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Fan Assessment Numeration System (FANS) Design and Calibration Specifications

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Abstract

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Keywords

Controlled environment, Emissions, Instrumentation, Livestock housing, Poultry, Ventilation

Disciplines

Agriculture | Bioresource and Agricultural Engineering

Comments

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FAN ASSESSMENT NUMERATION SYSTEM (FANS) DESIGN AND CALIBRATION SPECIFICATIONS

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ABSTRACT. A device for in-situ fan airflow measurement, known as the Fan Assessment Numeration System (FANS) device, previously developed and constructed at the USDA-ARS Southern Poultry Research Laboratory, was refined at University of Kentucky as part of a project for quantifying building emissions from mechanically ventilated poultry and livestock facilities. The FANS incorporates an array of five propeller anemometers to perform a real-time traverse of the airflow entering fans of up to 137 cm (54 in.) diameter. Details of the updated design, including hardware, software, and calibration methodology are presented. An error analysis of the flow rate, and calibration results from ten FANS units, is provided. Sufficient details of fabrication and calibration are presented so that interested readers can replicate a FANS for their use. Full design details are available at www.bae.uky.edu/IFAFS/FANS.htm.

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Gas and dust emissions from poultry houses vary with season and weather patterns, management practices, feeding practices, housing styles, and other factors. Little scientific-based data exists for poultry house ammonia emissions for modern U.S. poultry facilities, including laying hen houses, broiler chicken growout houses, and turkey production facilities (Bicudo et al., 2002). A project that involves a comprehensive team of engineers and animal scientists was funded to systematically and thoroughly obtain baseline data for ammonia emissions from broiler and layer housing in the U.S. (Gates et al., 2001). The team will assess the effects of manure and litter management practices and dietary manipulation as possible methods for reducing poultry house emissions.

Building emission rates are obtained as the product of two measurements: gas (or other pollutant) concentration difference between discharge air and ambient air, and the

ventilation rate. Considerable attention has been paid to accurate and robust methods of NH₃ concentration measurements, and a number of different technologies exist (Arogo et al., 2002; Xin et al., 2003). A principal source of uncertainty in measuring building emissions has to do with measurement of the building ventilation rate. Estimating ventilation rate of a whole building is difficult even for mechanically ventilated facilities because of the effects of time, harsh environment, incomplete or irregular maintenance, dynamic and irregular wind effects, equipment switching during measurement, and other factors such as construction methods. Standards, engineering practices, and/or procedures for determination of fan performance (AMCA, 1999; ASHRAE, 2001) and standards for laboratory airflow measurement (ASHRAE Standards, 1992) exist, but whole-building ventilation determination (with multiple inlets and outlets) is more problematic. In part, the difficulty is due to the lack of a reference method to which alternate measurements techniques can be compared and employed.

The purpose of this article is to document the design and performance testing of a device for in-situ airflow determination that was initially developed at the USDA Poultry Research Laboratory (Simmons and Hannigan, 2000; Simmons et al., 1998a, 1998b). The device, called a Fan Assessment Numeration System (FANS), can be used with in-situ fans in poultry and livestock buildings. Each ventilation fan can be calibrated individually with its exact equipment options such as shutters, louvers, and discharge cones. Once calibrated against building static pressure, real-time dynamic measurements of building ventilation can be obtained from readings of fan activity and static pressure. The FANS can serve as a field-based reference measurement technique so that other methods of estimating mechanically ventilated building ventilation rates can be objectively evaluated (e.g., CO₂ balance from livestock heat production relations, tracer methods, direct use of fan curves, etc.).

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FANS DESIGN FEATURES AND DETAILS

The FANS incorporates a horizontal array of five propeller anemometers to perform a real-time traverse of airflow entering ventilation fans up to 1370 mm (54 in.) diameter. Design features, including CAD drawings and a bill of materials, are online and may be downloaded from the Biosystems and Agricultural Engineering server at the University of Kentucky (www.bae.uky.edu/IFAFS/FANS.htm). Included are detailed design drawings of the frame, sheet metal layout, and anemometer rack and associated components in AutoCAD2000 and pdf formats. Details of fabrication are provided in this article.

