Method for Measuring Fan Ventilation Rates in Livestock Buildings—Velocity Pressure Measurement

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ABSTRACT

This study was conducted to develop and evaluate a method that can be used to continuously measure fan ventilation rates in livestock buildings. The method involved measurement of velocity pressures using two tubes with holes. The holes in one tube faced directly upstream to sense total pressure, while those in the second tube faced directly downstream to sense static pressure. Positioning the measuring unit upstream of the test fan and using a 27 in. × 27 in. (68 cm × 68 cm) duct provided the best results. For this setup and at a static pressure of 0.10 in. water (25 Pa), the absolute deviations of predicted values from measured values ranged from 18 cfm (31 m³/h) to 52 cfm (88 m³/h) for the axial fans tested. Additionally, this setup had only slight effects on fan performance. This study demonstrated the potential of using the method for ventilation rate measurement in livestock buildings.

INTRODUCTION

The ventilation rate through a building is considered to be one of the most important parameters in indoor environment control in livestock buildings (Berckmans et al. 1991). Many methods have been developed to measure ventilation rate in livestock buildings. These methods include using a tracer gas (e.g., Leonard et al. 1984), monitoring carbon dioxide concentration changes (e.g., Feddes et al. 1984) or temperature changes (Barber et al. 1988), and using a ventilation rate sensor (e.g., Berckmans et al. 1991). The ventilation rate sensor developed by Berckmans et al. (1991) appears to be a simple and reliable means for field measurement of ventilation rate in livestock buildings. This sensor consists of a two-blade impeller, which receives energy from the air movement and converts this into rotational energy. This ventilation rate sensor was used in a chimney, with a diameter of 20 in. (50 cm), of a livestock building. The sensor had a reported accuracy of ±35 cfm (±60 m^3/h) in a range from 120 cfm to 2940 cfm (200 m^3/h to 5000 m^3/h) for fan static pressure differences of 0 in. H_2O to 0.5 in. H_2O (0 Pa to 120 Pa).

There is a need to develop other simple and reliable methods to continuously measure fan ventilation rates in livestock buildings. This study was conducted to investigate a method that can be used to continuously measure fan ventilation rates in mechanically ventilated livestock buildings. The method involved measurement of the velocity pressure in a duct downstream or upstream of an exhaust fan. The performance characteristics of the measuring unit, as affected by the position (downstream or upstream) and the size of the test fan, were established.

METHODS AND MATERIALS

Description of the Method

This method was similar to the pitot-traverse method of measuring flow rate through a duct. With the pitot-traverse method, continuous measurement of flow rate is extremely difficult to accomplish. Instead of using the pitot-traverse method, two tubes with holes were used to measure the apparent mean velocity pressure in the duct (Figure 1). Each tube had a diameter of 0.3 in. (8 mm) and a length of 27 in. (688 mm). Each had a series of holes, which were 0.06 in. (1.5 mm) in diameter and 1.4 in. spaced apart (35 mm). The tubes were installed in a square duct, and tubes were oriented so that the holes in one tube faced directly upstream to sense total pressure and those in the second tube faced directly downstream to sense static pressure. A differential pressure transducer was used to measure velocity pressure, which is the difference between the total and static pressures. The pressure transducer had a 0 in. to 5.35 in. of H_2O (0 Pa to 1,333 Pa) range,

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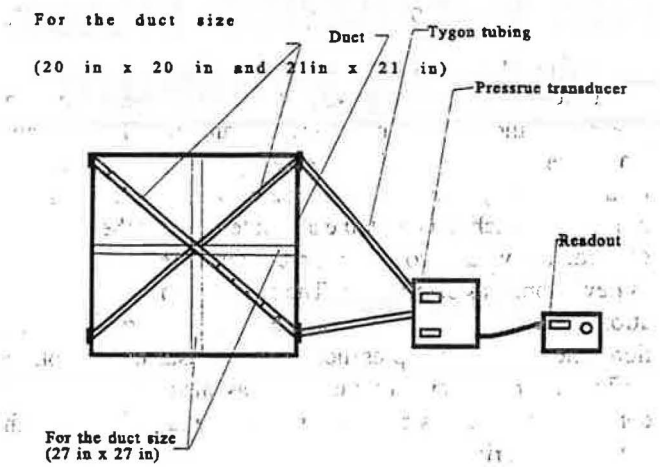


Figure 1 Schematic diagram of the flow-measuring device (not drawn to scale).

