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Real-Time Airflow Rate Measurements from Mechanically Ventilated Animal Buildings

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ABSTRACT
This paper describes techniques used to determine airflow rate in multiple emission point applications typical of animal housing. An accurate measurement of building airflow rate is critical to accurate emission rate estimates. Animal housing facilities rely almost exclusively on ventilation to control inside climate at desired conditions. This strategy results in building airflow rates that range from about three fresh-air changes per hour in cold weather to more than 100 fresh-air changes per hour in hot weather. Airflow rate measurement techniques used in a comprehensive six-state study could be classified in three general categories: fan indication methods, fan rotational methods, and airspeed measurement methods. Each technique is discussed and implementation plans are noted. A detailed error analysis is included that estimated the uncertainty in airflow rate between ±5 and ±6.1% of reading at a building operating static pressure, air temperature, relative humidity, and barometric pressure of 20 Pa, 25 °C, 50%, and 97,700 Pa, respectively.

IMPLICATIONS
Gas and particulate emissions from animal housing systems are coming under increasing scrutiny. This paper discusses the measurement techniques used to assess the airflow rate in animal buildings as part of a larger six-state air emission research effort. Airflow rate measurements are critical to any study involving emissions because any error in airflow rate results in a direct error in emissions. Animal housing facilities pose a unique challenge in that multiple emission points exist, corresponding to the multiple fans used to ventilate these facilities. This paper discusses the various techniques used to assess airflow rate in multiple emission source applications and the expected airflow rate uncertainty.

INTRODUCTION
A six-state research project[1] was conducted with the purpose of determining hydrogen sulfide, ammonia, carbon dioxide, particulate matter less than 10 μm (PM_{10}), total suspended particles (TSPs), and odor emissions from selected swine and poultry housing systems in six different states. A critical variable in emission studies of this type is accurate measurement of the airflow rate through the housing system. One of the major challenges is the fact that animal housing facilities use multiple fans that vary by size and operation times, and these fans can be either single-speed or variable-speed in operation. These unique features require specialized methods for assessing the actual airflow rate through these types of buildings.

Several options exist for continuous measurement of airflow rate, although the accuracy and ease of use vary. The tracer gas method has been the technique traditionally used.[2] This method uses an inert gas to predict the dilution potential of the ventilated space and the building's airflow rate is determined through back-calculation procedures. This method suffers from inaccuracy when there is incomplete mixing and is very instrument-intensive. Another option for measuring airflow rate is using a frictionless anemometer that is slightly smaller than the fan size itself.[3,4] This method has been shown to work well for fans smaller than 70 cm in diameter. However, installations for the more common larger fans, suffer from excessive pressure drop that requires the installation of a small duct section upstream of the fan, which could be problematic in many facilities.

Determining the airflow rate in fan-ventilated animal facilities is a challenging task because of the wide variations in airflow rates required, the wide array of operating static pressures, the wide array of fan configurations, and the harsh environment for any sensing method chosen. Figure 1A shows one example fan arrangement encountered. The example shows a swine finishing facility in Texas where all fans are located at one end of the building (i.e., tunnel ventilation method). Typical variations in daily average airflow rates encountered are shown in Figure 1B. Figure 1B shows the total barn airflow rate (±SD), the outside temperature (±SD), and the resulting inside barn temperature (±SD) for a two-month period in central Iowa. With animal housing, climate control is almost exclusively handled with airflow rate changes, and as the outside climate fluctuates, so does the resulting barn airflow rate requirements.

For this multistate research project, individual investigators were given the freedom to select and customize airflow rate measurement techniques. There was, however, a single requirement given for each housing system

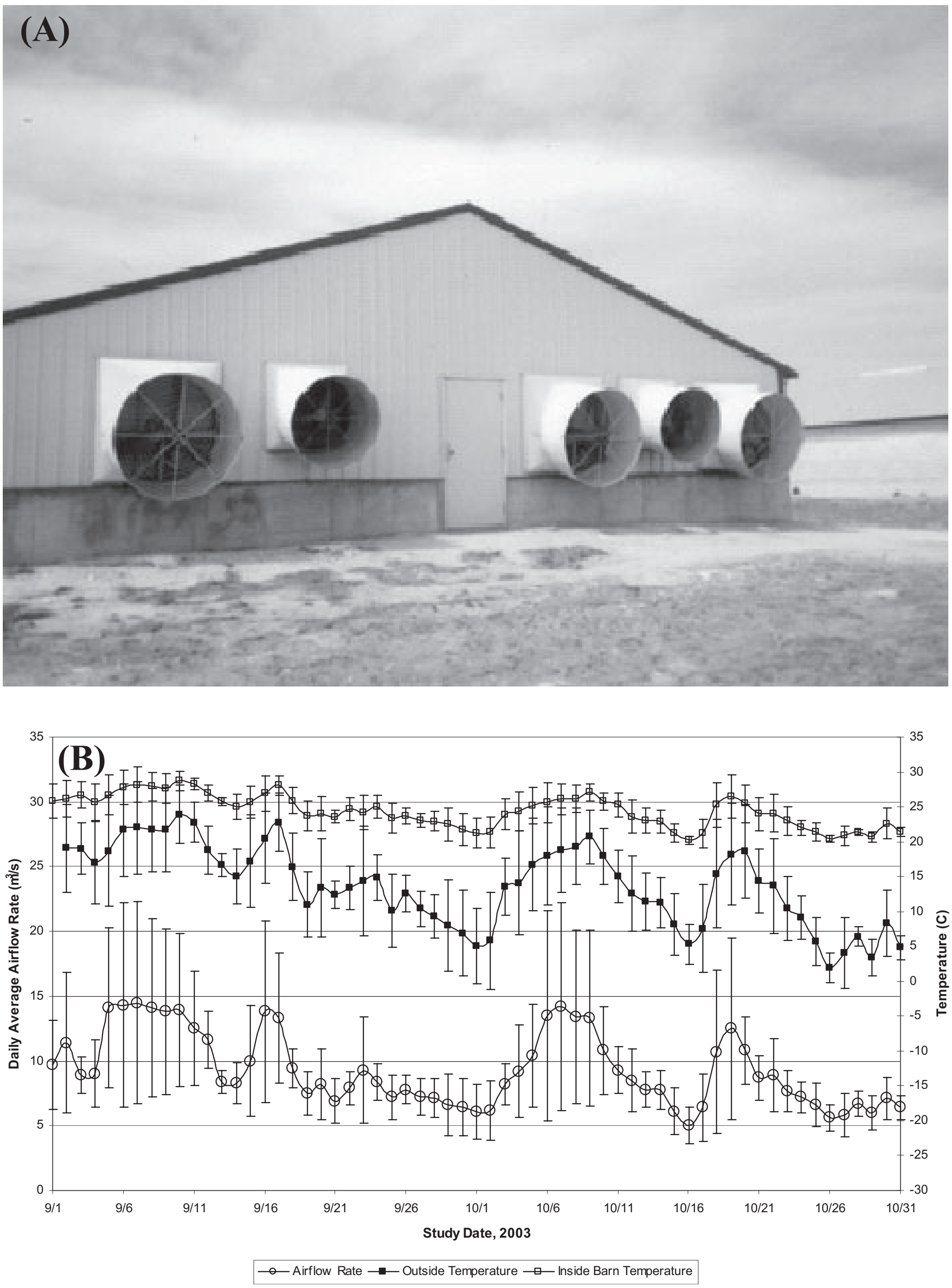


Figure 1. (A) A typical fan arrangement monitored and (B) typical variations in airflow rate as a function of climatic challenges.