FABRICATION

The purpose of the following discussion on fabrication of the FANS is to give a general concept of the processes, equipment, materials, and parts necessary for someone to construct their own FANS. The reader is referred to the project web site for detailed schematics, drawings, and a list of materials. The FANS consists of three main components: body and frame, drive system, and anemometer/controls. To reduce weight and the effects of corrosion, the FANS is constructed almost entirely of lightweight aluminum and corrosion-resistant stainless or zinc-plated steel fasteners. The FANS is depicted in the 3D CAD drawings in figure 1.

The frame is constructed from 25.4 mm (1 in.) square aluminum tubing with a 1.59 mm (1/16 in.) wall thickness. The entire frame system, consisting of independently fabricated top, bottom, and side modules, is welded. Vertical traverse of the anemometer rack is accomplished with linear bearings mounted on each side and driven by screws that are connected with chain and driven by a gear motor. Both top and bottom frame sections hold the drive assembly. The bottom frame section also has tubing for mounting the control box and motor mount. The top frame section has an additional tube and holes drilled for mounting the chain tensioner mount. The side frame sections are mirror images of one another, and have one vertical member with holes drilled every 50 mm (1.97 in.) for fastening the linear bearing, and a 4.8 mm (3/16 in) aluminum plate drilled and tapped for attaching carry handles. These handles are a critical ergonomic improvement over the original design.

A smooth airflow entrance is created from a four-member section that is inset 76 mm (3 in.) from the front of the frame. This section provides a rigid support for the aluminum sheeting to wrap over and to aid in maintaining a smooth, low dynamic loss shape. Aluminum sheet metal 0.4039 mm (0.0159 in., 26 ga) covers the top, bottom, and two mirror-imaged sides. All sheet metal sections are sheared to size, and finish fit using hand shears. To make the corners of the transition inlet, material is removed along two lines. During assembly, the side sections must be attached to the frame prior to the top and bottom sections. Either stainless steel rivets or sheet metal screws are used to fasten the sections to the frame using 102 mm (4 in.) spacing between fasteners.

The drive system consists of a commercially available linear actuator with attached motor and gearbox. The linear actuator is removed by cutting the drive shaft approximately 25.4 mm (1 in.) from its protrusion from the gearbox, and discarded. Flats are filed on the remaining shaft for setscrews. The motor's internal limit switch and associated small set of

nylon gears are removed. The motor's output shaft is joined to the vertically mounted, precision drive screw via a flexible coupling. This drive screw turns a sprocket and chain at the inside of the top frame section; the chain turns the opposite drive screw to assure simultaneous vertical movement of both linear bearings. The sprockets, chain, and chain tensioner are mounted to the top frame section of the FANS. The linear bearings are parallel to both drive screws, and the linkage between the drive screws, bearings, and anemometer rack is accomplished at bearing plates on each side. These plates rigidly fasten to the anemometer bar, allowing only in-plane movement of the anemometer rack. The plate on one side also contacts top and bottom limit switches (DPDT) to stop rack travel. The anemometer rack is constructed of 38.1 mm (1.5 in.) aluminum tubing, with five holes into which are welded the anemometer's threaded adaptors. A slot on the front of the rack is also milled to provide for the data cables connecting anemometers to the control box. Wire is routed from the rack, held away from the threaded screw with a simple hook, and then routed behind a separate shield made of thicker aluminum sheet (e.g., 1.2903 mm; 0.0508 in. or 16 ga) and down to the data acquisition box.

ELECTRONICS AND SOFTWARE

A schematic of control functions is provided in figure 2. Data acquisition is accomplished using a program called Anemometer2, written in VisualBasic. Solid-state relays (SSR) isolate the PC from AC power to the gear motor. TTL-compatible digital inputs are connected to one set of upper and lower limit switch contacts to determine whether either is active. When acquiring data from the anemometers, a loop is executed continuously as the rack travels from one limit switch to the other. For each of the five analog input channels in turn, 1000 samples are acquired at a rate of 10 kHz and averaged to obtain a single velocity reading per channel. The rack requires approximately 185 seconds to travel the full length of opening, thus about 1775 averaged velocity readings (1000 points/average), distributed uniformly over the opening, are acquired during a single traverse.

