Fluid Dynamical Characteristics of Pressure Drop of Air Flow in Two Kinds of Cylindrical Vortex Chambers for Control of Vortex Flow

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Abstract

Fluid dynamical characteristics of vortex flow are applied to many technological processes. Especially, a cyclone dust collector for the dust collection, a vortex diode for the control of flow processes by vortex flow in the vortex chamber, a vortex type dryer and also a vortex type combustor are the well-known equipments to the industries. Concerning the control of vortex flow in the cylindrical vortex chamber, the fundamental characteristics of the velocity distributions which depended on the flow conditions of the primary and secondary inlet flows to the vortex chambers were reported in detail. Then in order to estimate the energy dissipation due to vortex flow, the fluid dynamical characteristics of the pressure drop for the two kinds of the vortex chambers which were distinguished as the co-rotating flow system abbreviated by a symbol of Co-R-F-S and the counter-rotating flow system abbreviated by a symbol of C-R-F-S are experimentally studied. The former flow system intended to promote the tangential velocity of vortex flow in the vortex chamber due to the mutual sum up of the angular momentum fluxes with the same spin directions from the inlet pipes and the latter flow system intended to suppress the tangential velocity to the migration of the angular momentum fluxes with the clockwise and anti-clockwise spin directions. Therefore the pressure drops of the vortex flow for the two types of the vortex chambers showed the different characteristics deeply depended on the inlet velocity \( V_0 \) in the inlet pipe and the inlet velocity \( V_0 \) in the inlet pipe. Then the correlation between the pressure drop and the mean inlet velocities in the inlet pipe and pipe and also the correlation between the coefficient of the pressure drop and Reynolds numbers for the inlet pipe and inlet pipe were discussed in detail. Especially when the inlet velocity \( V_0 \) in the inlet pipe was equal to \( V_0 \) in the inlet pipe, the pressure drop tends to decrease sharply for the counter-rotating flow system, on the other hand the pressure drop tends to increase for the co-rotating flow system. The physical reason was closely related to the radial distributions of the tangential velocities in the vortex chamber as already described in our paper. Further based upon the experimental results of the pressure drop, the eddy viscosity estimated from the derived equation that was based upon the vorticity distribution in the vortex chamber was discussed. The above stated results were reported in detail.

Keywords: Vortex Chamber, Pressure Drop, Coefficient of Pressure Drop, Vorticity, Eddy Viscosity, Reynolds Number, Co-Rotating Flow System, Counter-Rotating Flow System

1. Introduction

There are many types of the vortex chambers which are applied to the industries. Historically a cyclone dust collector is one of the most general equipments for the collections of the solid particles and many types of dusts, however the forms and the sizes of the cyclone dust collectors corresponding to the applications are determined. Further the vortex types of the dryer and the combustor are equipped in the many chemical plants. And also a vortex diode for the control of the flow process by vortex flow in the flatten vortex chamber is well known equipment. However from the energy consumption point of view, these vortex flows in these vortex chambers necessitate high energy dissipation, so it is necessary to estimate the pressure drop of pure air flow for the design of these vortex chambers. Therefore in this paper, the pressure drop for the two kinds of the vortex chambers which are designated as a co-rotating flow system (Co-R-F-S) and a counter-rotating flow system (C-R-F-S) are constructed with the trans-

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The pressure drop where $Q_{1}$ of air flow in the vortex chamber by the connection of the two inlet pipes parent acrylic resin. The former flow system is an equipment with two inlet pipes to promote the tangential velocity of air flow in the vortex chamber by the connection of the two inlet pipes 1 and 2 tangentially to the outer wall of the vortex chamber as shown in Fig.1. Therefore the direction of rotational flow in the vortex chamber is the same, so when the both inlet velocities are the same, the tangential velocity increases about two times higher than the mean inlet velocity $V_{01}$ (m/s) in the inlet pipe 1 and the mean inlet velocity $V_{02}$ (m/s) in the inlet pipe 2. Consequently the pressure drop $\Delta P_{c}$ (Pa) becomes to increase larger. On the other hand the latter flow system is an equipment connected tangentially the two inlet pipes to prevent the promotion of the tangential velocity of air flow in that by the connection of the two inlet pipes 1 and 2 tangentially as shown in Fig. 2. Therefore when the both inlet velocities are the same, the tangential velocity near the vortex core region at the middle plane in the vortex chamber becomes to decrease sharply, but near the outer wall region at the two inlet pipes, the rotational flows due to the conservations of the angular momentum fluxes are maintained. The tangential velocity $V_{0c}$ (m/s) estimated by the conservation of the angular momentum fluxes near the outer wall region will be described briefly in appendix. However described in reference 11, the mixing processes due to the radial transfer of the angular momentum fluxes require the response time to the transient flow with the large amplitude of the disturbance velocity. Consequently the pressure drop $\Delta P_{c}$ (Pa) becomes to decrease\(^{11, 12}\). Further in order to make clear the fluid dynamical characteristics of the pressure drop, the theoretical consideration of the pressure drop basing upon the vorticity distribution is discussed.