Figure 2. In-field fan calibration with the FANS unit, a portable fan tester.

studied: all fans and/or fan stages had to be calibrated on-site at a range of expected operating static pressures using the Fan Assessment Numeration System (FANS) unit, a portable fan tester, described previously in Heber et al.[5] and Jacobson et al.[6] The FANS unit consists of an open-ended box that is placed in front (intake side) of (or behind if necessary) a fan as shown in Figure 2. The FANS unit has a row of five vane anemometers that traverse vertically across the face area of the fan. This gives a velocity map across the face area that is used to calculate the airflow rate entering (or leaving) the fan.[7–9] Although the FANS unit is useful for obtaining fan curves in the field, it is not applicable for continuous measurement of fan airflow rate delivery. In all discussions contained within this paper, all fan airflow rate calculations were the result of in-field calibration measurements with the FANS unit in which representative fans in a building were tested at various operating static pressures and at various variable-speed settings, where applicable. A wide array of techniques was developed for this multistate research effort, each having their own set of advantages and disadvantages. In all cases, determining the building's airflow rate was a formidable task.

The objective of this paper is to discuss the methods developed for assessing building airflow rate. In addition, a detailed analysis is presented that highlights the expected uncertainty in airflow rate measurements.

FAN AIRFLOW RATE MEASUREMENT METHODS AND EXAMPLE RESULTS

Four swine (Iowa, Minnesota, Illinois, and Texas) and two poultry (Indiana and North Carolina) housing systems were studied over a period of approximately 15 months. For each state, two side-by-side buildings were studied. Detailed descriptions of the project can be found in Heber et al.[5] and Jacobson et al.[6] In all cases, the housing systems incorporated 100% fan ventilation, a prerequisite for building selection for this research project. This requirement was imposed for the purpose of isolating emission points and to have controlled locations for determining airflow rate. The techniques for measuring a building's airflow rate can be grouped into three general methods: (1) fan indication methods (FIMs), (2) fan rotation methods (FRMs), and (3) airspeed measurement methods (AMMs). Each method is discussed below.

FIMs

FIMs were designed to determine whether a fan was on or off. With this method, no attempt was made to decipher variable-speed fan operation and thus this method was designed and suited for all single-speed fans. Iowa, Texas, North Carolina, and Indiana primarily used FIMs.

Iowa Fan Arrangement. For the Iowa housing system, a total of eight fans per building were used, arranged as two variable-speed 46-cm pit fans, one 61-cm variable-speed sidewall fan, one 61-cm variable-speed endwall fan, one 91-cm single-speed endwall fan, and three 122-cm single-speed endwall fans. The endwall fans were arranged to provide hot weather tunnel ventilation similar to the arrangement shown in Figure 1A. The eight fans were configured into seven ventilation stages. To decipher the seven ventilation stages, six of the eight fans were required to be monitored directly.

The Iowa method used limit switches arranged as sails to indicate fan operation. The method used customized sails fitted to an omnidirectional whisker switch (Model XCKP106, Telemecanique, Inc. [13.1 cm-N activation torque], or Square D, Inc., Model C54L [4.2 cm-N activation torque]) with a 28-cm long extension arm. The sail switch was used to indicate fan on/off status by allowing fan airflow rate to activate the switch. To ensure that a fan's operation would activate the sail switch, a wire connection was made between the sail switch and the back-draft louver on all fans where used. The final arrangement is shown in Figure 3. The combination of fan airflow rate and shutter opening produced adequate activation air for the sail. It should be noted that the FIM procedure applied to variable-speed fans for the Iowa arrangement was supplemented with FRM measurements as described below.

The sail switches (six total per building) were electrically arranged as a digital-to-analog (D/A) converter, which resulted in a single voltage output representative of

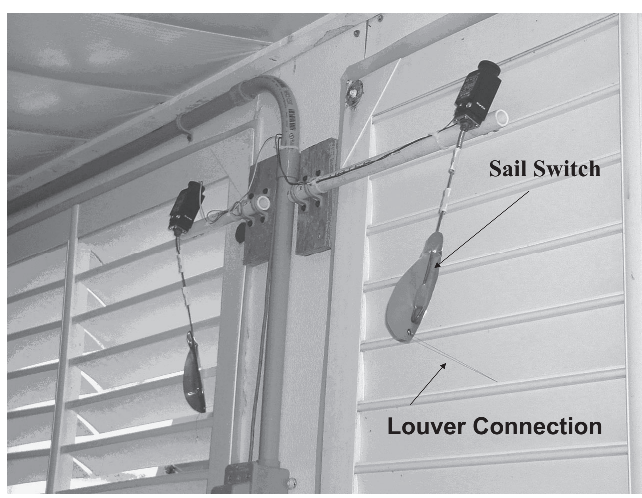


Figure 3. The Iowa sail switch arrangement. Sail switches attached to six of the eight building fans to represent one of seven ventilation stages.

Table 1. Voltage indicators for Iowa fan stages for a pig finishing building.

Switch(es) Active	Ventilation Stage	Voltage Output (V, dc)
0	0	8
S1	1	6
S1, S2	2	4
S1, S3	3	3.6
S1, S2, S3	4	2.77
S1, S3, S4	5	2.25
S1, S3, S4, S5	6	1.38
S1, S3, S4, S5, S6	7	0.82

the status of all fans in the building. The sail switches were assigned to six of the eight fans and the status of these six fans was used to represent one of seven possible ventilation stages. When a switch was activated, the developed circuit had an output voltage unique to each of the seven fan stages according to a status table (Table 1). The resulting airflow rate for a 10-day period in October 2003 is given in Figure 4 along with the variations in ambient temperature during this same period.

Texas Fan Arrangement. Two tunnel-ventilated swine finishing buildings with five total single-speed fans per building were monitored in Texas. The single-speed fans monitored were two 91-cm and three 122-cm fans distributed along the endwall (Figure 1A). The building was ventilated in four stages with the first stage consisting of the two 91-cm fans working continuously. Stages 2–4 consisted of the two 91-cm fans operating at 100% with successive stepwise 122-cm fans.