FANS CALIBRATION RESULTS

COMPONENT ERROR ANALYSIS

A component error analysis was performed by students in the BAE 599 PC-Based Data Acquisition and Control course (fall, 2001), per the sponsor's requirement for an educational component to the project. A breakdown of errors, expressed as mV and least significant bits (LSBs), is provided below:

Data Acquisition Card

For this project, a 12-bit analog-to-digital converter (ADC, model KPCMCIA-12AI, Keithley Instruments, Inc., Cleveland Ohio) was set to a bipolar ± 1.25 V input range. The resolution (LSB) is thus 0.61 mV bit⁻¹. Integral and differential linearity errors are each given as 1 LSB, and full-scale error is 0.5% (12.5 mV). The combined, maximum probable errors associated with ADC are thus:

$$\begin{aligned}\Delta_{\text{ADC}} &= \sqrt{0.61^2 + 0.61^2 + 12.5^2} \text{ mV} \\ &= 12.5 \text{ mV or } 20.5 \text{ LSB}\end{aligned}\quad (1)$$

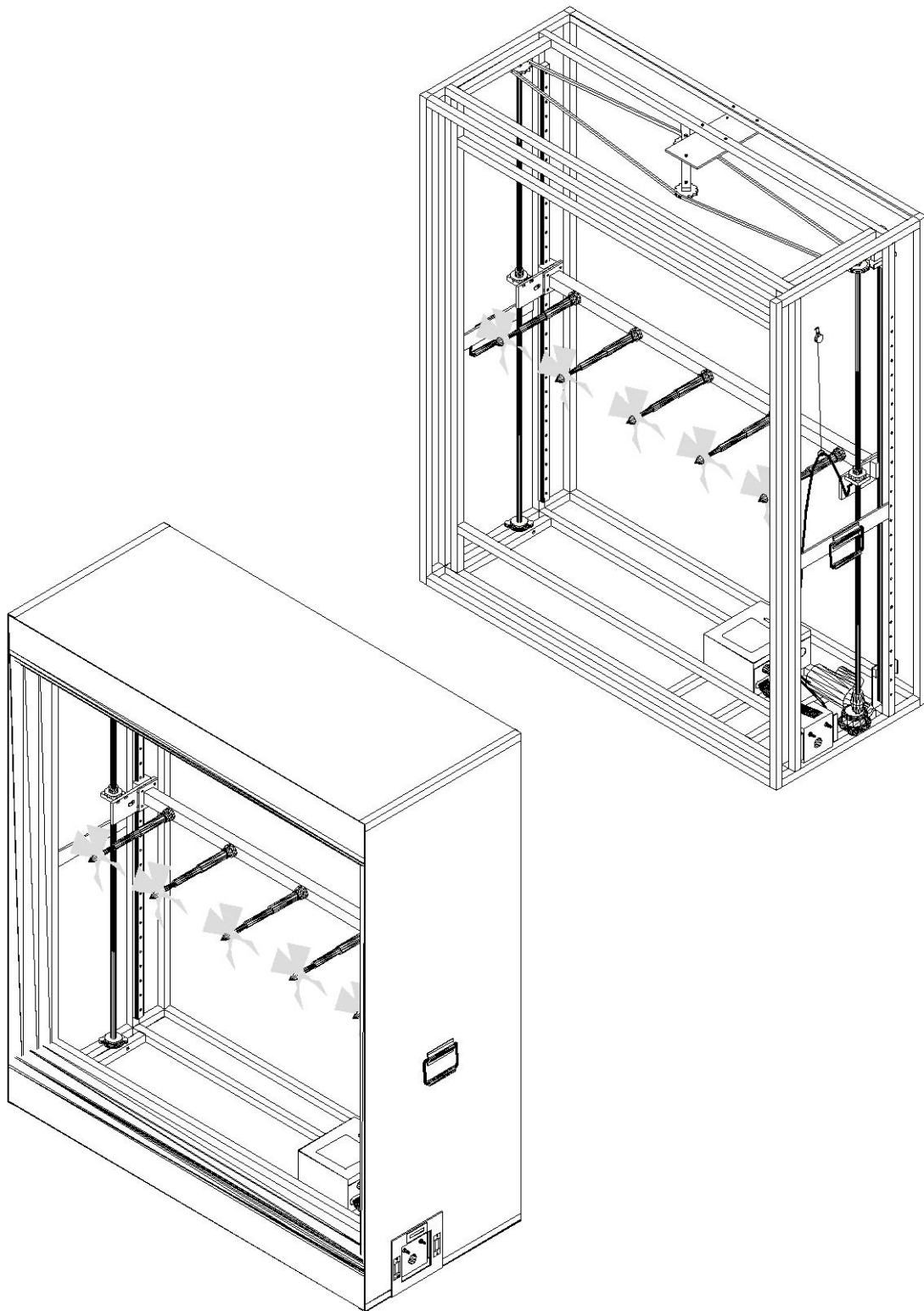


Figure 1. FANS 3D CAD drawings: (upper) FANS without external sheet metal showing frame and operating mechanisms, and (lower) fully assembled FANS.

Oversampling, i.e., the technique of using the mean of multiple AD conversions to estimate the “true” voltage, will reduce full-scale error. In this application, we utilize 1000 samples per observation. The resultant ΔADC is 6.3 to 1.5 mV (10.3 to 2.4 LSB), respectively, for a 5 to 10 fold reduction in full-scale error. Alternatively, an ADC with smaller uncertainties could be selected.

Anemometer DC Generator

The manufacturer provides calibration equations for velocity (U) as a function of either rpm or generated voltage:

$$U = 0.018 \times \text{mV} = 0.005 \times \text{rpm} \quad (2)$$

The anemometer generates a DC voltage proportional to propeller rotational velocity with accuracy within 1% of

