Decay of Rotational Airflow with Flow Conditioner in Larger Diameter Ducts for Dust Concentration Measurement using Isokinetic Sampling

Z. C. Tan, Y. Zhang, S. E. Ford

Department of Agricultural and Biological Engineering, University of Illinois at Urbana-Champaign, 1304 West Pennsylvania Avenue, Room 332, Urbana, Illinois 61801.
Phone: (217) 244-6316, Fax: (217) 244-0323. Email: ztan1@uiuc.edu

Abstract

Within this study the decay speed of centrifugal device induced rotational airflow in round ducts and its conditioning for dust concentration measurement was investigated. Tests were conducted under room conditions at average Reynolds numbers within the range of $0.7–1.14\times10^5$. Duct diameters were 152 mm and 203 mm. Airflow in the ducts was found nonuniform and unsteady within the axial location to diameter ratio (x/D) of 10.5. The nonuniformity of axial airflow was nearly independent of the average Reynolds number. The study proved that it is impractical to decay the rotational airflow simply by using straight extension duct. A flow conditioner with minimal particle loss is needed to make the air flow uniform and steady. As a case study, a simple flow conditioner was tested. It decayed the rotational airflow to an acceptable level within x/D=2. The measured dust concentrations at 5 radial locations at x/D=2 were uniform: $\sigma/\mu<1\%$ for total mass concentration, and $\sigma/\mu<10\%$ for all the particles smaller than 10 μm.

Keywords. Rotational airflow, airflow conditioning, dust sampling

Introduction

Cyclones have been widely used in many agricultural engineering applications. They were primarily used for larger particle separation (Carpenter 1986) or dust sampling at low airflow rates (Crook 1995, Lacey and Venette 1995). Although cyclones have been well industrialized and commercialized, uncertainties remain and they encourage researchers to further study the particle separation mechanism. Recent advances in this area indicate that it is very likely to be further developed for separating respirable particles at low cost and higher airflow rates (Zhang *et al.* 2001, Wang *et al*. 2002). During the investigation process, dust separation efficiency of a cyclone needs to be measured.

With the increasing concern of respirable particles in environments, air-cleaning devices are often tested based on particle size (ANSI/ASHRAE 1999). Isokinetic sampling systems are often used for dust concentration measurements in ventilation ducts. In order to minimize the anisokinetic effects (Hinds 1999), the sampling nozzles should be correctly aligned to match the

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air velocity profile. Consequently rotational airflow exiting a cyclone has to be conditioned to an acceptable steadiness before air velocity profile can be determined and dust concentration can be measured.

In addition, uniform airflow is also preferred to reduce other sampling errors. It is a challenge to pick a representative sampling point in the rotational airflow. A multipoint sampling system could be employed. One apparent disadvantage is that many sampling heads will inference with the airflow pattern and introduce a significant error factor. Another disadvantage is that more sampling points require more sampling tubes, and consequently increase particle loss in sampling tubes. Therefore, a single point sampling system is preferred. The upstream airflow is not rotational. A single sample point is acceptable. However, before a single downstream sample point can be taken, we must condition the rotational airflow to an acceptable uniformity. ANSI/ASHRAE (1999) standard requires that qualification tests shall be performed for uniformities of air velocity and aerosol in the test duct. The ANSI/ASHRAE standard also recommends bending the test duct 180^{o} to bring the downstream sample locations relatively close to the upstream location, which allows short sample lines to the particle counter. It reduces the overall length of the test duct, facilitating its placement within the test room. However, it should be noted that this introduces another error factor. The bending could create rotational airflow. This swirl poses challenges in accurate measurements of airflow rate (McManus *et al*. 1985, Laribi *et al*. 2001), and maybe consequently, the dust concentration.

Two methods are often considered for airflow conditioning. One is the use of long straight extension ducts and another is the use of flow conditioners. The rotational airflow in a duct will decay to an acceptable level provided the straight duct is sufficiently long. The question is how long it should be. If the duct is too long, excessive space is needed to setup the measurement system, and more importantly, there might be too much particle loss on the inner surface of the duct.