The diameter of the vortex chamber is $D_{1} = 2 \times R_{1} = 150$ mm, the diameters of the two inlet pipes are $D_{o} = 2 \times R_{o} = 24$ mm, the diameters of the two exit pipes are $D_{2} = 2 \times R_{o} = 50$ mm and the imaginary cylindrical length is $H_{i} = 150$ mm. The Reynolds numbers $Re_{1}$ and $Re_{2}$ are defined as follows;

\[
Re_{1} = \frac{Q_{1}}{H_{i} \cdot \nu}, \quad Q_{1} = \frac{\pi \cdot D_{o}^{2}}{4} \cdot V_{01}, \quad Re_{2} = \frac{Q_{2}}{H_{i} \cdot \nu}, \quad Q_{2} = \frac{\pi \cdot D_{o}^{2}}{4} \cdot V_{02}
\]

where $Q_{1}$ (m\(^{3}\)/s) and $Q_{2}$ (m\(^{3}\)/s) are the flow rates into the vortex chamber and $\nu$ (m\(^{2}\)/s) = $\eta / \rho$ is the kinematic viscosity of air. And the following driving conditions are arranged, for the inlet pipes 1 and 2

1. two inlet pipes are opened,
2. one of the inlet pipes is closed,

and for the exit pipes 1 and 2

1. two exit pipes are opened,
2. one of the two exit pipes is closed.
The above stated results are described in detail.

2. Fundamental Concept of Pressure Drop

In this section the fundamental equation of the pressure drop $\Delta P_r^s$ (Pa) based upon the radial distribution of the vorticity in the vortex chamber is explained for the two types of the flow systems under the driving condition of the inlet velocities to be $V_{01} = V_{02}$.

2.1 Co-Rotating Flow System

In case of the co-rotating flow system (Co-R-F-S), the angular momentum flux flowing into the vortex chamber from the inlet pipe 1 is promoted by that from the inlet pipe 2 due to the same spin directions. Therefore the maximum tangential velocity becomes to increase to $V_{01}\times V_{02}$, but when one inlet pipe is closed, so the maximum tangential velocity becomes to $V_{01}$ or $V_{02}$. Then the radial distribution of the tangential velocity can be expressed by Ogawa combined vortex model as schematically shown in Fig. 3. This model is composed of the quasi-forced (0 $\leq r \leq R_1$) and quasi-free ($R_1 \leq r \leq R_2$) vortex regions. Generally speaking, the eddy viscosity $\eta_{TV}$ (Pa·s) in the quasi-forced vortex region is higher than that of $\eta_{TV}$ (Pa·s) in the quasi-free vortex region. The empirical correlation between $\eta_{TV}$ and $\eta_{TV}$ can be represented as $\eta_{TV} \cdot \eta_{TV} \approx 10$. This value obtained by measuring the Reynolds stress was presented in the reference. Then in order to simplify the equation of the pressure drop, the radial distribution of the vorticity is assumed to be only one component of the z-axis as

$$\zeta = \frac{dV_\theta}{dr} + \frac{V_s}{r}$$

Therefore the pressure drop of the vortex chamber is defined as follows

$$\Delta P_r(R_a) = \Delta P_r^s + \rho \frac{V_s^2}{2} = \frac{\eta_{TV}}{Q_{TV}} \int \left( \zeta^2 dV + \rho \frac{r^2 V_s^2}{2} \right)$$

Then the equation of the pressure drop $\Delta P_r^s$ (Pa) can be written as follows

$$\Delta P_r^s = \frac{64 \cdot K_0 \cdot H_1 \cdot V_{im} \cdot \eta_{TV}}{R_2^3 \cdot V_0} \cdot \left( \frac{1 + n}{2 + n} \right)^2 \cdot \left\{ - \frac{n}{2 + n} + \frac{9}{8} \cdot \left( \frac{1 + n}{2 + n} \right)^2 + \frac{\eta_{TV}}{\eta_{TV}} + \frac{1}{4n} \cdot \left( \frac{1 - n}{2 + n} \right)^2 \cdot \left[ 1 - \left( \frac{n}{R_1} \right)^n \right] \right\}$$

In this equation, $n(1)$ is the velocity exponent and $K_0(1)$ is the constant value that depends on the flow condition and $r_i(m)$ is the boundary radius between the quasi-forced and quasi-free vortex regions. From eq. (4) the eddy viscosity $\eta_{TV}$ (Pa·s) can be estimated with the experimental results of the pressure drop of pure air flow. In addition to this, the coefficient of the pressure drop $\zeta_c(1)$ is defined as

$$\zeta_c(1) = \frac{\Delta P_r}{\rho \cdot V_0^2 / 2} = \frac{\Delta P_r^s}{\rho \cdot V_0^2 / 2} + 1$$

From eq. (5) it will be find that, when the rotational flow is the turbulent state and the eddy viscosity is proportional to the Reynolds number, so the coefficient of the pressure drop becomes to the constant value, on the other hand, when the rotational flow is the laminar state and the eddy viscosity is the constant value, so the coefficient of the pressure drop is inversely proportional to the Reynolds number.
2.2 Counter-Rotating Flow System

In case of the counter-rotating flow system, the angular momentum flux flowing into the vortex chamber is conserved in the space of the inlet pipe region, but at the contact plane of the middle space of the vortex chamber the angular momentum flux with the spin of the clockwise direction is extinguished by the angular momentum flux with the spin of the anti-clockwise direction, therefore the radial distribution of the tangential velocity as shown in Fig. 4 under the driving condition of $V_{o1} = V_{o2}$ may be assumed. In this figure, the value of $4 \times R_0$ was determined by the radial distribution of the tangential velocity measured with the cylindrical Pitot-tube of the diameter 2 mm made by the stainless steel pipe. The equation of this velocity distribution can be written as follows

$$V_{\phi}(r) = \left( - \frac{V_{\phi m}}{4 \times R_0^4} \right) \cdot \left( r - R_1 \right) \cdot \left| r - (R_1 - 4 \times R_0) \right|$$  \hspace{1cm} (6)