The Texas site used two FIMs to monitor fan status. A combination of sail switches similar to the Iowa arrangement coupled with fan-monitoring relays was used. Initially, only sail switches were used but errors in fan indication were found because of the inability of the fan airflow rate to deflect the sail. To remedy this situation, 110-V ac relays were added to monitor power inputs to each fan as a supplement to sail switch data. As with all other sites monitored, the FANS unit described previously was used to generate all fan airflow rates versus static pressure calibration curves. Figure 5 shows the airflow rate measurements recorded and the resulting inside building temperature for the first week in July 2003. Clearly, the airflow rate method captured the expected diurnal airflow rate changes. Figure 5 also indicates one very important detail related to determining fan airflow rates: if one used only fan test report data to estimate building airflow rate, the maximum capacity for the Texas buildings would have been approximately 37 m^3/sec, whereas the actual capacity based on in-field fan calibration using the FANS was approximately 28 m^3/sec. Therefore, the building's airflow rate capacity was approximately 76% of the published test report data and the use of test report data only in emission calculations would have overestimated emissions by 32%.

North Carolina Fan Arrangement. North Carolina monitored a broiler facility that consisted of 10 fans per building. The sidewall fans monitored consisted of one single-speed 91-cm fan and nine single-speed, belt-driven, 122-cm fans. The FIM used in North Carolina consisted of sail switch mechanisms similar to the Iowa arrangement.

Indiana Fan Arrangement. Indiana monitored a high-rise layer facility that consisted of 75 belt-driven, single-speed, 122-cm exhaust fans on each building. Ventilation air was introduced to the caged layer region from the attic through temperature-adjusted baffled ceiling air inlets over each row of cages. The 75 fans were distributed in the pit area of the high-rise layer with 37 fans distributed along the west sidewall and 38 distributed along the east sidewall. The fans were spaced 3.7 m apart in groups of

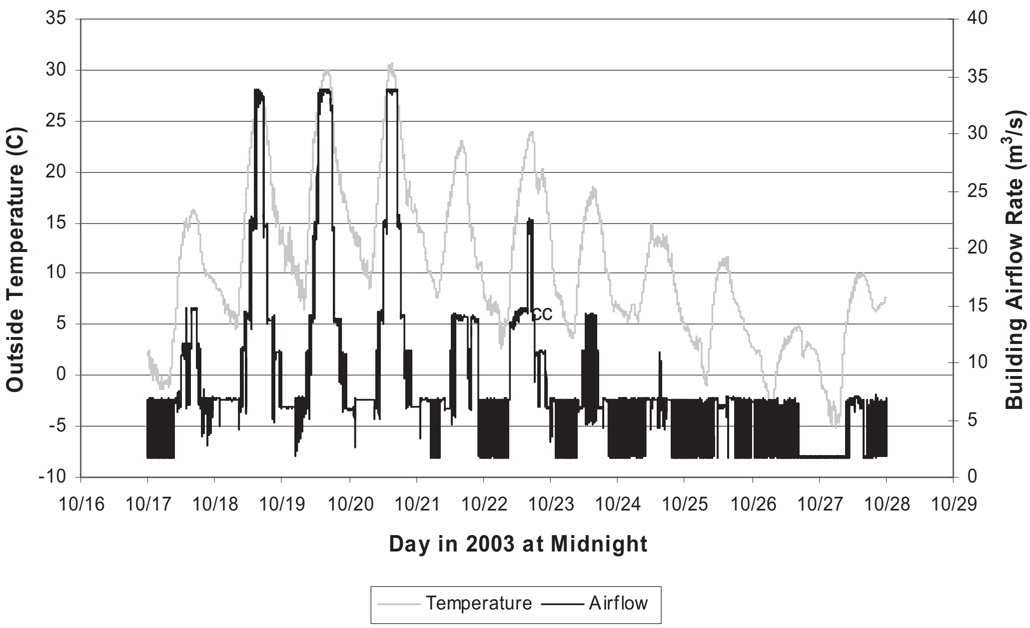


Figure 4. Example Iowa fan monitoring with the corresponding airflow rates as influenced by outdoor temperature changes.

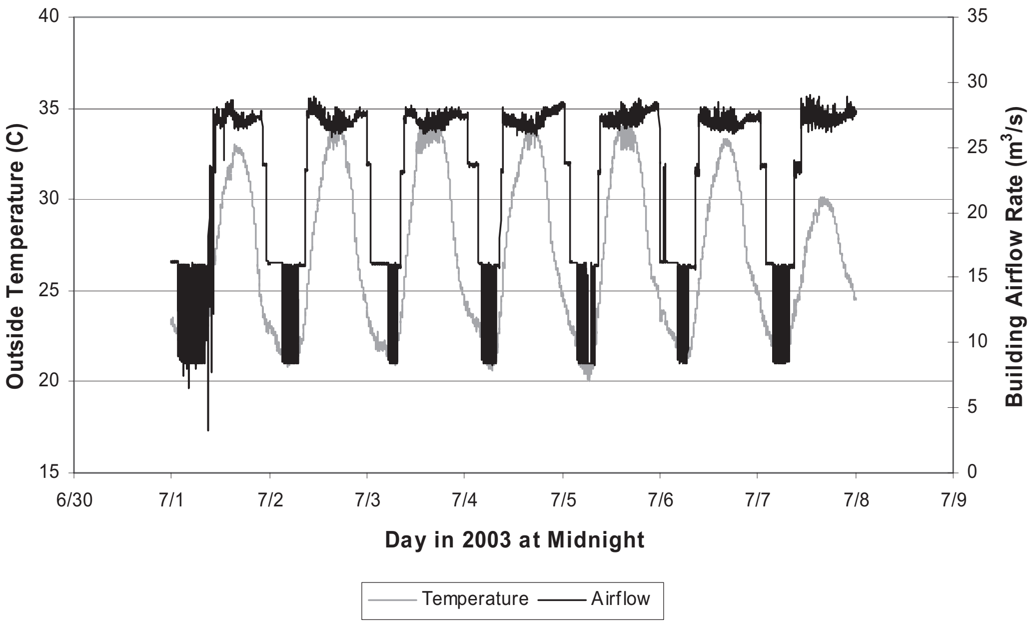


Figure 5. Example Texas airflow rate measurements for July 2003.

three or four fans and the groups were 7.3 m apart. Each building utilized nine ventilation stages as shown in Table 2.