Studies on rotational flow have been reported since the 1950s. The majority of these studies dealt with rotational flows in burners and air mixing in combustion areas. Only few of them are about decay of rotational flows. One of the early studies on rotational flow in ducts is an analysis of rotational flow of a constant property fluid in a straight duct of circular cross-section by Talbot (1954). The subject was steady and laminar flow in ducts. Kreith and Sonju (1965) later presented the average decay of tape-induced fully developed turbulent swirl in water flow through a duct. The duct was 25.4 mm in diameter and 2,540 mm long. The twisted tape inducers consisted of 0.85 mm thick galvanized steel strips. The experiments were done with Reynolds numbers from 10^4 to 10^5. It indicated that the rate of decay depended on the axial Reynolds number: the decay rate increased as the Reynolds number decreased, but the decay was found to be independent of the initial swirl intensity. It was observed that turbulent swirl decayed to about 10-20% of its initial intensity at an axial location to diameter ratio (x/D) of 50. Recently Laribi *et al*. (2001) experimentally studied the decay of swirling turbulent gas (petroleum) flow in a long duct. The inner diameter of the duct was 100 mm. The swirling flow was generated by double

90-degree elbow. Tests were conducted at Reynolds number ranging from $3x10^4$ to $1.1x10^5$. It was observed that the flow was fully developed at *x*/D=90. The decay of a swirling flow of nitrogen gas in long ducts was reported by McManus *et al.* (1985). Tests were conducted under ambient temperature and pressure of 40 kPa at Re=$8x10^5$ to $1.4x10^6$. The decay of 60 degrees of swirling across a 100 mm smooth-walled duct measured at x/D=0, 10, 33, and 70. It was found that at *x*/D=70, the decay was 59%. Based on 6 sets of experimental data, they gave a fit decay equation,

$$Percent\ Decay = 0.954 \exp\left(-0.0126 \frac{x}{D}\right) \tag{1}$$

where *x* is the axial location in the duct, D the inner diameter of the duct, and *x*/D the axial location to diameter ratio. From this equation, we can predict that the decay can be reduced to 10% at *x*/D=200.

Although the exact *x*/D numbers are different for the rotational flow to become fully developed, the above representative studies have shown that the duct has to be very long to reduce swirl to acceptable levels. In a laboratory, it might not be a practical solution for eliminating possible measurement errors attributed to swirl simply by using long ducts.

Objective

Among the limited amount of literature on decay of rotational fluid flow, few of them are about decay of rotational airflow in long ducts, and most of them were conducted in ducts less than 100 mm in diameter. The objective of this research is to study the decay of a rotational airflow created by a rotation device (*i.e.* centrifugal fan or cyclone) in ducts larger than 100 mm in diameter. It will be proved that a long duct is impractical, and a flow conditioner has to be used to condition the rotational airflow. As a case study, a flow conditioner with a large open ratio and with smooth surfaces was made and its effect on flow conditioning was tested. It is also expected that researchers who measure dust emission rate downstream of exhausting fans can find useful information from the results herein.

Experimental Methods

Ducts used in this research were commercial galvanized steel ducts with inner diameters of D= 152 mm and 203 mm. Maximum length of the ducts was 2,100 mm. The maximum length to diameter ratio, L/D, was about 10.8 due to space limitations in the laboratory. Axial velocity profile was measured using an anemometer from TSI, Inc.. The anemometer's measurement range was 0 to 20.3 m/s, with an accuracy of ±5.0% of reading or ±0.025 m/s whichever is greater. In this paper, airflow direction is our major concern for isokinetic sampling purpose. We wanted to condition the rotational airflow into a one-dimensional airflow. Therefore, only the axial velocity profile was measured. If axial velocity is not steady or uniform, the total velocity is not steady or uniform. While rotating the anemometer during test, other velocity components were obvious if the axial flow was not steady or uniform. On the other hand, other two velocity components were not apparent when the axial flow was steady and uniform. Samples were taken at different axial locations (*x*/D). At each location, velocities were measured at different radial locations.