Substituting eq. (6) into eq. (4), so the equation of the pressure drop due to the vorticity distribution can be obtained as follows

$$\Delta P^* = \frac{1}{8} \cdot \frac{K_a \cdot H_i \cdot \eta \cdot V_{\phi m}^2}{R_0^4 \cdot V_{o1}^2} \cdot \left( \frac{R_1}{R_0} \right)^4 \cdot \left[ 4 \times \frac{R_0}{R_1} \cdot \left( 1 - \frac{2 \times R_0}{R_1} \right) + \left( 1 - 4 \times \frac{R_1}{R_1} + 8 \times \frac{R_0^2}{R_1^2} + \frac{1}{2} \right) \right]$$ \hspace{1cm} (7)

From eq. (7) the eddy viscosity $\eta \cdot \eta$ (Pa·s) can be estimated with the experimental results of the pressure drop. Khlatov derived an empirical equation of the eddy viscosity $\eta = \eta \cdot \eta$ for the small types of the cylindrical cyclone dust collectors of the diameters $D_1$ = 30, 50, 69 and 99 mm. In that case, the eddy viscosity was proportional to the Reynolds number $Re$. The comparison of eq. (8) with the calculated results by eq. (7) will be discussed in the later section.

2.3 Definitions of pressured drop and the coefficient of pressure drop

The pressure drop $\Delta P_i$ (Pa) of this vortex chamber is defined as the differences of the total pressure in the inlet and the total pressure at the exit position of the exit pipe as shown in Fig. 5.

The total pressure drop $\Delta P_i$ (Pa), the local pressure drop $\Delta P^*$ (Pa) and the local pressure drop $\Delta P_2$ (Pa) for the both flow systems are defined as follows

$$\Delta P_i (Pa) = \frac{\rho \cdot V_{o1}^2}{2} \cdot \frac{C_{t1}}{} + \Delta P^*$$ \hspace{1cm} (9)

$$\Delta P_2 (Pa) = \frac{\rho \cdot V_{o2}^2}{2} \cdot \frac{C_{t2}}{} + \Delta P^*$$ \hspace{1cm} (10)

Fig. 4 Schematic illustration of the radial distribution of the tangential velocity

Fig. 5 Schematic illustration of the main symbols for measuring the pressure drop
\[\Delta P_i (\text{Pa}) = \frac{\zeta_{\alpha} \left( \rho \cdot V_0^2 + \rho \cdot V_0^2 \right)}{2} = \frac{Q_1 \cdot \Delta P_1 + Q_2 \cdot \Delta P_2}{Q_1 + Q_2}\]

In this equation, the symbols \(Q_1 (\text{m}^3/\text{s})\) and \(Q_2 (\text{m}^3/\text{s})\) are the flow rate of gas flow in each inlet pipe and also the dimensionless quantities \(\zeta_{\alpha1}(1), \zeta_{\alpha1}(1)\) and \(\zeta_{\alpha2}(1)\) are the coefficients of the total pressure drop and the coefficients of the local pressure drop.

3. Experimental apparatus and experimental method

In order to measure the pressure drop and the distribution of the tangential velocity of rotational flow by the hot-wire system with constant temperature type for the both flow systems, the schematic illustration is shown in Fig. 6. Fig. 7 shows the main construction of the co-rotating flow system (Co-R-F-S) and the main sizes. Fig. 8 shows the main construction of the counter-rotating flow system (C-R-F-S) and the main sizes. The inlet pipes 1 and 2 are tangentially connected to the outer wall of the vortex chamber and also the exit pipes 1 and 2 is not immerse into the vortex chamber, therefore the flow behaviours in these vortex chambers differ from those in the cyclone dust collectors generating the secondary rotational flow around the annular space between the outer wall and the exit pipe. In Fig. 6, a symbol ① is a Gottingen type manometer, ② is a digital oscilloscope, ③ is a personal computer, ④ is a CTA system with the constant temperature type for measuring the tangential velocity, ⑤ is the hot-wire probe and I-type tungsten wire of diameter 4 μm was applied, ⑥ is a valve for regulating the flow rate of gas flow, ⑦ is a vortex chamber as shown in Fig. 7 and Fig. 8, ⑧ is a manometer for measuring the static pressure in the inlet pipe in front of the vortex chamber and ⑨ is the large cylindrical vessel in which water cooled by many lumps of ice was kept for the temperature control of the hot-air feeding from the blower ⑩. In reference ⑭ the detailed results of the radial distributions of the tangential and axial velocities and also the static pressures measured with the cylindrical Pitot-tube were reported. The inserted positions of the cylindrical Pitot-tube of the diameter 2 mm with one hole of the diameter 0.5 mm and the hot-wire probe were indicated in the detailed illustration of the vortex chamber. Then these vortex chambers were made with the transparent acrylic resin. Therefore the inner surfaces of these vortex chambers were in very smooth condition. So there were no effects of the surface roughness to the radial distribution of the tangential velocity and to the pressure drop of pure air.
flow in the vortex chamber\textsuperscript{30}.

Here in order to make clear the influences of the flowing out processes depending on the pressure drop of the vortex chamber for these two flow systems, three kinds of the flow processes are established\textsuperscript{37}. First one of these outflow processes is that two exit pipes 1 and 2 are opened, so the inlet gas flows from the inlet pipes 1 and 2 are flowing out from these exit pipes 1 and 2. Second is that the exit pipe 1 is closed and the exit pipe 2 is opened, so the inlet gas flows from the inlet pipes 1 and 2 are together flowing out from the exit pipe 2. Third is that the exit pipe 1 is opened and the exit pipe 2 is closed, so the inlet gas flows from the inlet pipes 1 and 2 are together flowing out from the exit pipe 1.

4. Experimental results of the pressure drop and the estimation of the eddy viscosity

In this section the estimation of the eddy viscosity by eq. (4) and eq. (7) is discussed and also the experimental results of the pressure drop and the coefficient of the pressure drop are discussed in detail.