Building static pressure was measured between the center of the manure pit and both the north and south sides of the building. The outside port was located against the outside wall directly between two fans. These pressures were different with northerly and southerly winds. Fan operation was monitored by using unused contacts of fan motor relays in 5-V dc circuits in conjunction with digital inputs of the data acquisition system. The ventilation stage operation was monitored by sensing at least two fans in parallel per stage. Figure 6 shows a 7-day response of the Indiana setup in May 2003. The contact relay response, indicating fan ventilation stage, is given along with the outside temperature. The fan ventilation stage indicator was then used to determine the operating fans, and the building's total airflow rate could be determined based on FANS calibration and the operating static pressure.

FRMs

FRMs were those techniques used to detect fan operational status and simultaneously assess the variable-speed nature of fans by direct measurement of fan RPM. Minnesota used this method exclusively and Iowa used this method for the variable-speed fans in their setup.

Minnesota Fan Arrangement. Minnesota monitored a swine gestation and breeding facility. The fans monitored for RPM consisted of five single-speed, 122-cm, belt-driven fans per building. In the breeding building, each fan was on a separate thermostat control so that as the building temperature increased, an additional fan would operate. The gestation building had an integrated heating/ventilation/cool cell controller. For this building, the 122-cm fans were staged in pairs with a differential temperature setting of 0.5 °C. In addition to the 122-cm fans, each building used one 91-cm single-speed continuous fan.

Minnesota used magnetic pickup sensors (Model GS100501, Cherry) arranged as shown in Figure 7 to simultaneously assess fan status and rotational speed. Each magnetic sensor was mounted to detect the motion of the spokes on the cast-iron fan pulley. The magnetic sensor produced a digital signal that was then converted to a dc voltage signal. The measured revolution-per-minute

Table 2. Fan numbers and ventilation stages for the Indiana high-rise caged layer building.

Stage	Number	Identification of additional fans for each stage
1	5 (continuous)	1,20,37,47,67
2	5 + 5 = 10	10,29,39,57,75
3	10 + 8 = 18	5,15,24,33,43,52,62,71
4	18 + 8 = 26	6,16,25,34,44,53,63,72
5	26 + 8 = 34	4,14,23,32,42,51,61,70
6	34 + 8 = 42	7,17,26,35,45,54,64,73
7	42 + 14 = 56	3,9,12,18,22,28,31,38,40,49,55,59,65,69
8	56 + 19 = 75	2,8,11,13,19,21,27,30,36,41,46,48,50,56,58,60,66,68,74
9	75 − 19 = 56	Evaporative pads on, Stage 8 fans off

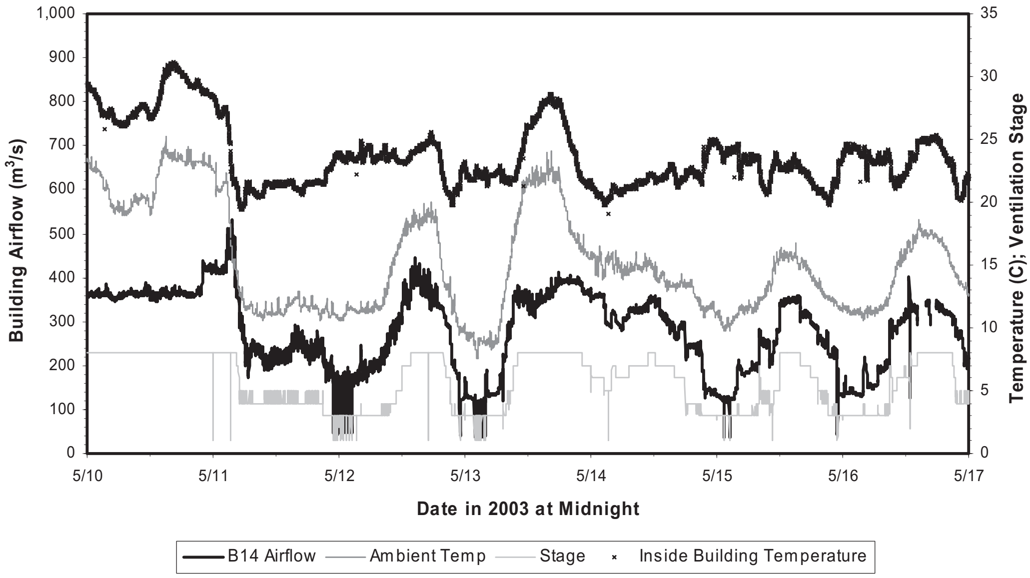


Figure 6. Example results from one of two Indiana high-rise layer facilities monitored showing ventilation stage monitoring and subsequent airflow rate calculated as affected by ambient temperature changes and the resulting temperature response inside of the building at one location.

(RPM) levels for each fan as a function of various operating static pressures were calibrated using the FANS unit and subsequent calibration curves were developed. Figure 8 summarizes the Minnesota airflow rate measurement method for one day for the gestation building (Figure 8A) and the breeding building (Figure 8B). In the gestation building, the second stage was set by the operators of the building to control two of the 122-cm fans with a tight 0.5 °C temperature differential. Such operation resulted in rapid cycling of the second stage fans, sometimes only running for less than 2 min and off for less than 2 min. The variability in early morning airflow rate shown in Figure 8A was typical of the resulting airflow rate during conditions that required more airflow rate than what the 91-cm minimum ventilation fan could provide. When ambient temperatures increased and required additional airflow rate, the operation of the fans was much more continuous. The RPM sensors were able to easily record this rapid cycling and thus produce an accurate input to calculate building airflow rate on a continuous basis.

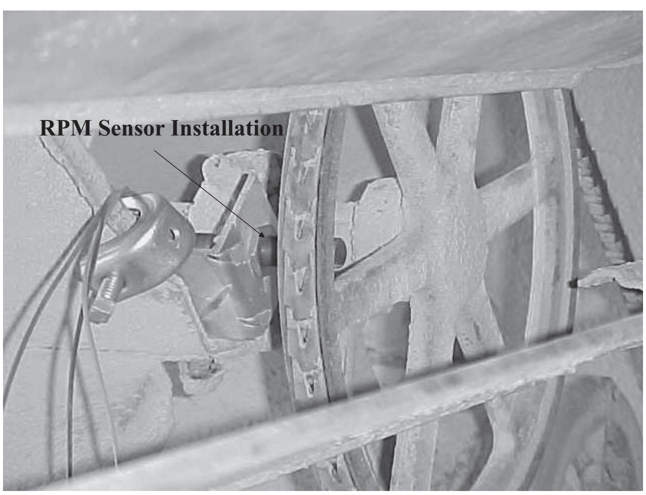


Figure 7. Installation of an RPM sensor on a fan-belt-driven fan for the Minnesota setup.