0.003 in. H_2O (0.67 Pa) resolution, and a 0 V to 1 V output. The voltage output was recorded using a datalogger. The pressure transducer was calibrated by using a micromanometer that had a resolution of 0.001 in H_2O (0.25 Pa).

Test Facility

The flow measurement device was tested by using a wind tunnel test chamber, which was designed and constructed according to ASHRAE Standard 51-1985 (ASHRAE 1985). The test chamber was a baffled, multi-nozzle chamber used for testing ventilation fans and passive devices, such as ventilation diffusers. Airflow through the chamber was generated using a belt-driven centrifugal fan driven by a 5 hp (3.7 kW) three-phase motor. Supply airflow rate and the corresponding static pressure across the test fan were adjusted remotely using an adjustable frequency motor controller. The test fan and the flow-measuring device were mounted on a board that was attached to the end of the chamber with toggle clamps and sealed with weather stripping and duct tape.

The static pressure drop across the nozzle bank was used to calculate the fan airflow rate. The airflow rate was calculated using a spreadsheet program that was developed based on the ASHRAE Standard 210-85 (ASHRAE 1985). To determine flow rate, the following parameters were measured or noted: (1) dry- and wet-bulb temperatures, (2) atmospheric pressure, (3) pressure drop across the nozzle, and (4) size and number of open nozzles. Dry-bulb and wet-bulb temperatures were measured using an aspirated psychrometer with a ±0.9°F (±0.5°C) accuracy. Atmospheric pressure was measured using a mercurial barometer with a ±0.1 in. H_2O (±25 Pa) accuracy. Static pressure drop across the nozzle bank was measured using the micromanometer mentioned above. Accuracy of the flow rate measurement was ±2% of the measured.

Experimental Design

To establish the characteristics of the measuring unit, velocity pressures were measured at different airflow rates using the wind tunnel and a test fan. The performance of the measuring device could be affected by several factors, including the size of the test fan, the length and cross section of a duct in which the tubes were installed, the position with respect to the test fan (downstream and upstream), and duct size. Eight tests were conducted to determine the effects of the above factors (Table 1).

TABLE 1
Experimental Design for Evaluating the Velocity Pressure Method[a]

Test Case	Fan Size in.	Fan Size cm	Position	Duct Dimension in. × in. × in.	Duct Dimension cm × cm × cm	Distance From Test Fan in.	Distance From Test Fan cm
1	12	30	Downstream	20 × 20 × 58	51× 51 × 147	44	112
2	16	41	Downstream	20 × 20 × 58	51 × 51 × 147	44	112
3	12	30	Upstream	21 × 21 x 21	53 × 53 × 53	22	57
4	16	41	Upstream	21 × 21 × 21	53 × 53 × 53	25	64
5	20	51	Upstream	21 × 21 × 21	53 × 53 × 53	25	63
6	12	30	Upstream	27 × 27 × 12	68 × 68 × 30	18	46
7	16	41	Upstream	27 × 27 × 12	68 × 68 × 30	21	53
8	20	51	Upstream	27 × 27 × 12	68 × 68 × 30	20	52
9	12	30	-	-	-	-	-
10	16	41	-	-	-	-	-
11	20	51	-	-	-	-	-

[a] Test cases 9 through 11 did not use the measurement unit and were conducted to establish the airflow characteristics of the fans.

Test fan size might affect the unit performance since the velocity distribution from one test fan might be different from another. Two test fan sizes were considered for the downstream position (12 in. and 16 in.; 30 cm and 41 cm). For the upstream position, three fan sizes (12 in., 16 in., and 20 in.; 30 cm, 41cm, and 51 cm) were used. These fan sizes were selected to bracket the range of fan sizes that are commonly used for small livestock buildings.

Tests were conducted to determine the effect of the unit position (upstream vs. downstream) on the characteristics of the measuring unit. For maintenance purposes and for ease in setting up the measuring unit in a building, a downstream position would be desirable. However, flow pattern is expected to be more uniform on the inlet side of the fan than on the outlet side. Additionally, storm hoods are sometimes used in ventilation fans for weather protection, making it difficult to use the downstream position.

