental Thermal and Fluid Science*, (accepted).
4. Fu, W. S., and Ger, J. S., 1998, "A Two-Reservoir Model of a Pulse-Jet Cleaning System," submitted to *Journal of the Air & Waste Management Association*, (accepted).
5. Hajek, S., and Peukert, W., 1996, "Experimental Investigations with Ceramic High-temperature Filter Media," *Filtration & Separation*, Vol. 33, No. 1, pp. 29-37.
6. Morris, W. J., 1984, "Cleaning Mechanisms in Pulse Jet Fabric Filters," *Filtration & Separation*, Vol. 21, pp. 50-54.
7. Saad, M. A., 1993, Compressible Fluid flow; 2nd ed.; Prentice Hall: New Jersey, Chapter 3.
8. Sievert, J., and Loffler, F., 1987, "The Effect of Cleaning System Parameters on the Pressure Pulse in a Pulse-Jet Filter," *Particulate and Multiphase Processes Conference*, Vol. 2, pp. 647-662.

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# 由噴氣管上噴嘴所噴出脈衝空氣性質之研究

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**摘　要**

本文針對脈衝式袋式集塵器的清洗系統，以實驗及理論的方法，對由噴氣管上的噴嘴所噴出之空氣脈衝作一研究。研究結果顯示：由噴嘴所噴出之空氣脈衝，其方向並不垂直於噴氣管而與噴嘴的法線間存在一傾斜角；各噴嘴所噴出之空氣脈衝其質量流量亦不相同。空氣脈衝的傾斜角與質量流率與噴嘴的位置與大小有關。應用本文的數學模式，藉改變噴嘴的大小，可使每一噴嘴的質量流率皆相同。

關鍵詞：脈衝式袋式集塵器、清洗、可壓縮流。