The rotational airflow was generated with an in-line centrifugal fan. The airflow rate was adjusted with a variable transformer. The transformer's maximum output settings were 120 V and 140 V. It could be adjusted from 0% to 100% of each maximum output. In this project, seven voltage levels of fan power were used: 90% and 100% of 140V, and 60%, 70%, 80%, 90% and 100% of 120V. For each power level and each *x*/D in duct, axial air velocities at up to 13 evenly distributed points were measured, with increase of 12.7 mm or 25.4 mm (see Figure 5).

To verify the conditioning effect of the simple airflow conditioner, dust spatial distribution downstream was measured using an isokinetic sampling system. Particle concentrations were measured using APS 3321 (TSI, Inc.). The sampling points were 2D downstream of the flow conditioner, and their relative radial locations are shown in Figure 1. The corresponding radial locations of #1, #2, #3, #4 and #5 are 1/3 R, 1/6 R, 1/6 R, 7/12 R, and 5/6 R, respectively. 10 samples were taken at each of the 5 radial locations. The tests were conducted under Reynolds number of Re=2.2x10^4.

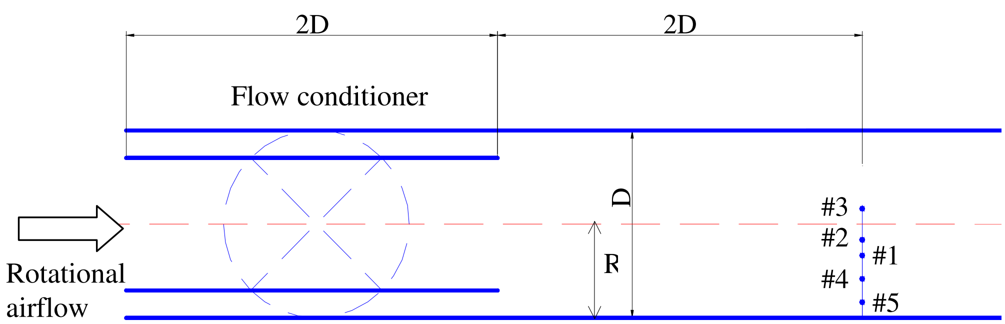

Figure 1. Particle sampling locations

Standard Arizona test dust was dispersed using a turntable dust generator. The size distribution of particles entering the air stream is shown in Figure 2. As seen in the graph, 95% by volume are less than 12 μm with a mass median of 4.8 μm.

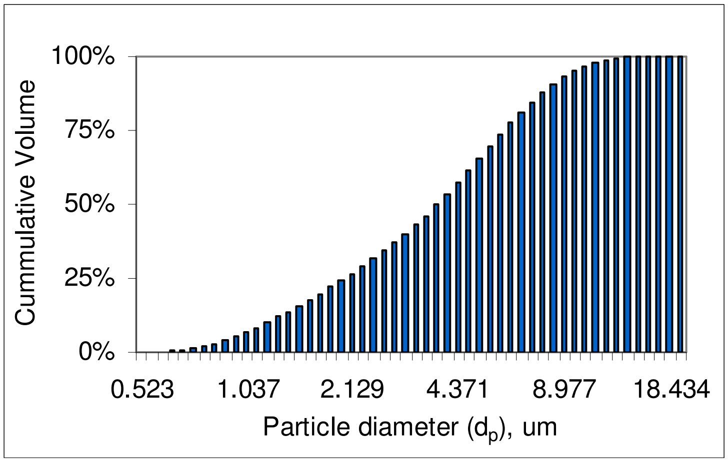

Figure 2. Particle size distribution

Results and Discussion

In this paper, a coordinate system as illustrated in Figure 3 is used. Axial direction is *x*, and radial direction is *r*. The length, radius and diameter of the duct are L, R, and D, respectively. At any axial location *x*, the velocity at different radial location *r* is Vr. The velocity on the centerline is Vc. The average velocity at *x* is V.