The test layouts are shown in Figures 2 and 3 for the downstream and upstream positions, respectively. The supply fan, which was installed on the left end of the chamber, was used to provide desired airflow rates. The test fan, which was mounted on the right end of the chamber, was used to simulate an exhaust fan in a building. For test cases 1 and 2, a square duct with a side dimension of 20 in. (51 cm) and a length of 58 in. (147 cm) was mounted downstream of the test fan (Figure 2).

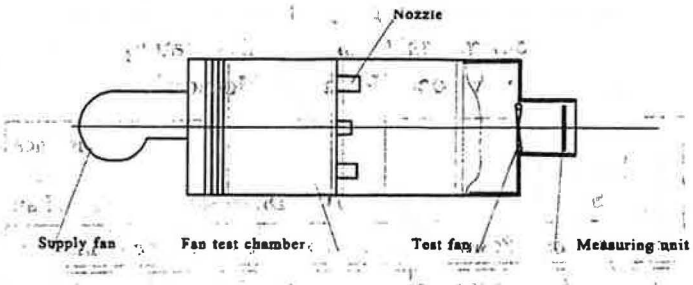


Figure 2 Layout for tests involving the downstream position (not drawn to scale).

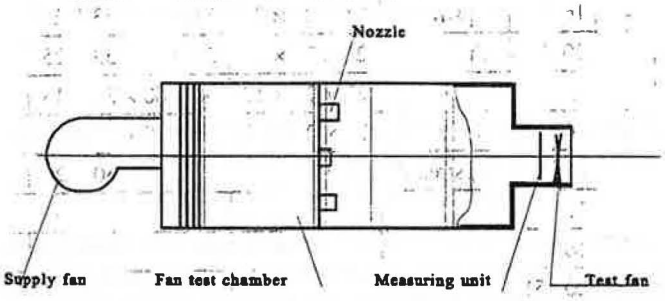


Figure 3 Layout for tests involving the upstream position (not drawn to scale).

For the upstream position and for test cases 3 through 5, a duct with side dimensions of 21 in. (53 cm) and length of 21 in. (53 cm) was used (Figure 3). For test cases 6 through 8, a duct with side dimensions of 27 in. (68 cm) and length of 12 in. (30 cm) was used (Figure 3).

Additionally, the effect of the measuring unit on the airflow characteristics of the test fan was established for all test cases (1 through 8). Test cases 9 through 11 were used to determine the fan airflow performance without the measuring unit. Results were compared with the corresponding results with the measuring unit.

Data Analysis

Volumetric flow rates were plotted against the square root of velocity pressure for each test case. The square root of velocity pressure was used because, theoretically, volumetric flow rate is directly proportional to the square root of velocity pressure. If the velocity pressure on the duct is known, airflow rate can be determined using the equation:

$$Q = A\sqrt{\frac{P_v}{\rho}}K \quad (1)$$

where

Q = volumetric flow rate, cfm (m^3/h);

A = cross-sectional area of the duct, ft^2 (m^2);

P_v = velocity pressure, in. H_2O (Pa);

ρ = air density, lb_m/ft^3 (kg/m^3); and

K = a conversion factor.

To account for errors due to velocity profile variations and other sources, Equation 1 was recast into the following form:

$$Q = C_1\sqrt{P_v} + C_2 \quad (2)$$

where C_1 and C_2 are regression coefficients.

Linear regression analysis was used to establish the linearity of the relationship between Q and $\sqrt{P_v}$. The R^2 value was determined for each regression equation to provide a measure of the goodness of fit of the linear relationship. Additionally, for each test point, the absolute value of the deviation of predicted values from measured values, herein referred to as deviation, was determined. The mean and maximum deviations provided a measure of accuracy of the regression equation. The effect of the presence of the measuring unit on the airflow characteristics of the fan was also determined by comparing the airflow characteristics of the fan with and without the measuring unit.