Due to the performance of the blower, the driving conditions of the mean inlet velocity $V_{01}$ and $V_{02}$ are limited as follows for the co-rotating flow system the mean inlet velocity is restricted to $V_{01} = V_{02} = 25.0$ m/s, and for the counter-rotating flow system the mean inlet velocity is restricted to $V_{01} = V_{02} = 30.0$ m/s.

4.1 Estimation of the eddy viscosity

In order to estimate the eddy viscosity by eq. (4) for the radial distribution of the tangential velocity as shown in Fig. 3 on the co-rotating flow system and by eq. (7) for the radial distribution of the tangential velocity as shown in Fig. 4 on the counter-rotating flow system, the following experimental results of the total pressure drops are applied. Figure 9 shows the correlation between the total pressure drop defined by eq. (11) and the mean inlet velocity of $V_{02}$ (m/s) in the inlet pipe 2 for the both flow systems of the co-rotating flow system (C-R-F-S) and the counter-rotating flow system (C-R-F-S) on the condition of $V_{01} = 0$ m/s (inlet pipe 1 closed condition) in the inlet pipe 1.

From this figure, the total pressure drop $\Delta P$ (Pa) for the both flow systems are nearly proportional to the square of

![Fig. 9 Correlation between the total pressure drop and the mean inlet velocity for the both flow systems on the three outflow conditions of the both exit pipes opened, the exit pipe 1 closed and the exit pipe 2 closed under the condition of $V_{01} = 0$ m/s.](image)

![Fig. 10 Correlation between the coefficient of the total pressure drop and the Reynolds number for the both flow systems on the three outflow conditions of the both exit pipes opened, the exit pipe 1 closed and the exit pipe 2 closed on the condition of $V_{01} = 0$ m/s.](image)
the mean inlet velocity $V_0$ (m/s). However the total pressure drops of the co-rotating flow system on the three types of the outflow conditions of the both exit pipes opened, the exit pipe 1 closed and the exit pipe 2 closed showed a little higher values than those of the counter-rotating flow system. One of the main reasons is based on that the constructions of these both types of the vortex chambers attached two inlet pipes tangentially to the outer wall of the vortex chamber were not always the same. Then following the experimental results of the total pressure as shown in Figure 9, Figure 10 shows the correlation between the coefficient of the total pressure drop and the Reynolds number. From this figure it will be found that the turbulent flow state was dominated on the higher Reynolds number than the Reynolds number $Re_{2} = 2000$.

Then applying eq. (4), to the outflow condition of the both exit pipes opened for the both flow systems, assuming that the eddy viscosity $\eta\tau$ (Pa·s) in the quasi-free vortex region is $\eta\tau = 0.1 \times \eta\tau$, the velocity exponent $n = 0.75$ and also the boundary radius $r_t = 0.347 \times R_1$, so the empirical equation of $\eta\tau / \eta \ (1)$ can be obtained as follows

\begin{align}
\text{for Co-R-F-S} & \quad \eta\tau / \eta \ (1) = 0.0026 \times Re_{c2} \\
\text{for C-R-F-S} & \quad \eta\tau / \eta \ (1) = 0.0023 \times Re_{c2}.
\end{align}

On the other hand, the empirical equation of $\eta\tau / \eta \ (1)$ which was obtained from the many types of the vortex chambers by Naotaro is represented as

$$\eta\tau / \eta \ (1) = 0.00223 \times Re^{0.03}$$

Therefore the empirical equations of eq. (2) and eq. (3) of the estimated eddy viscosity are approximately coincided with Naotaro's empirical equation. And also, in the quasi-forced vortex region the eddy viscosity $\eta\tau$ (Pa·s) can be estimated as ten times higher than that of eq. (2) and eq. (3).

On the other hand, for the special case of the counter-rotating flow system as shown in Fig. 4, the radial distribution of the tangential velocity can be represented by eq. (6), so the equation of the pressure drop due to the vorticity can be represented by eq. (7). Here in order to recognize the flow pattern of how to change the angular momentum along the Z-axis, Fig.11 shows the experimental results of the tangential velocities at the radii $r = 10, 15, 20, 25, 30, 35, 45, 50, 55, 60, 65$ and 70 mm measured with the cylindrical Pitot-tube on the driving condition of $V_1 = V_0 = 30.0 \text{ m/s}$ for the counter-rotating flow system. In these figures, the locations of the axial position $Z(1)$ were indicated in Fig. 6. Figure 12 shows the correlation between the total pressure drop and the mean inlet velocity on the driving condition of $V_0 = 15.0 \text{ m/s}$ for the both flow systems. Figure 13 shows the correlation between the coefficient of the total pressure drop $\zeta(1)$ and the Reynolds number of the total flow rate $Re_{c1}$ (1) defined as $Re_{c1} = (Q_1 + Q_2) / \rho \cdot \nu$ on the driving condition of $V_0 = 15.0 \text{ m/s}$ for the both flow systems. From the results of the total pressure drop as shown in Fig.12, the minimum values of the total pressure drop of the counter-rotating flow system (C-R-F-S) for the three types of the outflow processes were generated on the driving condition of $V_2 = 15.0 \text{ m/s}$. Therefore the minimum values of the total pressure drop are always generated on the driving condition of $V_0 = V_2$ in spite of these three outflow processes. The physical reason of why this kind of the phenomena were occurred can be explained by considering the mixing process of the same quantity of the angular momentum fluxes entering into the vortex chamber with the clockwise and anti-clockwise spin directions as shown in Fig.11. Consequently in the middle plane of the vortex chamber, the radial distribution of the angular momentum about the vortex core region was vanished, so the pressure drop became the minimum value. Then from these experimental results of the total pressure drop on the condition of the both exit pipes opened, the eddy viscosity $\eta\tau$ (Pa·s) in the annular region can be estimated by eq. (7). Assuming $V_{s0} = V_0$ and $K_n = 0.5$, the calculated results are as follows

\begin{align}
\text{for } V_0 = V_2 = 5.0 \text{ m/s,} & \quad \eta\tau / \eta = 8.57 \\
\text{for } V_0 = V_2 = 10.0 \text{ m/s,} & \quad \eta\tau / \eta = 5.11 \\
\text{for } V_0 = V_2 = 15.0 \text{ m/s,} & \quad \eta\tau / \eta = 5.66 \\
\text{for } V_0 = V_2 = 20.0 \text{ m/s,} & \quad \eta\tau / \eta = 7.22 \\
\text{for } V_0 = V_2 = 25.0 \text{ m/s,} & \quad \eta\tau / \eta = 26.0.
\end{align}