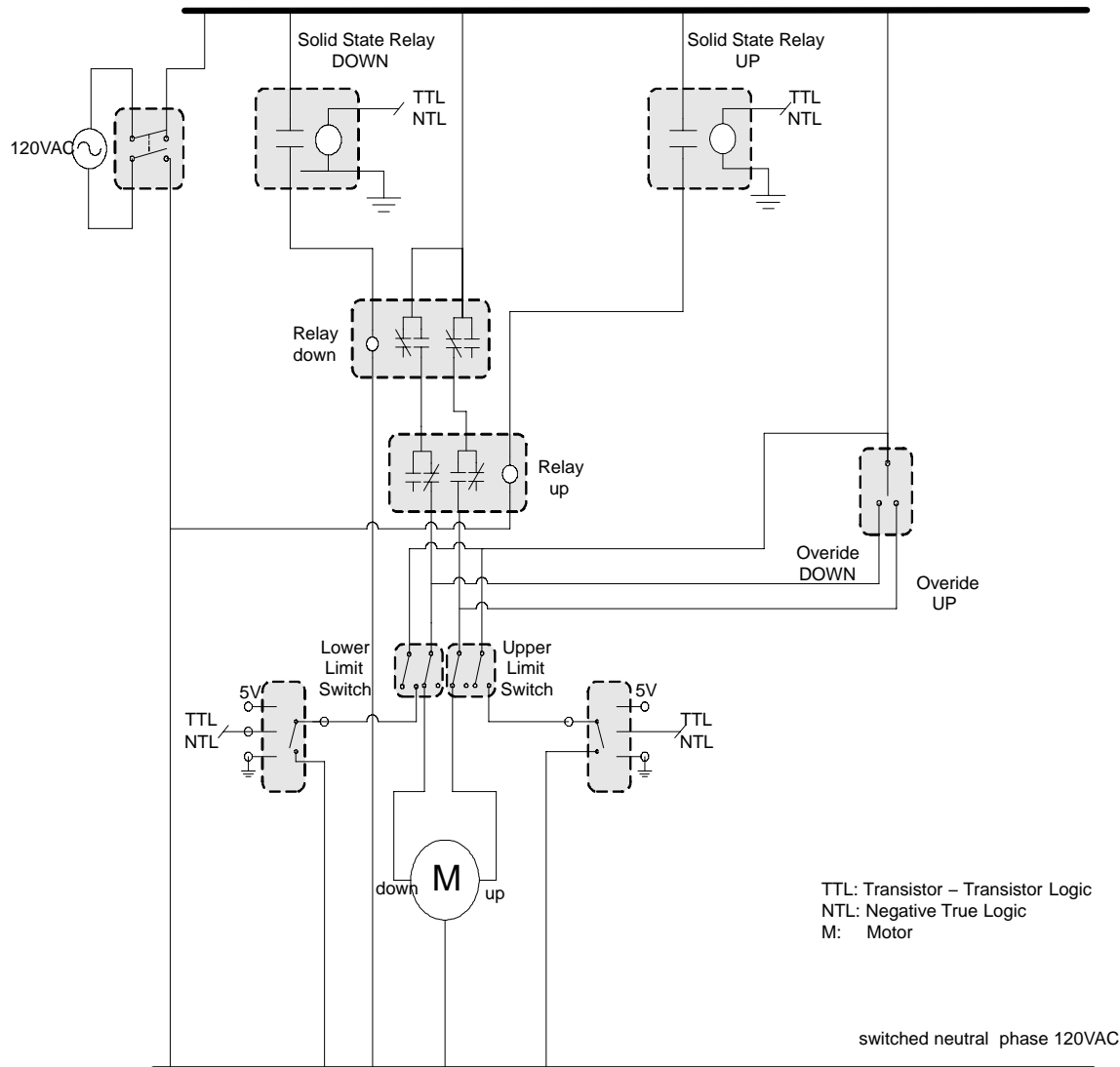


Figure 2. Schematic of data acquisition and controls circuits.

reading. The DC generator is calibrated at 500 ± 2 mV at 1800 rpm (0.3 mV/rpm). Zero offset is negligible.

Anemometer accuracy is expressed by the manufacturer both as ± 2 mV and relative to readings. At a maximum expected velocity of 8 m s^{-1} (1575 fpm), the nominal rotational speed is 1600 ± 1 rpm (or 480 ± 4.8 mV), and the DC generator calibration error is < 2 mV (7.3 LSB). Combined, the maximum total anemometer probable error is:

$$\begin{aligned} \Delta A_{\text{anemometer}} &= \sqrt{4.8^2 + 2^2} \text{ mV} \\ &= 5.2 \text{ mV or } 8.5 \text{ LSB} \end{aligned} \quad (3)$$

and the minimum error is simply 2 mV. Expressed in units of velocity, these are 0.09 and 0.02 m s^{-1} , respectively.

Velocity Error

The maximum probable error of a velocity reading is the combination of maximum ADC and maximum anemometer errors. At 8 m s^{-1} , this is:

$$\begin{aligned} \Delta U &= \sqrt{12.5^2 + 5.2^2} \text{ mV} \\ &= 13.5 \text{ mV or } 22.2 \text{ LSB } (0.24 \text{ ms}^{-1}) \end{aligned} \quad (4)$$

Total Error

The expected error of the flow measurement system can be estimated from the component errors above, using the following relation between airflow rate (Q) and measured velocity (U) and inlet cross-sectional area (A).