RESULTS AND DISCUSSION

Relationship Between Ventilation Rate and Square Root of Velocity Pressure

Results for the downstream position (test cases 1 and 2) are summarized in Figure 4 and Table 2. Using one regression equation for the two fans (12 in. and 16 in.) resulted in a mean deviation of 76 cfm (129 m^3/h) and a maximum deviation of 235 cfm (400 m^3/h) in the measurement range of 422 cfm to 3803 cfm (718 m^3/h to 6466 m^3/h) (Table 2). Additionally, at a fan static pressure of 0.10 in. H_2O (25 Pa), the deviations were 91 cfm (155 m^3/h) for the 12 in. (30 cm) fan and 31 cfm (53 m^3/h) for the 16 in. (41 cm) fan.

Results for the upstream position and for a duct size of 21 in. × 21 in. (53 cm × 53 cm) (test cases 3, 4, and 5) are presented in Figure 5 and Table 2. Fan size (12 in., 16 in., and 20 in.) had little influence on the response of the measuring unit. Using one regression equation for the three test cases resulted in a mean deviation of 35 cfm (60 m^3/h) and a maximum deviation of 118 cfm (201 m^3/h) in the measurement range of 266 cfm to 4275 cfm

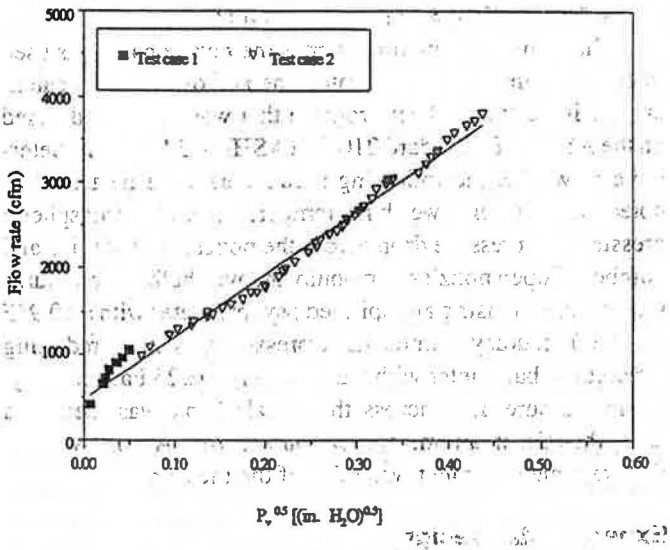


Figure 4 Flow rate vs. $\sqrt{P_v}$ for the downstream position [test case 1 (12 in. fan) and 2 (16 in. fan)].

TABLE 2
Relationship Between Flow Rate and $\sqrt{P_v}$ for Test Cases 1 through 8[a]

Test Case	Regression Equation	R^2	Std. Err. (cfm)	Mean Dev. (cfm)	Max. Dev. (cfm)
Test Cases 1 and 2[b]	$Q = 7296.5\sqrt{P_v} + 471$ (3)	0.989	92	76	235
Test Cases 3 through 5[c]	$Q = 7142.3\sqrt{P_v} + 68.2$ (4)	0.998	47	35	118
Test Cases 6 through 8[d]	$Q = 11944.5\sqrt{P_v} + 3.4$ (5)	0.998	49	36	137

[a] Q is the fan airflow rate (cfm); P_v is the velocity pressure (in. H_2O).
[b] Downstream position.
[c] Upstream position and 21 in. × 21 in. duct.
[d] Upstream position and 27 in. × 27 in. duct.

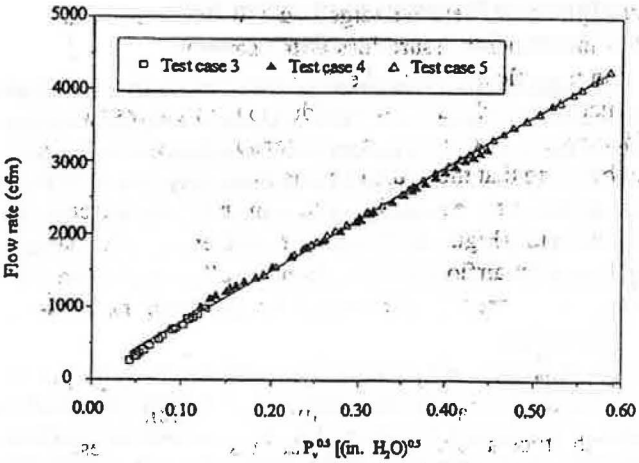


Figure 5 Flow rate vs. $\sqrt{P_v}$ for the upstream position and 21 in. × 21 in. square duct [test cases 3 (12 in. fan), 4 (16 in. fan), and 5 (20 in. fan)].