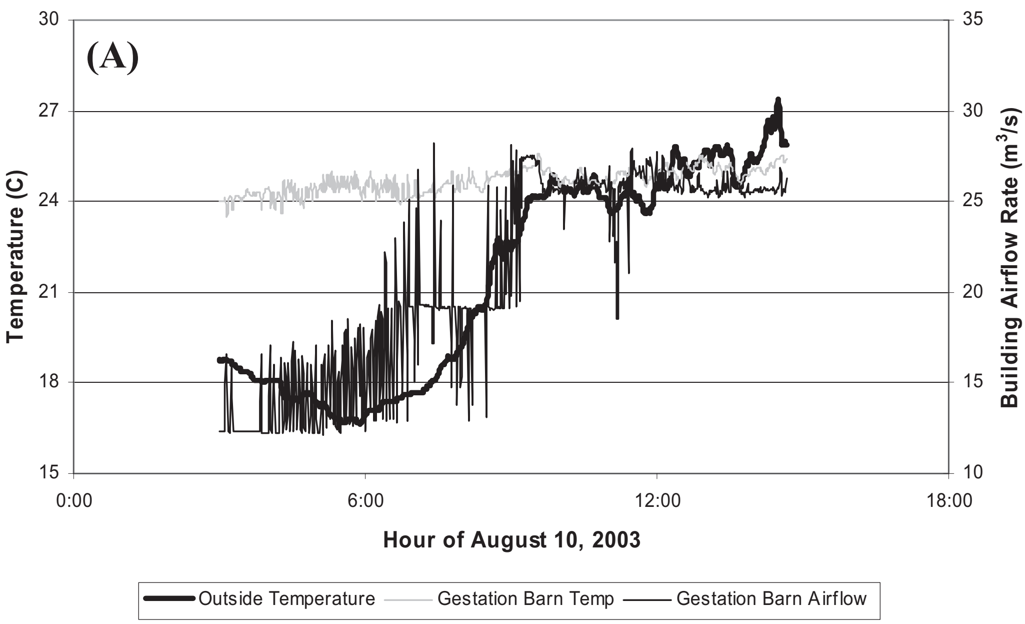
AMMs

AMMs were those techniques developed to simultaneously detect fan operational status and assess the variable-speed nature of fans by measuring a representative airspeed through the flow net of the fan.

Illinois Fan Arrangement. The Illinois site monitored was a swine farrowing facility consisting of sidewall fans that varied between 61 and 122 cm in diameter. The Illinois AMM used a single propeller (RM Young Model 27106RS with Model 08234 propeller) positioned at either the intake (Figure 9A) or exhaust side of each fan and the rotational speed of this propeller was calibrated against the fan airflow rate as a function of operating static pressure (Figure 9B).

The propeller anemometer consisted of an 18-cm vane attached to a sealed bearing dc generator that produced a 0- to 1-V dc output proportional to rotational speed. The anemometer was placed at the inlet side of the fan for all 61- and 76-cm fans and at the exhaust side for all 91-cm fans. The exact location of the anemometer in front of the fan depended on the size of the fan and the airflow rate pattern. For the two smaller fans, the anemometer was positioned as close to the fan as possible and faced upstream. With the larger fan the anemometer was placed just inside of the cage of the cone facing into the exhaust stream.

To test the anemometer's ability to measure the airflow rate of agricultural fans, the Air Movement and Control Association (AMCA) standard[10] test facility at the University of Illinois at Urbana–Champaign was used. This facility is the industry standard test site for agricultural fans. It has the capacity to test all typical fan sizes at static pressures ranging from 0 to 5 kPa, well above any typical operating pressure. Several tests were conducted to



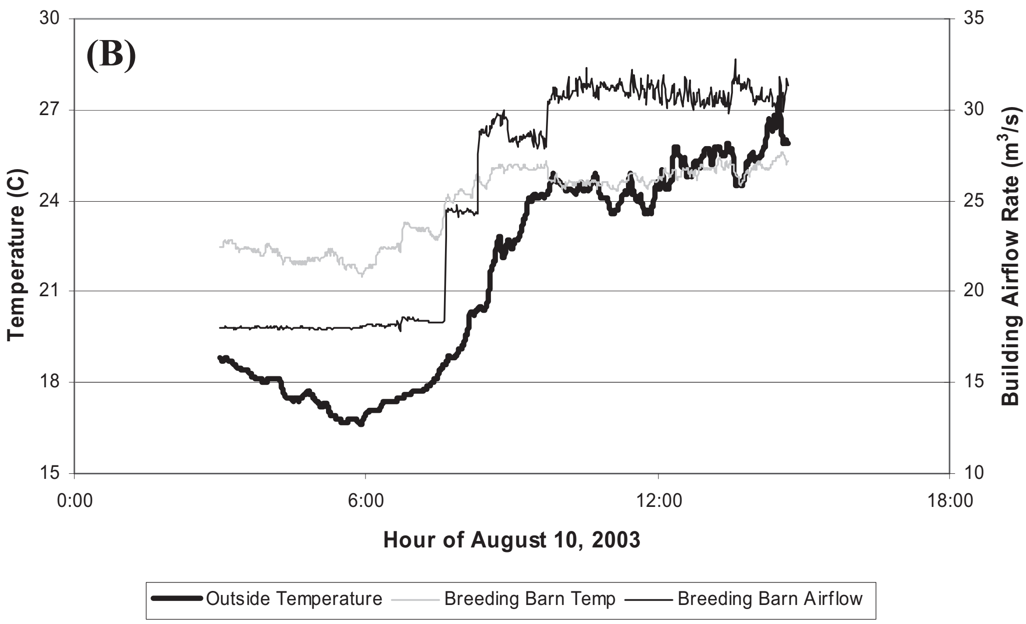


Figure 8. Example airflow rate measurements for the Minnesota (A) gestation building and (B) breeding building for August 10, 2003 as a function of outside temperature and resulting inside building temperature.

test propeller position relative to the fan's flow net and to calibrate propeller output with actual fan airflow rate. Details of this work can be found in McClure et al.[11]

Figure 9B shows the propeller anemometer output voltage as a function of fan airflow rate for the 61-cm fan tested with the propeller positioned at the four compass points about the intake side of the fan. The anemometer tested in this study showed good linearity with the fan airflow rate for all of the fans tested. On the basis of this analysis, it is best to calibrate the anemometer for the specific fan and location of interest. As expected, the anemometer response is sensitive to location relative to the fan as well as the arrangement of the building in which it is installed. It appears that the anemometer will perform well as long as the proper calibration is done. It does not appear that a general calibration curve for a certain fan curve can be offered with great accuracy. For application where high accuracy is not necessary, then general curves could be developed in the lab for use in the field.