These values are nearly the same quantities of eq. (2) and eq. (3) in the quasi-free vortex region. On the other hand, the eddy viscosity of the turbulent rotational flows in the various types of the vortex chambers were studied in the book of Aerodynamics in Cyclone-Type of Vortex Chamber written by ИИЯМ [19], and the maximum values of the eddy viscosity were located in the quasi-free vortex region. However from the experimental results of the Reynolds stress, the
Fig. 11  Axial distributions of the tangential velocities at the each radii for \( V_{o1} = V_{o2} = 30.0 \, \text{m/s} \) of the counter-rotating flow system on the both exit pipes opened.
value of the eddy viscosity in the quasi-forced vortex region was about ten times higher than that in the quasi-free vortex region.

4.2 Total pressure drop and the coefficient of the total pressure drop

4.2.1 In case of Co-Rotating Flow System

Figure 14 shows the correlation between the total pressure drop $\Delta P_t$ (Pa), the local pressure drop $\Delta P_1$ (Pa) and the local pressure drop $\Delta P_2$ (Pa) and the mean inlet velocity $V_{o2}$ (m/s) on the mean inlet velocity in the inlet pipe $V_{o1}=20.0$ m/s to be the constant state for the exit pipe 1 closed condition as one of the experimental methods. These pressure drops between $V_{o2}=5.0$ m/s and 25.0 m/s are nearly proportional to 0.6 power of the mean inlet velocity. The total pressure drop was calculated by eq. (1). On the other hand, when the exit pipe 2 is closed or the both exit pipes 1 and 2 are opened, so the remarkable effects to the pressure drops $\Delta P_1$, $\Delta P_2$ and $\Delta P_t$ do not appear in comparison with those of the results of Fig. 14.

Figure 15 shows the correlation between the total pressure drop $\Delta P_t$ (Pa), the local pressure drop $\Delta P_1$ (Pa) and the local pressure drop $\Delta P_2$ (Pa) and the mean inlet velocity $V_{o2}$ (m/s) on the mean inlet velocity $V_{o1}=20.0$ m/s for
the exit pipe 2 closed condition. These pressure drops between \( V_{O_2} = 15.0 \text{ m/s} \) and 25.0 m/s are proportional to the mean velocity.

Figure 16 shows the correlation between the total pressure drop \( \Delta P_1 (\text{Pa}) \), the local pressure drop \( \Delta P_1 (\text{Pa}) \) and the local pressure drop \( \Delta P_2 (\text{Pa}) \) and the mean inlet velocity \( V_{O_2} (\text{m/s}) \) on the mean inlet velocity \( V_{O_1} = 20.0 \text{ m/s} \) for the both exit pipes to be the opened condition. These pressure drops between \( V_{O_2} = 15.0 \text{ m/s} \) and 25.0 m/s are proportional to the mean inlet velocity. Therefore reviewing the pressure drops relating to these three outflow processes, there is no remarkable influence for the co-rotating flow system, so the co-rotating flow system is one of the simple equipments for the vortex control to promote the tangential velocity without the remarkable change of the velocity distribution in the vortex chamber and at the same time the pressure drop is also increased.

Figure 17 shows the correlation between the color bands of the equi-total pressure drop \( \Delta P_t (\text{Pa}) \) and the mean inlet velocities \( V_{O_1} (\text{m/s}) \) and \( V_{O_2} (\text{m/s}) \) for the exit pipe 1 closed condition.

Figure 18 shows the correlation between the color bands of the equi-value of the coefficient of the total pressure drop and the Reynolds numbers \( \text{Rec}_1 (1) \) and \( \text{Rec}_2 (1) \) for the exit pipe 1 closed condition. The correlation between the total pressure drop \( \Delta P_t (\text{Pa}) \) and the coefficient of the total pressure drop \( \xi_{ct}(1) \) is defined by eq. (11). From this figure it will be found that the maximum value of the coefficient of the total pressure drop is occupied in the domains of \( \text{Rec}_1 = 2000 \sim 3000 \) and \( \text{Rec}_2 = 2000 \sim 3000 \).

Figure 19 shows the correlation between the color bands of the equi-total pressure drop \( \Delta P_t (\text{Pa}) \) and the mean inlet velocities \( V_{O_1} (\text{m/s}) \) and \( V_{O_2} (\text{m/s}) \) for the exit pipe 2 closed condition.

Figure 20 shows the correlation between the color bands of the equi-value of the coefficient of the total pressure drop \( \xi_{ct}(1) \) and the Reynolds numbers \( \text{Rec}_1 (1) \) and \( \text{Rec}_2 (2) \) for the exit pipe 2 closed condition. From this figure it will be found that the maximum value of the coefficient of the total pressure drop is occupied in the domains of \( \text{Rec}_1 = 4000 \sim 5000 \) and \( \text{Rec}_2 = 3500 \sim 4500 \).