(452 m^3/h to 7269 m^3/h). For a fan static pressure of 0.10 in. H_2O (25 Pa), the deviations were 28 cfm (48 m^3/h) for the 12 in. (30 cm) fan, 55 cfm (94 m^3/h) for the 16 in. (41 cm) fan, and 9 cfm (15 m^3/h) for the 20 in. (51 cm) fan.

Results for the upstream position and duct size of 27 in. × 27 in. (68 cm × 68 cm) (test cases 6, 7, and 8) are shown in Figure 6 and Table 2. Fan size also had little influence on the measuring unit. Using one regression equation for test cases 6 through 8 resulted in a mean deviation of 37 cfm (63 m^3/h) and a maximum deviation of 137 cfm (233 m^3/h) in the measurement range of 264 cfm to 4234 cfm (449 m^3/h to 7199 m^3/h). For a fan static pressure of 0.10 in. H_2O (25 Pa), the deviations were 36 cfm (61 m^3/h) for the 12 in. (30 cm) fan, 52 cfm (88 m^3/h) for the 16 in. (41 cm) fan, and 18 cfm (31 m^3/h) for the 20 in. (51 cm) fan.

The differences in performance characteristics between the downstream (Figure 4) and upstream (Figures 5 and 6) positions can be explained by the difference in flow distribution upstream and downstream of the test fan. Air distribution is expected to be more uniform before passing through the fan than after passing through the fan. Downstream of the fan, the flow distribution

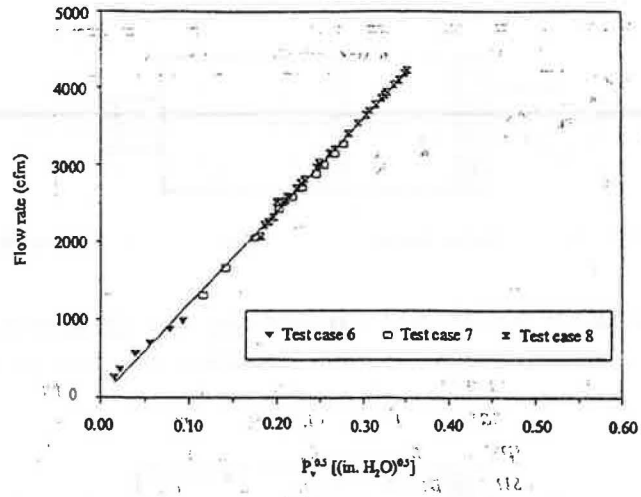


Figure 6 Flow rate vs. $\sqrt{P_v}$ for the upstream position and 27 in. × 27 in. square duct [test cases 6 (12 in. fan), 7 (16 in. fan), and 8 (20 in. fan)].

depends on fan size. Hence, the upstream position would result in a more accurate linear relationship between Q and $\sqrt{P_v}$. For the upstream position, duct size had little effect on the measuring unit (Figure 5 vs. Figure 6). The R^2 values and the deviations were practically the same for the two duct sizes (Table 2).

Effect of the Measuring Unit on the Airflow Characteristics of the Test Fans

From fan laws, the airflow rate produced by an axial fan at a certain voltage is sensitive to the static pressure that must be overcome by the fan (Berckmans et al. 1991). Consequently, it is important to minimize the pressure loss caused by the presence of the measuring unit.