DISCUSSION AND RECOMMENDATIONS

The emission study that formed the basis for the work described in this paper required the assessment of building airflow rate. Flexibility was given to each research team to develop methods that best suited their research site. Using this approach fostered a climate of innovation that resulted in the techniques described within this paper. Each of the methods discussed has advantages and disadvantages, and Table 3 categorizes these. For future animal housing emission studies, the recommendation would be to make as direct of a measurement as possible

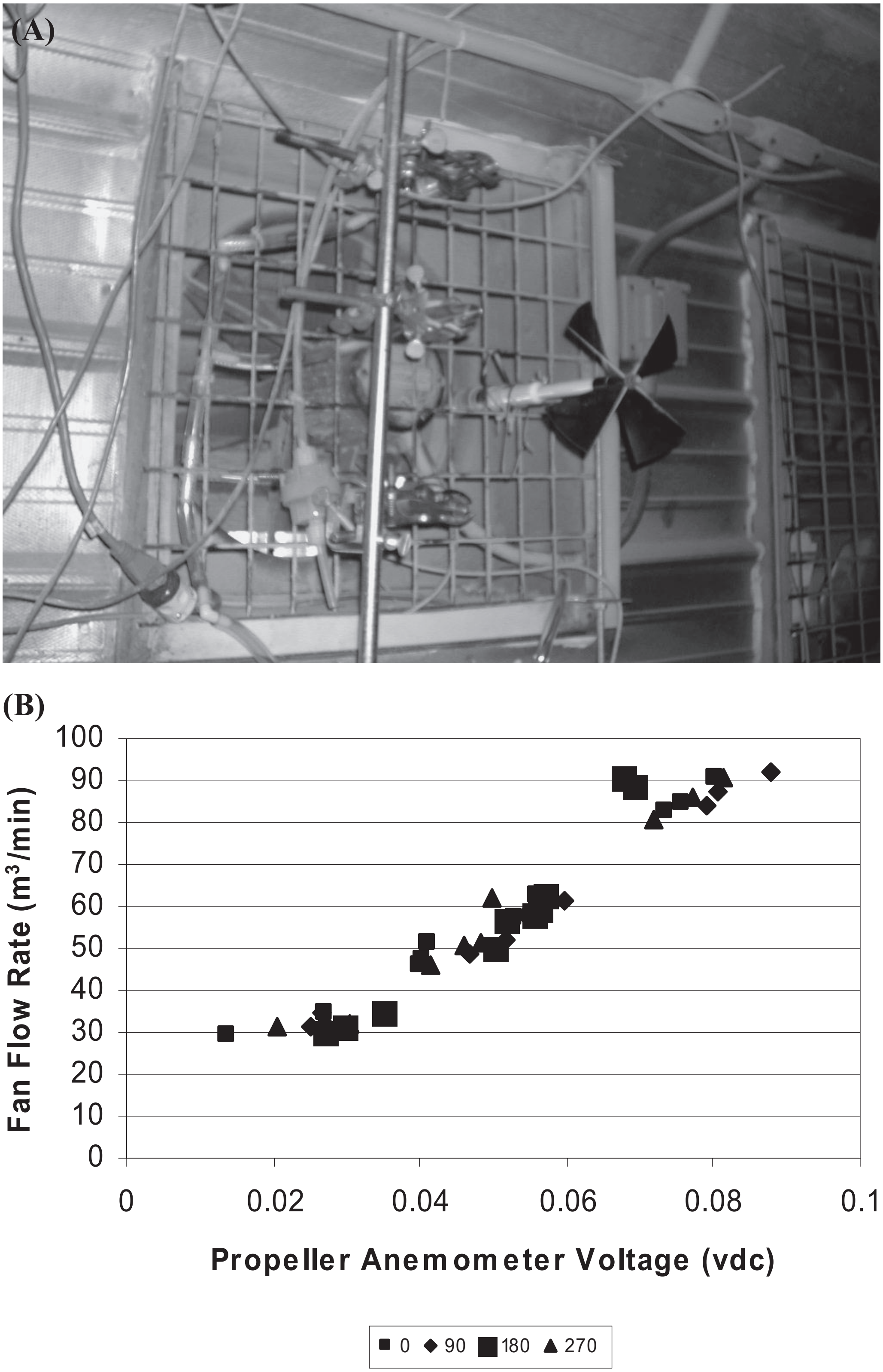


Figure 9. Example (A) propeller anemometer positioned at the intake side of a 61-cm diameter fan monitored in the Illinois setup and (B) propeller anemometer results as a function of fan airflow rate. The designations 0, 90, 180, and 270 correspond to the four compass point locations where the propeller anemometer was tested.

for fan operation and rotational properties. Using RPM sensors is recommended as the preferred method for satisfying both needs, followed by in-field calibration of fan RPM under a suitable range of static pressures. The resulting fan performance information is believed to yield the most reliable and accurate assessment of fan

Table 3. Summary of fan airflow rate measurement methods used with advantages and disadvantages of each based on experiences gained from this research project.

Category	Method	Advantages	Disadvantages
FIM	Sail switch	Simple Inexpensive (∼\$20/fan) Many fans can be configured into one analog output (as with any digital signal)	Requires attachment to a shutter Susceptible to mechanical failure Sensitive to low airflow rate events caused by high static pressure on variable-speed action Sensitive to dust buildup
	Contact relays	Inexpensive (∼\$2/fan) if dry contact available on existing relay Configurable into one analog output Can sense entire stage with one signal	Sensing a stage of multiple fans assumes all fans are functional May require fan wiring intervention If no dry contact, then \$50-\$100/fan needed for 240-V ac relays to sense fan voltage Susceptible to false positives (burnt motor, belt off, impeller off, disengaged fan motor) Susceptible to false negative (failed contact)
	Vibration Sensor	Inexpensive (∼\$3/fan) Easy to install in fan housing	Must be adjusted to match fan vibration Susceptible to mechanical failure
	Current sensor	Not very susceptible to false positives	Expensive (\$75/fan) Requires fan wiring intervention
FRM	RPM sensor	Fan status and fan rotation data in one signal Can be used to assess belt slippage	Susceptible to dust buildup Requires pulse input counters or supplemental signal conditioning Costly at ∼\$70/fan
AMM	Small propeller anemometer (SPA)	Fan status and fan variable flow in one signal Output directly readable by most DAC systems No excitation voltage required Does not require static pressure knowledge calibrated against a fan/SP combination Not susceptible to false positives Cut-in air speed an issue with variable-speed fans Bidirectional airflow measurement	Mounting can be a challenge Expensive (\$450/fan) Sensitive to dust (must be cleaned regularly) Sensitive to placement within the fan's airstream Incompatible with small fans

Notes: DAC = data acquisition and control.

airflow rates to be used in animal housing emission studies.

The airflow rate techniques used in this research were designed to yield real-time airflow rate measurements. The real-time airflow rate measurements were then used, along with gas and particulate matter concentration data, to estimate emissions. The emission rates of interest are those summarized on an hourly or daily basis. However, inherent within these longer-period averages are the real-time airflow rates, and therefore it is important to know the expected uncertainty in these measurements. The following section summarizes the expected measurement uncertainty in real-time airflow rate using the techniques presented in this paper.