Figure 21 shows the correlation between the color bands of the equi-total pressure drop \( \Delta P_t (\text{Pa}) \) and the mean inlet velocities \( V_{O_1} (\text{m/s}) \) and \( V_{O_2} (\text{m/s}) \) for the both exit pipes opened condition. These curves of the color bands show
a little difference in comparison with those of Fig.17 and Fig.19. Therefore it will be found that the flow pattern on the both exit pipes to be the opened condition is a different behavior in comparison with that on one of the exit pipes closed condition.

Figure 22 shows the correlation between the color bands of the equi-value of the coefficient of the total pressure drop ζ_{ct(1)} and the Reynolds numbers Rec_{1(1)} and Rec_{2(1)} for the both exit pipes opened condition. From this figure it will be found that the maximum value of the coefficient of the total pressure drop is occupied in the domains of Rec_{1(1)} = 2700~3500 and Rec_{2(1)} = 1700~4300.

Fig. 19 Correlation between the total pressure drop and the mean inlet velocities for exit pipe 2 closed condition.

Fig. 20 Correlation between the total pressure drop and the Reynolds numbers for the exit pipe 2 closed condition.

Fig. 21 Correlation between the total pressure drop and the mean inlet velocities for the both exit pipes opened condition.

Fig. 22 Correlation between the coefficient of the total pressure drop and the Reynolds numbers for the both exit pipes opened condition.

4.2.2 In case of Counter-Rotating Flow System

Figure 23 shows the correlation between the total pressure drop ΔP_{t}(Pa), the local pressure drop ΔP_{l}(Pa) and the local pressure drop ΔP_{f}(Pa) and the mean inlet velocity Vo_{2} (m/s) on the inlet velocity Vo_{1} = 20.0m/s in the inlet pipe 1 for the exit pipe 1 closed condition. From this figure, it will be found that the pressure drops become the minimum value at the driving condition of Vo_{1} = Vo_{2} = 20.0m/s. Therefore one of the control methods for suppressing the vortex flow in the vortex chamber can be obtained by the equipment of the counter-rotating flow system. Espe-
cially as shown in Fig.11 the axial distributions of the tangential velocity along the Z-axis, the vortex core region with the high vorticity of the general distribution of the tangential velocity was disappeared, so the pressure drop becomes to decrease enormously.

Figure 24 shows the correlation between the total pressure drop $\Delta P_1$ (Pa) and the local pressure drop $\Delta P_1$ (Pa) and the local pressure drop $\Delta P_2$ (Pa) and the mean inlet velocity $V_{01}$ (m/s) and the mean inlet velocity $V_{02}$ (m/s) for the exit pipe 2 closed condition. In this case, the correlation between the pressure drops and the mean inlet velocity gives nearly the same tendency towards the results as shown in Fig.23.

Figure 25 shows the correlation between the total pressure drop $\Delta P_1$ (Pa) and the mean inlet velocity $V_{01}$ (m/s) and the mean inlet velocity $V_{02}$ (m/s) for the both exit pipes to be opened condition. The minimum pressure drops are occurred nearly at the driving condition of $V_{01} = V_{02} \approx 20.0$ m/s. As one of the experimental results as shown in Figs.23, 24 and 25, the minimum pressure drop is occurs at the driving condition of $V_{01} = V_{02}$ which is independent of the three types of the outflow processes.

Figure 26 shows the correlation between the color bands of the equi-total pressure drop $\Delta P_1$ (Pa) and the mean inlet velocities $V_{01}$ (m/s) and $V_{02}$ (m/s) for the exit pipe 1 closed condition. From this figure it will be found that the minimum values of the total pressure drop are located at the driving condition of $V_{01} = V_{02}$. On the other hand, the maximum value of the total pressure drop is located at the driving condition of $V_{01} = 30.0$ m/s and $V_{02} = 0.0$ m/s.

Figure 27 shows the correlation between the color bands of the equi-value of the coefficient of the total pressure drop $\xi_{d}(1)$ and the Reynolds numbers $Re_{c1}(1)$ and $Re_{c2}(1)$ for the exit pipe 1 closed condition. The minimum values of the coefficient of the pressure drop are located at the flow condition of $Re_{c1}$.$\approx Re_{c2}$.

Figure 28 shows the correlation between the color bands of the equi-total pressure drop $\Delta P_1$ (Pa) and the mean inlet velocities $V_{01}$ (m/s) and $V_{02}$ (m/s) for the exit pipe 2 closed condition. A general configuration is very similar to that of Fig.26. The maximum value of the total pressure drop is located at the driving condition of $V_{01} = 30.0$ m/s and $V_{02} = 0.0$ m/s. On the other hand, the minimum value of the total pressure drop is located at the driving condition of $V_{01} = V_{02}$.

Figure 29 shows the correlation between the color bands of the equi-value of the coefficient of the total pressure drop $\xi_{d}(1)$ and the Reynolds numbers $Re_{c1}(1)$ and $Re_{c2}(1)$ for the exit pipe 2 closed condition. Here it is noted that the location of the minimum value of the total pressure drop does not always coincide with that of the coefficient of the total pressure drop defined by eq. (11)
Fig. 26 Correlation between the total pressure drop and the mean inlet velocities for exit pipe 1 closed condition.

Fig. 27 Correlation between the coefficient of the total pressure drop and the Reynolds numbers for the exit pipe 1 closed condition.

Fig. 28 Correlation between the total pressure drop and the mean inlet velocities for exit pipe 2 closed condition.

Fig. 29 Correlation between the coefficient of the total pressure drop and the Reynolds numbers for the exit pipe 2 closed condition.

Fig. 30 Correlation between the total pressure drop and the mean inlet velocities for the both exit pipes opened condition.