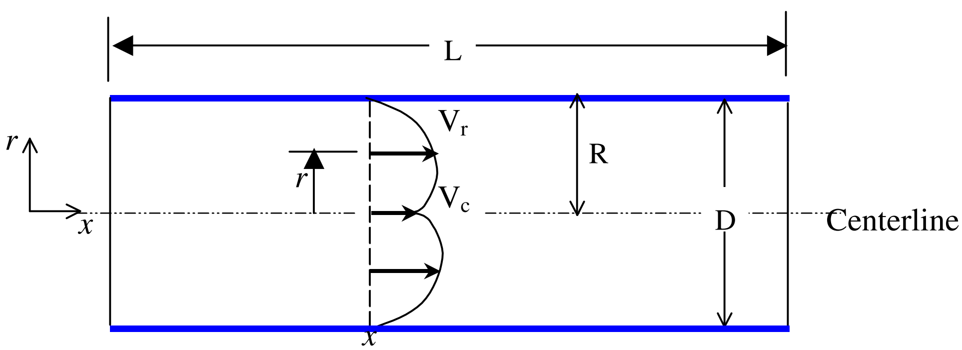

Figure 3. Illustration of the parameters referred in this paper

In this paper, dimensionless parameters including Reynolds numbers, nonuniformity parameter, and dimensionless velocity are used for data analysis. For airflow inside a duct, Reynolds number is defined as (ANSI/ASME 1979),

$$\mathrm{Re}=\frac{Vl}{v} \tag{2}$$

where: V is the average fluid velocity (m/s). , l a characteristic dimension of the system in which the flow occurs (m), and v the kinematic viscosity of the fluid (m^2/s). The characteristic dimension is the inner diameter of the duct, D. The kinematic viscosity is 1.5×10^{-5} m^2/s for room air at 20 °C and 1 atm. Average Reynolds numbers are calculated using the averaged axial velocities. The average Reynolds numbers in ducts in our tests were within the range of 0.7-1.14×10^5.

1. Effect of centrifugal device location

The airflow pattern inside a duct created by pushing air into a duct is different from pulling air out. Curves in Figure 4 are dimensionless velocities, the velocity at *r* over the average velocity (Vr/V), against dimensionless radial location, radial location *r* over the radius of the duct (r/R). For the example cases in Figure 4, the average Reynolds number was 1.14×10^5. The effects under other operation conditions were similar. When the fan was pulling air out of the duct, Vr/V's at all the measured locations, even at *x*/D=0.46, were very close 1. It indicates that airflow was uniform. However, when the duct was downstream the device, Vr/V's at most of the measured points are way off unity, even at *x*/D=8.8. It indicates that downstream centrifugal device does not create strong rotational flow in duct, but upstream centrifugal device does.

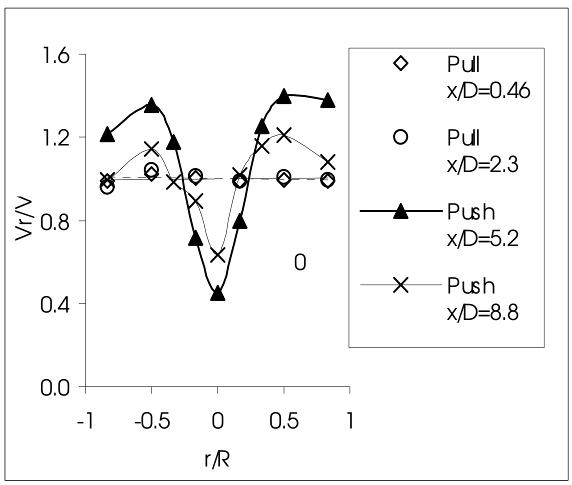

Figure 4. Effect of device location (pulling air out of and pushing air in duct) (r is the radial location, R is the Radius of the duct, Vr is the velocity at r, V is the average velocity, *x* is the axial location, and D is the diameter of the duct).

Since the rotational effect was negligible when the duct was upstream of our fan, only those cases where the duct was downstream of the fan are presented below. An example velocity profile is first given to visualize the airflow inside the ducts with and without flow conditioner.

Then the uniformity and steadiness of the airflow are compared. At the end, as a case study, the dust concentration distribution after the flow conditioner is presented.