Airflow Rate Measurement Uncertainty

The uncertainty in real-time airflow rate measurements is discussed in this section. First, the actual airflow rate and expected measurement uncertainty is discussed using the Iowa arrangement as an example. This is followed by an uncertainty analysis for the conversion of actual airflow rate to a dry-standard airflow rate. The conversion of actual airflow rate to dry-standard airflow rate was a requirement for this research project because all particulate matter concentrations were available only on standard basis.

Actual Airflow Rate Measurement Uncertainty. The uncertainty in actual airflow rate (ΔQ) will be analyzed first with the Iowa arrangement used as an example. Each Iowa building used seven ventilation stages to control inside temperature at desired conditions. Each of these seven fan stages corresponded to combinations of fans available in each building. In turn, each fan had a unique calibration curve and in total each fan was affected differently by the building's operating static pressure (*SP*).

The actual airflow rate (Q) was estimated for this research project by calibrating all fans against a known standard as described earlier. From the calibration equations for each fan, actual airflow rate was related to the building's operating *SP* measured across the fan. In general, a third-ordered polynomial was used to describe actual airflow rate for each fan as a function of building *SP* as

$$Q = \sum_{i} \{\beta_{1,i} + \beta_{2,i}SP + \beta_{3,i}SP^2 + \beta_{4,i}SP^3\} \tag{1}$$

ΔQ can be evaluated using the following:

$$\Delta Q = \left[\left(\frac{\partial Q}{\partial SP}\,\Delta SP\right)^2 + \Delta Q_{95\%\mathrm{CI}}{}^2\right]^{1/2} \tag{2}$$

The first term in the uncertainty represents the *SP* measurement uncertainty and the second term represents the uncertainty associated with the regression equation used to estimate actual airflow rate estimated with a 95% confidence interval. The subscripts in eq 1 represent the calibration coefficients that result for each fan "i" with the sum dictated by the number and type of fans operating for each ventilation stage. For the purpose of this discussion, eq 1 was reduced to a linear function resulting in

$$Q = \sum_{i} \{\beta_{1,i} + \beta_{2,i}\, SP\} \tag{3}$$

or as

$$Q = \sum_{i} \beta_{1,i} + \sum_{i} \beta_{2,i}\, SP \tag{4}$$

with an offset represented by $\sum\beta_{1,i}$ and a *SP*-affected multiplier of $\sum\beta_{2,i}$. The summations by stage for the Iowa arrangement are given in Table 4. Differentiating eq 4 with respect to *SP* results in

$$\frac{\partial Q}{\partial SP} = \sum_{i} \beta_{2,i} \tag{5}$$

Substituting eq 5 into eq 2 results in the estimation of ΔQ as

$$\Delta Q = \left[\left(\sum_{i} (\beta_{2,i}) \Delta SP \right)^2 + \Delta Q_{95\%CI}{}^2 \right]^{1/2} \tag{6}$$

The uncertainty given in eq 6 is inherently barn- and fan-specific as a result of the specific multiplier $\beta_{2,i}$ and the uncertainty in using a regression equation for each fan ($\Delta Q_{95\%CI}$). For example, the seven-stage Iowa arrangement, consisting of individually calibrated fans varying in size and capacity, had the *SP* multiplier sums ($\sum\beta_{2,i}$) as shown in Table 4. These multipliers were determined by the regression equation as a result of calibrating each fan against a series of building *SP* values. With each fan operating in parallel, at any ventilation stage, the multipliers for each fan at each ventilation stage were additive and in total were represented by the summations given in Table 4 for each stage. As shown in Table 4, the stage-specific multiplier sums progressively increased from -0.01 m^3/sec·Pa for Stage 1 to -0.24 m^3/sec·Pa for Stage 7. The uncertainty in using the regression equation (95% confidence interval [CI]) for each fan in the Iowa arrangement ranged from 1.03 to 4.64% of reading. A conservative single value of $\Delta Q_{95\%CI} = 4.64\%$ of reading was used (Table 4). The approach given in eqs 3–6 for assessing the uncertainty in actual airflow rate combines the *SP* measurement uncertainty and the measurement uncertainty associated with the regression equation used for each fan. An additional uncertainty is associated with the FANS calibration unit, but this uncertainty was not included in this analysis.

Table 4 summarizes *Q* and estimated ΔQ using eqs 3 and 6, respectively, for the seven ventilation stages used in the Iowa arrangement, assuming an operating static pressure of 20 Pa and a measurement uncertainty for static pressure (ΔSP) based on a conservative estimate from manufacturer's specifications of ±5 Pa. As shown in Table 4, the uncertainty ranged from a low of ±5% of reading to a high of ±6.1% of reading.

Dry-Standard Airflow Rate Measurement Uncertainty. All *Q* measurements (m^3/sec) for this research project were converted to a dry-standard airflow rate (Q'', dsm^3/sec) as given below.

$$Q'' = (1 - W)\, \frac{QPT'}{P'T} \tag{7}$$

where Q'' is the dry-standard airflow rate (dsm^3/sec); *Q* is the actual moist airflow rate (m^3/sec); and *W* is the absolute humidity (kgw/kga), given by

$$= 0.62198\, \frac{P_w}{P_{act} - P_w} \tag{8}$$

T' is the standard temperature, 293.15K (20 °C); *T* is the actual dry-bulb temperature (K); *P* is the actual pressure

Table 4. Iowa building uncertainty analysis for actual and dry-standard airflow rate measurements made.[a]

Fan Stage	SP Multiplier, $\sum\beta_{2,i}$, $\partial Q/\partial SP$ (m^3/sec·Pa)	Offset, $\sum\beta_{1,i}$ (m^3/sec)	Uncorrected Airflow Rate, Q (m^3/sec)	Uncorrected Airflow Uncertainty: ΔQ_{SP} (m^3/sec)	$\Delta Q_{95\%CI}$ (m^3/sec)	ΔQ_{total} (m^3/sec)[b]	Reading ±% Error	Corrected Airflow Rate: Q', (dsm^3/sec)	Actual $\Delta Q'$ (dsm^3/sec)	Corrected Airflow Uncertainty Reading ±% Error
1	−0.010	2.44	2.23	0.05	0.10	0.12	5.2	2.20	0.16	7.2
2	−0.026	7.97	7.45	0.13	0.35	0.37	5.0	7.32	0.38	5.2
3	−0.043	10.3	9.45	0.22	0.44	0.49	5.2	9.29	0.49	5.3
4	−0.050	13.0	12.0	0.25	0.56	0.61	5.1	11.8	0.61	5.2
5	−0.106	15.8	13.7	0.53	0.64	0.83	6.1	13.5	0.82	6.1
6	−0.156	23.9	20.8	0.78	0.97	1.24	6.0	20.5	1.23	6.0
7	−0.237	35.2	30.5	1.18	1.42	1.85	6.0	29.9	1.82	6.1

Notes: [a]Measurement uncertainties for absolute humidity ($\Delta W = \pm 6.17 \times 10^{-4}$ kgw/kga), atmospheric pressure ($\Delta P = \pm 0.05$ atm), static pressure ($\Delta SP = \pm 5$ Pa), and temperature ($\Delta T = \pm 0.25$K) assumed in all calculations. Static pressure across fans assumed at 20 Pa. [b]Calculated using eq 6 as $(\Delta Q_{SP}{}^2 + \Delta Q_{95\%CI}{}^2)^{1/2}$.