Fig. 31 Correlation between the coefficient of the total pressure drop and the Reynolds numbers for the both exit pipes opened condition.
Figure 30 shows the correlation between the color bands of the equi-total pressure drop $\Delta P_t$ (Pa) and the mean inlet velocities $V_{o1}$ (m/s) and $V_{o2}$ (m/s) for the both exit pipes to be opened condition. The minimum values of the total pressure drop is located at the driving condition of $V_{o1} \neq V_{o2}$.

Figure 31 shows the correlation between the color bands of the equi-valu of the coefficient of the total pressure drop $\zeta_{d}(1)$ and the Reynolds numbers $Re_1(1)$ and $Re_2(1)$ for the both exit pipes to be opened condition. The minimum value of the coefficient of the total pressure drop $\zeta_{d}(1)$ is located at the flow condition of $Re_1 \neq Re_2$.

5. Conclusions

In order to obtain the fundamental information about the mutual influences of the angular momentum flux from the inlet pipe 1 to the angular momentum flux from the inlet pipe 2 for the vortex control in the cylindrical vortex chamber, the following flow processes for the inflow conditions from the inlet pipes 1 and 2 and for the outflow conditions from the exit pipes 1 and 2 were arranged. For the inflow processes, the driving conditions were arranged as the two inlet pipes opened and one of the inlet pipes closed. For the exit pipes 1 and 2, the outflow conditions were arranged as the two exit pipe opened and one of the exit pipes closed. From these experimental results of the pressure drop and the coefficient of the pressure drop for the co-rotating flow system and the counter-rotating flow system and the three kinds of the outflow processes of the two exit pipes, the following conclusions are obtained:\(^{23-24}\).

1. The values of the eddy viscosity in the quasi-free vortex region were estimated for the both flow systems of Co-R-F-S and C-R-F-S. These estimated values were approximately equivalent to those of Halatov's empirical equation.

2. From Fig.14 to Fig.23 the correlations between the experimental results of the total pressure drops, the local pressure drops and the mean inlet velocities were shown for the co-rotating flow system (Co-R-F-S). In this case, in spite of the three kinds of the outflow processes, the pressure drops were increased due to the augmentation of the tangential velocities by the mutual mixing of the angular momentum fluxes with the same spin directions at the middle plane in the vortex chamber. However under the driving condition of $V_{o1} = \text{constant state},$ the augmentation of the circulation in the vortex chamber does not always contribute below $V_{o2}/V_{o1} < 1.0.$ Therefore in order to control the vortex flow for promoting the circulation, the driving condition must be kept to $V_{o2}/V_{o1} > 1$ under $V_{o1}$ to be constant state. This kind of the mixing process of the angular momentum fluxes was applied to the rotary flow dust collector for separating the fine solid particles of the particle diameter $X_p = 0.5 \mu m^{21}.$ This kind of the co-rotating flow system with the flat vortex chamber was applied to the vortex whistle for producing sound by Vonnegut\(^2\).

3. From Fig.24 to Fig.31 the correlation between the experimental results of the total pressure drops, the local pressure drops and the mean inlet velocities were shown for the counter-rotating flow system (C-R-F-S). In this case, the pressure drops were closely related to the both inlet velocities $V_{o1}$ and $V_{o2}.$ When the both inlet velocities were set up as the same, so the pressure drops became the locally minimum value due to the reduction of the tangential velocities by the mutual mixing of the angular momentum fluxes with the clockwise and counter-clockwise spin directions at the middle plane in the vortex chamber as shown in Fig.11 under the driving condition of $V_{o1} = V_{o2} = 30.0 \text{ m/s}.\$ And also the direction of the rotational flow in the vortex chamber is related to the ratio of $V_{o2}$ to $V_{o1}$.

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Appendix

The pressure drop of the vortex flow in the vortex chamber is depended on the radial distribution of the tangential velocity. When the tangential velocity becomes to higher, so the pressure drop becomes to higher. Therefore in order to recognize qualitatively the fluid dynamical characteristics of the pressure drop, one of the main factors is to estimate the circulation \( \Gamma \) \( (m^2/s) = \rho \cdot V_T \cdot r \) of the vortex flow which is defined as the product of the tangential velocity \( V_T \) and the radius \( r \). So for the co-rotating flow system and for the counter-rotating flow system, the equation of the circulation can be obtained \( (A \cdot 1) \).

\[
\rho \cdot Q_1 \cdot V_0_1 \cdot (R_1 - R_0) + \rho \cdot Q_2 \cdot V_0_2 \cdot (R_1 - R_0) = \rho \cdot (Q_1 + Q_2) \cdot r \cdot V_T
\]

Therefore the equation of the dimensionless circulation can be obtained as
The mean tangential velocity \( V_{oi} \) near the outer wall surface is also estimated by the mixing of the angular momenta through the cross sectional area \( Hi \times Ro \) near the outer wall region as

\[
\frac{V_{oi}}{V_{oe}} = 2 \times \left[ \frac{\pi \cdot R_0 \cdot (R_1 - R_0)}{H \cdot (2 \times R_1 - R_0)} \right] \left[ 1 \frac{V_{o2}}{V_{o1}} \right] \left[ 1 + \left( \frac{V_{o2}}{V_{o1}} \right)^2 \right] \tag{A-3}
\]

For this vortex chamber, \( V_{oi} / V_{oe} \) is independent of the velocity ratios and obtained as \( V_{oi} / V_{oe} = 0.677 \).