$$Q (\text{m}^3 \text{ h}^{-1}) = (U \times 3600) \times A \quad (5)$$

Velocity is obtained from the anemometer via ADC. The nominal inlet area of the FANS is 1.664 m^2 ($d = 1.290 \text{ m} \pm 1.6 \text{ mm}$ square) with error on the order of $2d \cdot \Delta d = 0.004 \text{ m}^2$.

An estimate of maximum probable uncertainty in flow measurement can be obtained from the maximum expected velocity through the FANS (i.e., 8 m s^{-1}). This is equivalent to a nominal airflow rate (Q) through the FANS of $47,923 \text{ m}^3 \text{ h}^{-1}$ ($28,207 \text{ cfm}$). Maximum probable uncertainty at this rate is obtained from a component error analysis:

$$\Delta Q = \sqrt{\left(\frac{\partial Q}{\partial U} \Delta U\right)^2 + \left(\frac{\partial Q}{\partial A} \Delta A\right)^2} \quad (6)$$

or:

$$\begin{aligned} \Delta Q &= \sqrt{(A \cdot \Delta U)^2 + (U \cdot \Delta A)^2} \\ &= \sqrt{(1.290 \times (0.24 \times 3600))^2 + ((8 \times 3600) \times 0.004)^2} \\ &= 1120 \text{ m}^3 \text{ h}^{-1} \text{ (660 cfm)} \end{aligned} \quad (7)$$

This probable error is about 3% of the flow reading, assuming the worst-case full-scale ADC error of 0.5%. Of course, ΔQ decreases with velocity (eqs. 3 to 5).