(atm); P' is standard pressure, 1 atm (101,325 Pa); P_{act} is actual pressure (Pa); P_w is actual vapor pressure (Pa; $\varphi P_{ws} = \varphi e^{f(T)}$); φ is the relative humidity (RH; decimal); and P_{ws} is the saturation vapor pressure (Pa).

$$f(T) = \frac{C_1}{T} + C_2 + C_3 T + C_4 T^2 + C_5 T^3 + C_6 Ln(T) \quad (9)$$

(for all T > 273.15K; $C_1 = -5.800 \times 10^3$, $C_2 = 1.391$, $C_3 = -4.864 \times 10^{-2}$, $C_4 = 4.176 \times 10^{-5}$, $C_5 = -1.445 \times 10^{-8}$, $C_6 = 6.545$).

To evaluate the overall uncertainty in the conversion to a dry-standard airflow rate (eq 7), the uncertainty, $\Delta Q''$, was evaluated as

$$\Delta Q'' = \left\{ \left[\left(\frac{\partial Q''}{\partial W} \right) \Delta W \right]^2 + \left[\left(\frac{\partial Q''}{\partial P} \right) \Delta P \right]^2 + \left[\left(\frac{\partial Q''}{\partial Q} \right) \Delta Q \right]^2 + \left[\left(\frac{\partial Q''}{\partial T} \right) \Delta T \right]^2 \right\}^{1/2} \quad (10)$$

where

$$\frac{\partial Q''}{\partial W} = \frac{-QP(293.15)}{1.0T} \quad (11)$$

$$\frac{\partial Q''}{\partial P} = (1 - W)\,\frac{Q(293.15)}{1.0T} \quad (12)$$

$$\frac{\partial Q''}{\partial Q} = (1 - W)\,\frac{P(293.15)}{1.0T} \quad (13)$$

$$\frac{\partial Q''}{\partial T} = -(1 - W)\,\frac{QP(293.15)}{1.0T^2} \quad (14)$$

The uncertainty in atmospheric pressure (ΔP) and temperature (ΔT) were estimated using manufacturer's specifications. The ΔQ was estimated using the results given from eq 6 (Table 4). The uncertainty in absolute humidity (ΔW) was estimated using the following:

$$\Delta W = \left\{ \left[\left(\frac{\partial W}{\partial T} \right) \Delta T \right]^2 + \left[\left(\frac{\partial W}{\partial \varphi} \right) \Delta \varphi \right]^2 \right\}^{1/2} \quad (15)$$

where, on the basis of eqs 8 and 9, the following can be written:

$$\frac{\partial W}{\partial T} = 0.62198 \times \left\{ \varphi (P_{act} - e^{f(T)}\varphi)^{-1} + e^{f(T)}\varphi^2 (P_{act} - e^{f(T)}\varphi)^{-2} \right\} e^{f(T)} \times \left\{ -\frac{C_1}{T^2} + C_3 + 2C_4 T + 3C_5 T^2 + \frac{C_6}{T} \right\} \quad (16)$$

$$\frac{\partial W}{\partial \varphi} = 0.62198 \{ e^{2f(T)}\varphi (P_{act} - e^{f(T)}\varphi)^{-2} + e^{f(T)} (P_{act} - e^{f(T)}\varphi)^{-1} \} \quad (17)$$

Within eq 15, the uncertainty in RH ($\Delta\varphi$) can be estimated using manufacturer's specifications. To evaluate the uncertainty in converting Q to $\Delta Q''$, eq 10 was evaluated with the uncertainty in W (ΔW) evaluated using eq 15 and ΔQ evaluated using eq 6. The uncertainties in P (ΔP), T (ΔT), φ ($\Delta\varphi$), and SP (ΔSP) were conservatively estimated from manufacturer's specifications as ±0.05 atm, ±0.25K, ±3%, and ±5 Pa, respectively.

As an example, if the Iowa building was at 50% RH, an inside temperature of 25 °C, and at 1 atm pressure, then eqs 15–17 can be solved to obtain the uncertainty in absolute humidity of $\Delta W = \pm 0.00062$ kgw/kga. Using this value and the ΔQ shown in Table 4, the uncertainty in the conversion to dry-standard airflow rate for each of the seven ventilation stages can be determined using eqs 10–14. As shown in Table 4, $\Delta Q''$ is a maximum of ±7.2% of reading for Stage 1 and a minimum of ±5.2% of reading for Stage 2.

Overall Assessment

The ΔQ averaged ±5.5% of reading across all seven fan stages (the Iowa arrangement). This uncertainty, compared with the daily variations in actual airflow rate shown in Figure 1B, were small in comparison where the average daily coefficient of variation (±daily SD × 100/daily average) for the 2 months shown ranged from a low of ±11% to a high of ±60%.

CONCLUSIONS

A multistate research project was conducted in which the emission of hydrogen sulfide, ammonia, carbon dioxide, PM_{10}, TSP, and odor were measured. A critical variable in all emission work is the ability to accurately measure airflow rate through the housing unit. This research project used a wide variety of methods, all with the intention of measuring building airflow rate as accurately as was economically possible. Three general methods were incorporated and they fall under one of three categories, identified as FIMs, FRMs, and AMMs. Each method has advantages and disadvantages and the best method to use for a site depends on several factors. It is recommended that a combination of methods be used at a site. For example, an FIM combined with an FRM would provide one of two methods that yield at the very least an indication of fan status. This recommendation would help in data preservation, because sensing methods will fail at times throughout long-term emission studies. A detailed propagation of error analysis indicated that the uncertainty in actual airflow rate ranged between ±5% and ±6.1% of reading. A conversion to dry-standard airflow resulted in an uncertainty between ±5.2 and ±7.2% of reading at a building operating static pressure, air temperature, RH, and barometric pressure of 20 Pa, 25 °C, 50%, and 97,700 Pa, respectively. This range of uncertainty was very small in comparison to the daily average variations in airflow rate required to control the internal building climate.

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