Results are summarized in Figures 7, 8, and 9 for the 12 in. (30 cm), 16 in. (41 cm), and 20 in. (51 cm) axial fans, respec-

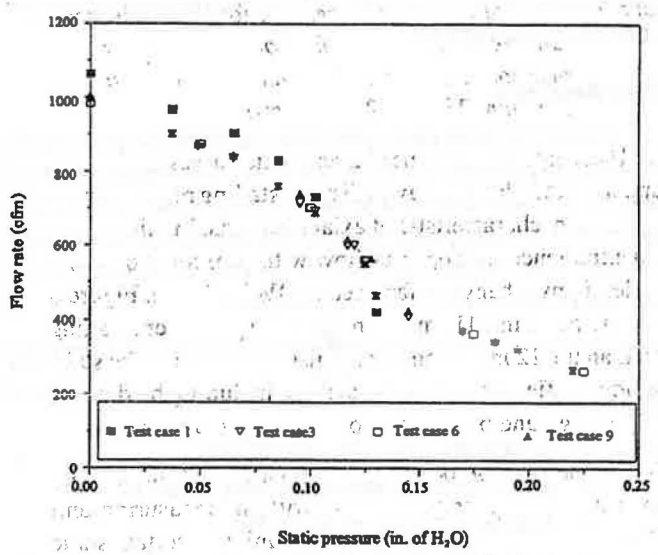


Figure 7 Flow rate vs. static pressure for the 12 in. test fan [test cases 9 (fan only), 1 (downstream position), 3 (upstream position, 21 in. × 21 in. duct), and 6 (upstream position, 27 in. × 27 in. duct)].

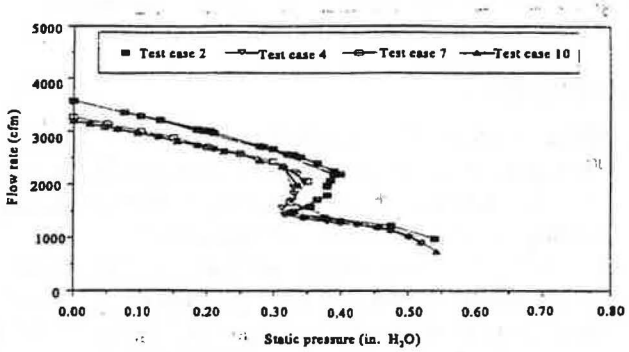


Figure 8 *Flow rate vs. static pressure for the 16 in. test fan [test cases 10 (fan only), 2 (downstream position), 4 (upstream position, 21 in. × 21 in. duct), and 7 (upstream position, 27 in. × 27 in. duct)].*

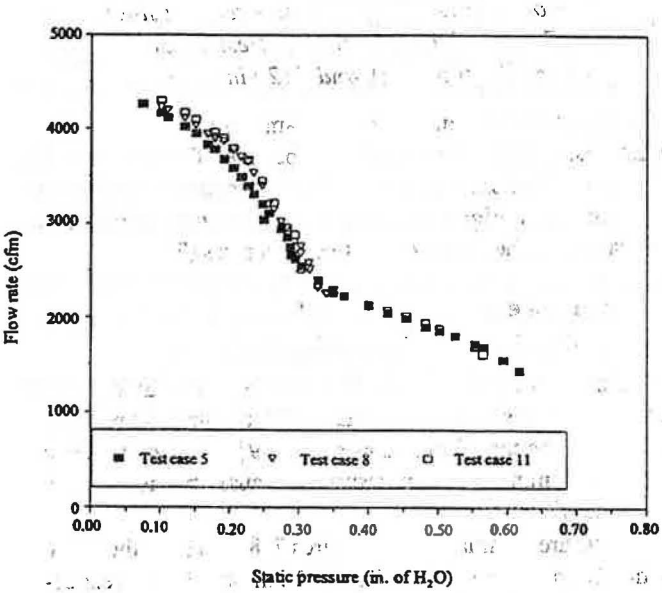


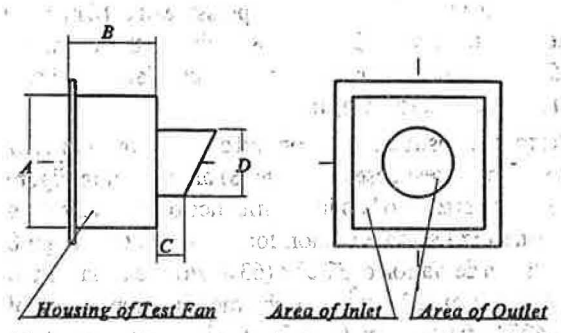
Figure 9 *Flow rate vs. static pressure for the 20 in. test fan [test cases 11 (fan only), 5 (upstream position, 21 in. × 21 in. duct), and 8 (upstream position, 27 in. × 27 in. duct)].*

tively. These figures show the characteristic curves of the fans. It is apparent that the fans have a distinct stalling region, which is an important characteristic of axial flow fans. In this region, the fan experiences a drop in airflow with additional pressure and little, if any, change in fan speed (MWPS 1990). Figure 8 shows that the 16 in. (41 cm) fan had a more prominent stalling region than the 12 in. (30 cm) or 20 in. (51 cm) fans. The stalling region is affected by several factors, including blades and housing design and presence of obstruction to airflow.