(b) In case of counter-rotating flow system

The main spin direction of the angular momentum fluxes flowing from the inlet pipe 1 and the inlet pipe 2 into the vortex chamber is depended on the quantity of the total energy. When the total energy from the inlet pipe is higher than that from the inlet pipe 2, so the spin direction in the vortex chamber is followed up to the spin direction of the angular momentum from the inlet pipe 1. For simplicity, in case of the perfect mixing with two angular momenta, the direction of the rotation of the angular momentum is mutually opposed, then assuming that the angular momentum in the center region of the vortex chamber is cancelled, so the equation of the dimensionless circulation can be obtained as

\[
\Gamma^* = \frac{V_o \cdot r}{V_{o1} \cdot (R_1 - R_0)} = 1 - \frac{V_{o2}}{V_{o1}} \tag{A-4}
\]

Consequently Fig. A-1 shows the correlation between the dimensionless circulation and the velocity ratio for the both flow systems. From this figure, it will be found that the dimensionless circulation of the co-rotating flow system for the velocity ratio between \( V_{o2} / V_{o1} = 0 ~ 1.0 \) has the minimum value \( \Gamma^* = 0.829 \) at \( V_{o2} / V_{o1} = 0.414 \). And then when the velocity ratio \( V_{o2} / V_{o1} \) is larger than 1, so the dimensionless circulation becomes to increase. Therefore it is predicted that the remarkable pressure drop will be increased. On the other hand, for the counter-rotating flow system, the spin direction of the dimensionless circulation becomes to change from the positive value to the negative value at the transition point of \( V_{o2} / V_{o1} = 1 \). Therefore at the driving condition of \( V_{o2} / V_{o1} = 1 \), the pressure drop becomes to the minimum value. However in case of the counter-rotating flow system, practical point of view, the spin direction of the whole fluid element in the vortex chamber is followed up to the spin direction of the higher total energy. Since the flux of the higher total energy pushes aside against the flux of the lower total energy which will be returned and followed up to the spin direction of the flux of the higher total energy. Then the tangential velocity \( V_o \) near the outer wall region can be obtained by the conservation of the angular momentum fluxes. When the total energy \( E1 \) from the inlet pipe 1 is larger than that of the inlet pipe 2, so \( V_{oi} / V_{oe} \) can be obtained as

\[
\frac{V_{oi}}{V_{oe}} = 2 \times \left[ \frac{\pi \cdot R_0 \cdot (R_1 - R_0)}{H \cdot (2 \times R_1 - R_0)} \right] \left[ 1 - \frac{V_{o2}}{V_{o1}} \right] \left[ 1 + \left( \frac{V_{o2}}{V_{o1}} \right)^2 \right] \tag{A-5}
\]

On the other hand, when the total energy \( E2 \) from the inlet pipe 2 is larger than that of the inlet pipe 1, so \( V_{oi} / V_{oe} \) can be obtained as

\[
\frac{V_{oi}}{V_{oe}} = -2 \times \left[ \frac{\pi \cdot R_0 \cdot (R_1 - R_0)}{H \cdot (2 \times R_1 - R_0)} \right] \left[ 1 - \left( \frac{2 - \frac{V_{o2}}{V_{o1}}}{1 + \left( \frac{V_{o2}}{V_{o1}} \right)^2} \right) \right] \tag{A-6}
\]

In addition to this, the maximum tangential velocity for the co-rotating flow system can be estimated by the modified
In this equation, a velocity factor for the both exit pipes opened is given as $K_v = 0.805$. 

Figure A-2 shows the experimental results of the tangential velocity on the driving condition of $V_{o1} = 25$ m/s, $V_{o2} = 15$ m/s at $Z = 2, 6$ and 10 for the co-rotating flow system. In this case, the velocity exponent is $n = 1$, so substituting the values of the sizes of the vortex chamber and the value of $n = 1$ into eq. (A-7), so it can be obtained as $V_{s m}/V_{oe} = 1.60$ for $V_{o2} = 0$ m/s, $V_{s max}/V_{oe} = 1.42$ for $V_{o2} = 5$ m/s and $V_{s max}/V_{oe} = 0.95$ for $V_{o2} = 15$ m/s, which are nearly in agreement with the experimental results. Especially it is very interesting to note that the tangential velocity at $Z = 6$ for $V_{o1} = V_{o2} = 25$ m/s is nearly cancelled, but in the upper part of the vortex chamber the spin direction is anti-clockwise and in the lower part of the vortex chamber the spin direction is clockwise. For $V_{o2} = 0, 5$ and 15
m/s, the spin directions of the tangential velocity are followed up to the spin direction of $V_{o1} = 25 \text{m/s}$. These phenomena are closely related to the characteristics of the pressure drop which depends on the inlet velocities $V_{o1}$ and $V_{o2}$, and also the maximum tangential velocity $V_{\text{max}}$. Figure A-4 shows the correlation between the maximum tangential velocity $V_{\text{max}}$ (m/s) for the both flow systems and the mean inlet velocity $V_{o2}$ (m/s) under the condition of $V_{o1} = 25 \text{ m/s}$ to be the constant state. Fig. A-5 shows the correlation between the maximum pressure drop $\Delta P_{ct}$ (Pa) and the mean inlet velocity $V_{o2}$ (m/s) for the both flow systems under the driving condition of $V_{o1} = 25 \text{m/s}$ to be the constant state.

Figure A-6 shows the equi-flow rate bands for the co-rotating flow system under the driving condition of $V_{o1} = 25 \text{m/s}$ and $V_{o2} = 5 \text{ m/s}$. Figure A-7 shows the equi-flow rate bands for the counter-rotating flow system under the driving condition of $V_{o1} = 25 \text{m/s}$ and $V_{o2} = 5 \text{ m/s}$. Roughly speaking, even if the both flow systems are different, the equi-flow rate bands show the very similar patterns. Since the total energy from the inlet pipe 1 is larger than that from the inlet pipe 2, so the wholly spin direction in the vortex chamber is followed up to the spin direction of the rotational flow from the inlet pipe 1. Therefore the both equi-flow rate bands show the similar pattern.