Figure 3 demonstrates probable error as a function of flow, as affected by oversampling. The effect of flow rate is relatively small. The full-scale ADC error is shown to be critical, and warrants careful selection of ADC. It should be pointed out that increasing the number of bits of the ADC has negligible improvement on the RMS error of airflow rate. However, as is shown in the following section, calibration can reduce the uncertainty further.

LABORATORY CALIBRATION OF FANS

Ten newly constructed FANS were individually calibrated at the University of Illinois BESS fan test facility (www.age.uiuc.edu/bee/facility/bess/bess.htm). Figure 4 is a graph of measured vs. “true” airflow calibration curves for these ten units.

Two slightly different means of expressing the calibration equations are possible: regression of measured (y) versus reference airflow rate (x) as obtained (i.e., of the form $y = a + bx$), or inclusion of a zero flow reading and then subtracting this offset (y_0) from each measured reading and regressing the result (i.e., of the form $y - y_0 = bx$). Expressed in these two ways, the calibration equation for the ten FANS together was determined as follows (units are $\text{m}^3 \text{ h}^{-1}$, and numbers in parentheses are standard errors of regression coefficients):

$$Q_{\text{FANS}} = 1.015 (\pm 0.0009) \times Q_{\text{Actual}} - 322 (\pm 29) \quad (8)$$

$$Q_{\text{FANS}} = y_{0n} + 1.011 (\pm 0.0011) \times Q_{\text{Actual}} \quad (9)$$

where y_{0n} depends on each device (10-unit average = $-158 \text{ m}^3 \text{ h}^{-1}$).

Airflow rate from a given FANS is obtained by inversion of the calibration equation:

$$\begin{aligned} Q_{\text{Actual}} &= \frac{(Q_{\text{FANS}} + 322)}{1.015} \\ &= 317 + 0.985 \times Q_{\text{FANS}} \end{aligned} \quad (10)$$

$$\begin{aligned} Q_{\text{Actual}} &= \frac{(Q_{\text{FANS}} - y_{0n})}{1.011} \\ &= 0.985 \times (Q_{\text{FANS}} - y_{0n}) \end{aligned} \quad (11)$$

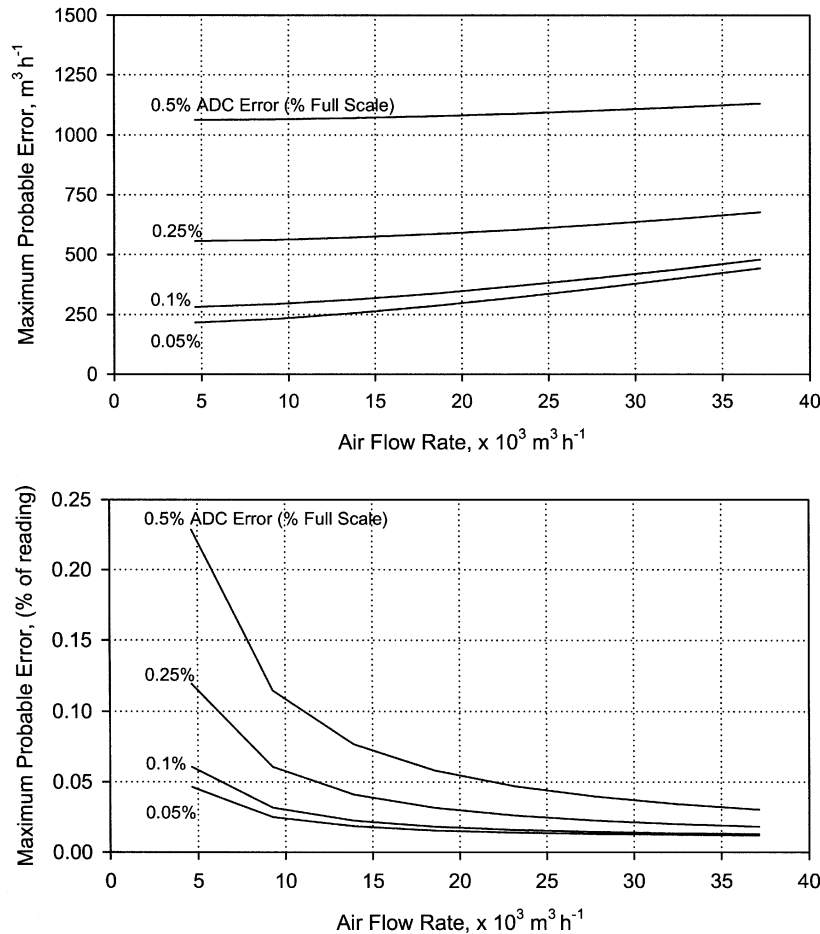


Figure 3. Graphs illustrating the error in the FANS reading as a function of airflow rates of pressure drop: (upper) $\text{m}^3 \text{ h}^{-1}$, and (lower) % of reading.