When the unit was positioned downstream of the test fan, airflow rate was higher for the case with the measuring unit than for the case without the measuring unit at a certain static pressure. For example, for the 12 in. (30 cm) fan and at a static pressure of 0.1 in. H_2O (25 Pa), airflow rates were 690 cfm (1172 m^3/h) without the measuring unit and 733 cfm (1246 m^3/h) with the measuring unit (Figure 7). For the 16 in. (41 cm) fan, at a static pressure of 0.1 in. of H_2O (25 Pa), airflow rates were 2973 cfm (5052 m^3/h) without the unit and 3288 cfm (5587 m^3/h) with the unit (Figure 8). It appears that the duct on which the unit was mounted tended to reduce the turbulence at the outlet side of the fan. Similar observations were noted by Ford et al. (1993), who reported that discharge cones and well-designed housing can improve fan efficiencies by 15% or more.

For the upstream position, the presence of the measuring unit had little influence on the airflow characteristics of the 12 in. (30 cm) and 16 in. (41 cm) fans for both duct sizes (Figures 7 and 8). However, for the 20 in. (51 cm) fan, the presence of the unit affected the airflow characteristics of the fan when a small duct was used (Figure 9). For example, at a static pressure of 0.10 in. H_2O (25 Pa) and using the small duct, airflow rates for the 20 in. (51 cm) fan were 4298 cfm (7303 m^3/h) without the measuring unit and 4174 cfm (7092 m^3/h) with the measuring unit. When the larger duct was used, the unit resulted in smaller pressure loss at the same flow rate. The reduction in airflow rate for the 20 in. (51 cm) fan when the small duct was used was primarily due to the flow restriction provided by the duct on which the sensor was mounted. It should be noted that the housing of the axial fans consisted of a square section on the inlet side of the fan and a round section on the outlet side (Figure 10). Using the smaller duct upstream tended to restrict airflow because of the smaller area covered by the duct compared to the area of the inlet side of the housing.

The above results indicate the potential of the method in measuring fan ventilation rates in livestock buildings, especially for the type and sizes of axial fans tested. These results also indicate that the upstream position would provide more accurate results for measuring fan ventilation rates than the downstream position. For the upstream position, pressure loss associated with the presence of the measuring unit can be



Dimensions	Fan Size 12	16	20
A	16.0	21.0	25.0
B	11.5	11.5	12.5
C	4.7	4.7	5.5
D	12.3	16.0	20.0

Figure 10 *Test fan housing. All dimensions are in inches.*

minimized by using a square duct with side dimensions of 27 in. (68 cm) for the sizes of the axial fans tested.

Future Work

This research demonstrated the potential of using this method in measuring fan ventilation rates for the axial fans tested. The applicability of the method for other types and models of axial fans needs to be investigated. Additionally, field testing of the measuring unit will be conducted. The field test will show whether or not consistency in measurement accuracy and reliability of the device can be achieved in a livestock building application. It will also give an indication as to how often the tubes have to be cleaned to minimize clogging of tubes. Other operational issues will become apparent in the field test.

CONCLUSIONS

A method that can be used to continuously measure fan ventilation rates in livestock buildings was evaluated. The method involved measurement of velocity pressure using two tubes with holes, which were installed on a square duct. The setup involving the upstream positioning with respect to the test fan and a 27 in. × 27 in. (68 cm × 68 cm) duct was found to be the best for the axial fans tested. At a fan static pressure of 0.10 in. H_2O (0.25 Pa), this setup had an absolute deviation of predicted values from measured values ranging from 18 cfm (31 m^3/h) to 52 cfm (88 m^3/h). Additionally, with this setup the effect of the measurement device on fan performance was small.

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