2. Axial velocity profile

Axial velocity patterns were similar for all 7 average Reynolds numbers (fan levels). Figure 5 is an example that illustrates the axial velocity profile in ducts with and without flow conditioner at different *x*/D. The corresponding average Reynolds number was 1.14×10^5 (140V x 100% fan power). The flow conditioner was made of galvanized steel sheet metal. It had 4 vanes with L/D=2. When a flow conditioner is used, we have to take into consideration the particle loss on the flow conditioner. Particle loss on the flow conditioner should be minimized as much as possible. Therefore, the open area of the conditioner should to be as large as possible, and the surfaces should be as smooth as possible. Flow conditioners that are made up of multiple small tubes (i.e. 1 mm diameter) were not considered. The sheet metal is about 1 mm thick with smooth surface. The open ratio of a conditioner in a 152 mm duct is about 98.3%. Considering the large open ratio and smooth surface of the vanes, and small downstream particles, it is expected that the pressure drop and particle loss were minimal. This should be verified.

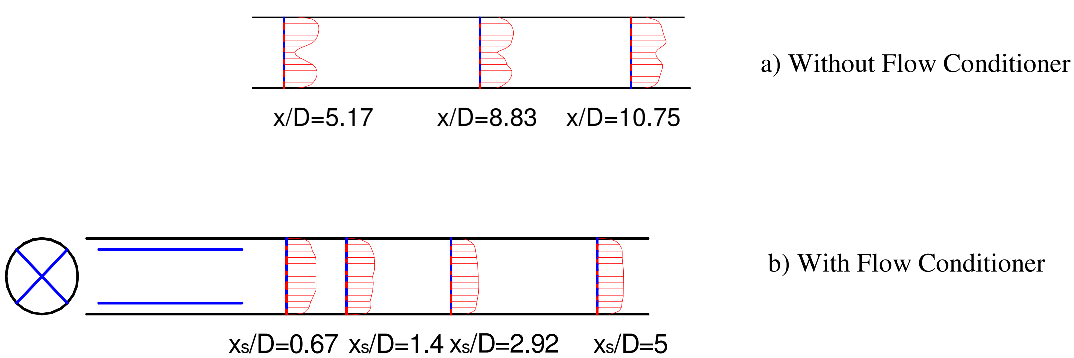

Figure 5. Decay of rotational airflow in a long duct without and with a flow conditioner

3. Nonuniformity

As mentioned above, at each axial location (*x*/D), velocities at about 10 radial locations (r/R's) were measured. Then we have an average velocity (V) and a standard deviation (σ) for each *x*/D. A nonuniformity parameter was defined to quantitatively compare the rotational effect at different *x*/D's under different operational conditions. The nonuniformity parameter is defined as the standard deviation over the average value of the velocities. The higher the standard deviation over the average, the higher the level of nonuniformity is.

Figure 6 shows some examples of the variation of σ/V with average Reynolds number. In most cases of rotational airflow, σ/V dropped slightly with average Reynolds number. For conditioned airflow, the variation is negligible. For any fixed point in the duct, the value of σ/V did not change significantly with fan power. This also explained that the velocity profile mentioned above was similar at different average Reynolds numbers (velocities). Therefore, only the average σ/V was used for each Reynolds number to quantitatively illustrate the nonuniformity at each *x*/D. Figure 7 is based on this idea.