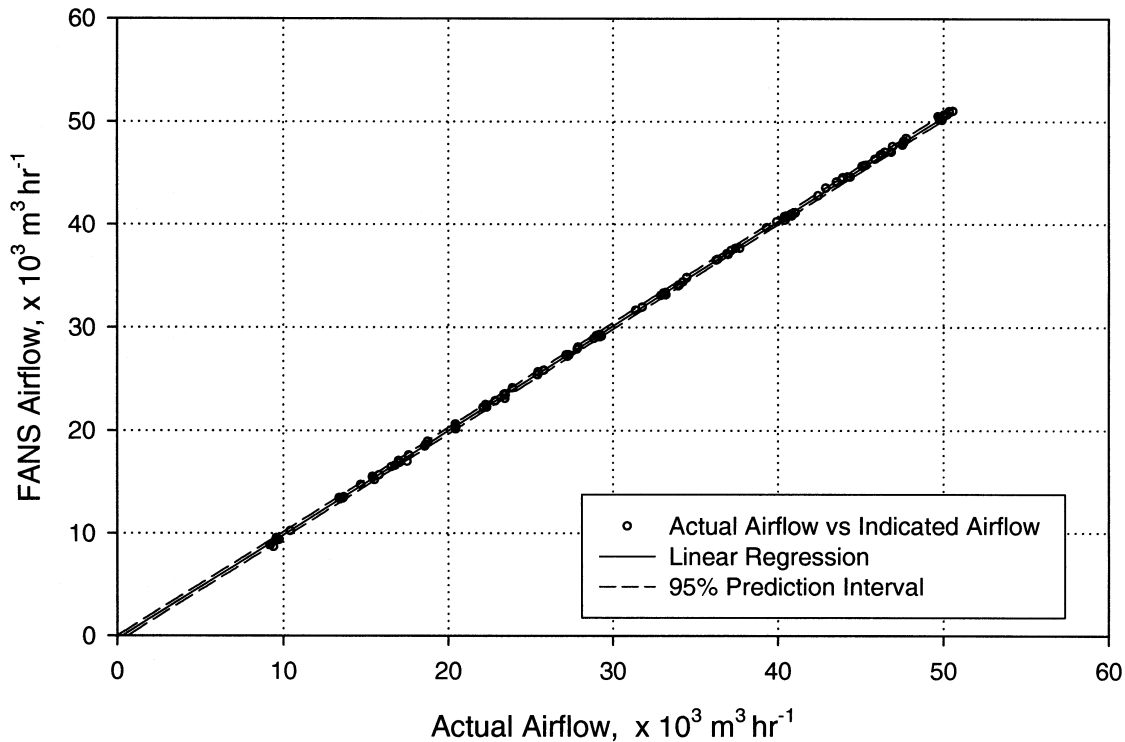


Figure 4. Composite graph illustrating the uniformity of measurement between ten different FANS. Reference flow obtained from standard flow nozzle equations for the nozzles in the University of Illinois BESS lab from manometer readings of pressure drop.

Regression slopes obtained from calibration of individual FANS were remarkably similar; it is thus recommended that a given unit can be used with either of the two equations above. The second relation, i.e., subtraction of any zero-flow offset, has the convenience of occasionally determining whether drift in zero offset has occurred by a simple check with no airflow during use.