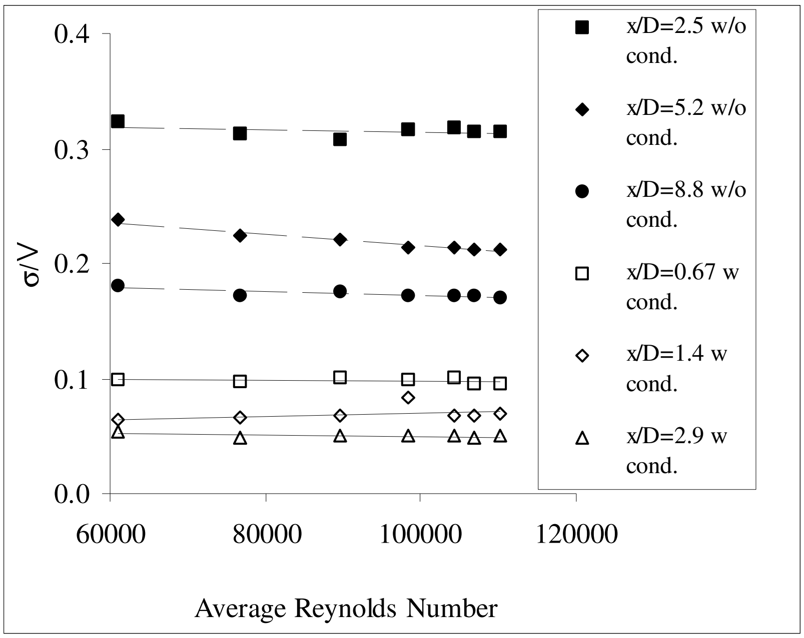

Figure 6. Nonuniformity of velocity under test conditions (V is the average velocity, σ is standard deviation of velocity. *x*/D is the axial location over diameter of the duct)

Figure 7 compares the nonuniformity of airflow in duct. Each point in Figure 7 is the average of 7 values of σ/V's as shown in Figure 6. Within the accuracy in the tests, σ/V of air velocities at any axial location in round ducts was higher than 0.1 (or 10%) (for *x*/D<10.5) for rotational air flow without conditioner. By fitting the curve, the relationship between σ/V and *x*/D can be described using the following equation.

$$\frac{\sigma}{V} = 0.91\exp\left(-0.194\frac{x}{D}\right) \qquad (R^2=0.95) \qquad (3)$$

Flow after the flow conditioner (for *x*/D>2/3) is always less than 0.1. The experimental results here show that airflow approached uniformity level of 0.05 (5%) after *x*/D=1.5. On the other hand, even with a flow conditioner, it is not recommended to take samples right after the conditioner. Taking samples at *x*/D>1.5 after the flow conditioner is a safe choice, unless it is impractical like in the situations reported by Marsh *et al*. (2003). The fitting curve gives the relationship between σ/V and *x*/D as,

$$\frac{\sigma}{V} = 0.081\left(\frac{x}{D}\right)^{-0.33} \qquad (R^2=0.89) \qquad (4)$$

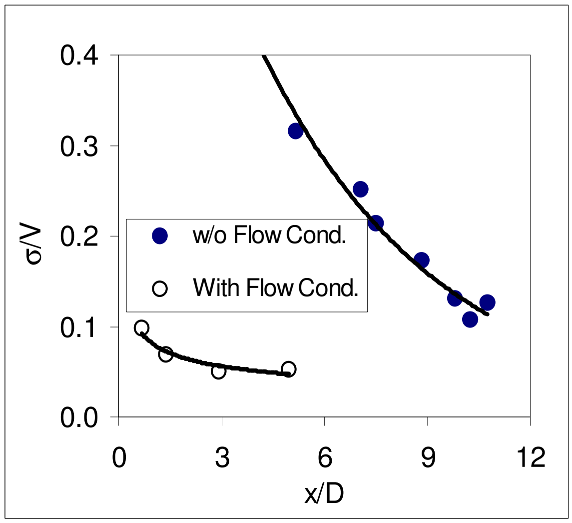

Figure 7. Nonuniformity of velocity over *x*/D for flow with and without flow conditioner (V is the average velocity, σ is standard deviation of velocity. *x*/D is the axial location over diameter of the duct)

4. Steadiness

Another parameter to evaluate the effect of the flow conditioner is flow steadiness. In this paper steadiness of the flow is characterized using centerline dimensionless velocity, V_c/V. Where, V_c is air velocity at the centerline, and μ is the average velocity at *x*/D. It shows the velocity change along the duct. For tested conditions without flow conditioner, the centerline axial velocity keeps increasing along the duct length (see Figure 5).

Figure 8 gives an example when average Reynolds number is 1.14×10^5. It shows that the centerline velocity approached the average velocity as x/D increased. For airflow without flow conditioner, the axial velocity kept increasing and approaching steady state, but it did not reach steady state within our test conditions (*x*/D<10.5). For flow after conditioner, the centerline velocity number reached the average velocity quickly after *x*/D=1.5. It indicates that the flow conditioner made the rotational air flow quickly approach steady state.

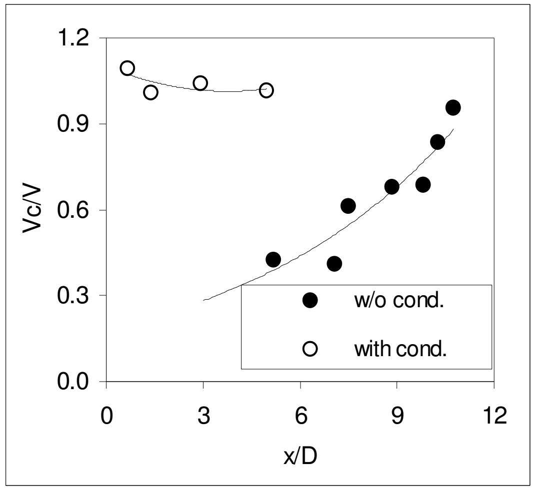

Figure 8. Steadiness over *x*/D for flow with and without flow conditioner (Vc/V is the velocity at the centerline over average velocity at *x,* and *x*/D the axial location over diameter of the duct)

Overall, the tests confirmed that it is impractical to decay the rotational airflow in ducts to an acceptable level simply by using long ducts.

5. **Particle Concentration Steadiness and Uniformity**

Steadiness at each location as illustrated in Figure 1 was characterized by the standard deviation over the average concentration (σ/μ) of the 10 samples. Steadiness of particle concentration at each of the 5 locations is shown in Figure 9. For each location, as seen in the graph, the standard deviation of the average particle concentration is less than 15% with majority less than 10%. The steadiness of the turntable might also have contributed to the variation.

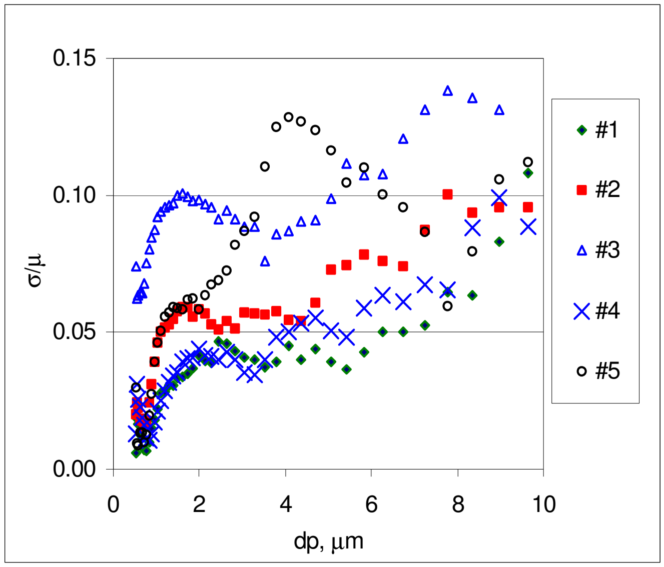

Figure 9. Steadiness downstream the flow conditioner (Re=2.3×10^5) (σ is standard deviation and μ is the average particle concentration; d_p is the particle diameter)

At each radial location, there is an average particle concentration calculated from the 10 samples. Then the average particle concentration at the axial location (x/D=2) is calculated based on the 5 concentrations at the 5 radial locations. The average particle size distribution at x/D=2 is shown in Figure 10. Each error bar marks the corresponding standard deviation (σ). The corresponding standard deviation over average (σ/μ) is also shown in the same graph on the right y-axis. As seen in the graph, σ/μ is less than 10% for any particle that is smaller than 10 μm. A higher σ/μ is associated with lower particle concentrations. Note that at lower particle concentration, the sampling efficiency of APS 3321 is low, and it introduces errors. This is out of the scope of this paper and will be addressed separately. On the basis of mass concentration, the average concentration of the 5 locations was μ=8.42 mg/m^3 with a standard deviation of σ=0.075 mg/m^3, which indicates that σ/μ is less than 1%.