The standard error of regression provides a simple estimate of measurement precision for comparison to the theoretical value obtained from a component error analysis in the previous section. For the second regression equation (eq. 11), the standard error (SE) = $141 \text{ m}^3 \text{ h}^{-1}$ (83 cfm), and the estimated imprecision in a measurement is thus $SE/b = 141/1.011 = 139 \text{ m}^3 \text{ h}^{-1}$ (82 cfm). The range in SE/b for the ten FANS was 71 to $232 \text{ m}^3 \text{ h}^{-1}$ (42 to 137 cfm). In terms of 910 or 1220 mm (36 or 48 in.) diameter ventilation fans, nominally $17,000$ or $34,000 \text{ m}^3 \text{ h}^{-1}$ (10,000 or 20,000 cfm), the mean imprecision is thus 0.8% and 0.4% of reading, respectively. The error from simply neglecting the calibration equation amounts to 101 and $360 \text{ m}^3 \text{ h}^{-1}$ (59 and 212 cfm), or 0.6% and 1.1% of reading, respectively, for these two fan sizes.

COMPARISON BETWEEN COMPONENT ERROR ANALYSIS AND CALIBRATION

From the preceding two sections, actual performance of the FANS device is shown to be better than the probable performance as predicted from the component error analysis. Thus, when characterizing the FANS performance, the recommended method is to use representative statistical values from the calibration.

FANS USER EXPERIENCES

A survey of users after approximately two years of field use was performed. Comments, observations, and suggestions are summarized in this section.

Use of the FANS upstream of a ventilation fan adds some pressure drop for the fan to work against and hence may reduce fan airflow rate. The penalty depends on the dynamic losses of the fan and the FANS in tandem, just as the addition of a shutter will differently affect the performance of different fan models. Field experience of testing seven different fan models ranging in size from 450 mm to 1270 mm suggests that there is no penalty for fans with a free air capacity of less than $30,000 \text{ m}^3 \text{ h}^{-1}$ (17,650 cfm). The penalty on larger, higher performing fans increases with size and performance. Further research is required to quantify this penalty.

The connection between the computer PCMCIA card and the data cable is not very robust and is easily damaged in the field. Care must be taken to ensure that the cable connection is not subjected to strain or pulled at an angle. Use of a PCMCIA strain relief bracket (Part No. 777550-01, National Instruments) may be of assistance in reducing strain on this connection.

Maneuvering and placing the FANS within a chicken house can be cumbersome due to its size and weight, the varying heights of fans and walls, and the presence of other obstructions including electrical conduits, knee braces, and water and feed lines. The handles on the sides of the FANS are an important aid in its positioning; however, these have been located to suit persons of average height and may therefore not be universally comfortable. Positioning the FANS at the correct height at each fan is made easier with the use of a modified, hydraulic lift table, as first constructed by researchers at Pennsylvania State University.

A run should be carried out at the start of each day under no-airflow conditions to check FANS operation and check that the FANS offset has not varied substantially.

The linear bearing slides and acme threads should be lightly coated with a silicone spray lubricant (e.g., Lubrimatec heavy-duty silicone lubricant) at the start of each day. The lubricant is sprayed onto the moving drive screws and chain while an absorbent paper towel is held behind to catch the overspray. This saturated paper towel is then wiped along the linear bearing slides. A fresh paper towel is then used to wipe down the bearing surfaces. Excess lubricant should not be applied as it will encourage the accumulation of dust and other airborne particulates on the lubricated surfaces. The FANS should be thoroughly cleaned following each period of field use with particular attention given to removing accumulated dirt and foreign material on the bearing and driving surfaces.

It has been reported that the anemometer arm can become jammed during operation, resulting in damage to the drive motor. Where an appropriate program of lubrication and maintenance has been used, this has not been experienced. Care also needs to be exercised that the anemometer bar does not become distorted through inappropriate tie-down during transport.

At each fan, it is normal to make about six runs with static pressures increasing from free air up to 35 to 40 Pa. With a team of three people, it is possible to generate a fan performance curve in approximately 30 minutes. It can take 30 to 60 minutes to relocate the FANS to the next fan and prepare it for use, depending on travel distance and difficulty in positioning and sealing the FANS. In field practice, it has been possible to complete the characterization of 11 to 14 fans in a typical broiler house in one day.

SUMMARY

Details of the design, fabrication, and performance of a device to measure airflow rate through propeller fans in-situ were presented. Ten of the FANS units have been fabricated and calibrated. The units predicted airflow rate within 1%, and after calibration had an imprecision of 71 to 232 m³ h⁻¹ (42 to 137 cfm) over the ten units. Drawings and CAD files are available at www.bae.uky.edu/ifafs/fans.htm.

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