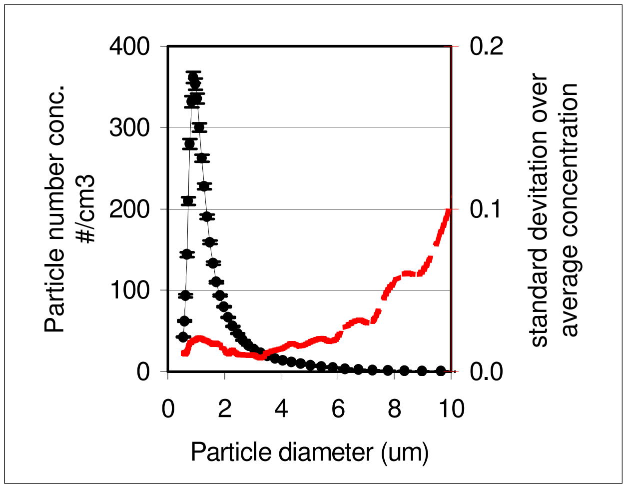

Figure 10. Uniformity downstream the flow conditioner (5 locations, Re=2.3x10^5)

6. Comparison with previous studies

Results in this paper also expanded the database of study of rotational flow in long ducts. A comparison of previous tests with ours is shown in Table 1.

Table 1 Comparison of test conditions with previous studies

	Kreith & Sonju (1965)	McManus *et al.* (1985)	Laribi *et al.* (2001)	This research
Fluid	Water	Nitrogen (N_2)	Gas (Petroleum)	Room Air
Duct Diameter	25.4 mm	101.6 mm	101.6 mm	152.4 mm, 203.2 mm
Reynolds #	10^4 - 10^5	8-14x10^5	3-11x10^4	7-11.4x10^4
Swirl generation	Tape-induced	Heat exchange and 90^o elbow	Double 90^o elbow	Centrifugal Device
Axial location for acceptable level	*x*/D>50 (tested)	*x*/D=200 (Predicted from fitting equation)	*x*/D=90 (tested)	*x*/D>>10. (*x*/D<2 with flow conditioner)

Table 1 shows that rotational flow exists in many general engineering applications. Rotational flow could be created by, but not limited to, 1) bending vanes, such as vanes used in cyclones, 2) sudden changes in flow direction, including 90^o bends (McManus *et al*. 1985) and double 90^o elbows (Laribi *et al.*, 2001), and 3) air flow downstream of a centrifugal device such as a centrifugal fan. The rotational airflow will affect the accuracy in airflow rate measurement using an orifice meter (Laribi *et al.*, 2001) as well as dust concentration measurements using isokinetic sampling techniques, as addressed in this paper.

Summary and Conclusions

This research investigated decay of centrifugal device created rotational airflow in round ducts under room condition. An example velocity profile was first given to visualize the airflow inside the ducts with and without flow conditioner. Then the uniformity and steadiness of the airflow were compared. As a case study, the dust concentration distribution after the flow conditioner was presented. The following conclusions can be drawn from this study:

1. Centrifugal device has negligible effect in creating rotational flow on upstream airflow in round ducts. The same device can create very strong nonuniform and unsteady rotational airflow in downstream ducts.

2. Once a rotational airflow is created in a duct, it is impractical to lessen it to an acceptable level simply by using long ducts.

3. The nonuniformity parameter of the airflow at certain point in a duct is nearly independent of the average Reynolds number. When $\sigma/\mu \leq 0.05$, the airflow is considered uniform.

4. A simple flow conditioner can significantly reduce the airflow swirl. The flow conditioner with length to diameter ration (L/D) of 2 lessens the rotational airflow to an acceptable level quickly within x/D=2. Duct concentrations at 5 radial locations at x/D=2 downstream the conditioner was observed uniform (σ/μ<0.01). On the other hand, it is not recommended to take samples right after the flow conditioner.

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Nomenclature

d_p	Particle diameter (µm)
D	Duct diameters (m)
l	Characteristic dimension (m)
L	Length of the duct, length of the flow conditioner (m)
r	Radial location (m)
R	Radius of the duct (m)
r/R	Dimensionless radial location
Re	Reynolds numbers
V	Average fluid velocity (m/s)
Vc	Velocity along the center line (m/s)
Vr	Velocity at radial location at r (m/s)
x	Axial location of the duct (m)
x/D	Axial location to diameter ratio
σ	Standard deviation of velocity (m/s), particle concentration (number/cm^3)
µ	Average value particle concentration (number/cm^3)
σ/µ	Non-uniformity of velocity or particle concentration
v	Kinematic viscosity of the fluid (m^2/s).