In-Field Quantification of Fan Performance in Tunnel-Ventilated Freestall Barns

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ABSTRACT

Heat stress is a major cause of decreased milk production of cattle during the hot summer months. It has been shown that tunnel ventilating dairy barns is an economical way to effectively cool livestock thereby increasing overall animal comfort and productivity. This study served to quantify the overall in-place fan capacity and determine the best placement and layout for maximum performance. From the analysis of three (3) New York State tunnel-ventilated dairy facilities, it was determined that air was not uniformly discharged from each fan, therefore a contributing area method that weighted each air velocity according to position on the fan face was the best measuring technique. Fan capacity for barn No. 1 was 24,581 cfm, for barn No. 2 was 37,257 cfm, and for barn No. 3 was 27,466 cfm. Efficiencies for barn Nos. 1 and 3 were 82.2% and 91.8% respectively. Based individual fan data, it was determined that tunnel-ventilated barns should have ceilings to decrease the cross-sectional area and eliminate the needless exchange of air far from the animals, fans should not have any objects blocking the airflow path, and placing the fans on an endwall angled at 60° funnels the air out of the barn for an overall increase in fan performance compared to a flat endwall.

INTRODUCTION

Tunnel ventilation of dairy barns is a cost-effective way to ventilate a barn and reduce the level of heat stress on cattle during summer months. Heat stress results in decreased feed intake of dairy cattle, which consequently decreases milk production and reduces overall health. Improving the cow environment by use of tunnel ventilation will increase the productivity of the herd on many farms.

Tunnel-ventilated dairy barns are usually designed based on achieving a desired air exchange rate (1,000 cfm/cow) and air velocity throughout the barn. Research has shown that 500 to 600 fpm of air velocity is needed to effectively cool cows during summer heat stress conditions (Shearer, 1991). The cross-sectional area of the barn perpendicular to the direction of airflow, number of fans, number of cows, and fan output all determine the airflow.

Individual fan capacity is generally based on the theoretical airflow rate provided by the manufacturer, or preferably by the fan analysis data obtained by a third party testing lab such as BESS Lab. Theoretical fan capacities reported in this paper were obtained at BESS Lab and provided by the manufacturer. During laboratory fan testing, each fan is analyzed at a specific static pressure by itself, in a wind tunnel; it has no competition from other fans, or any obstructions blocking airflow. Although this is a good estimate of in-place fan capacity, fan performance is expected to differ from the theoretical value. Static pressure differences, ambient wind speed, fan placement, competition with neighboring fans for air, fan cleanliness, and/or objects blocking the airflow of the fan all can preclude achieving optimum performance of each fan.

This study evaluated the in-place capacity of each individual tunnel ventilation fan in the presence of the various competing environmental factors and also evaluated the overall system performance of three (3) production dairy barns in New York State. This project is aimed at finding an easy, accurate, low cost method for quantifying the in-place capacity of fans used in tunnel-ventilated dairy barns and to determine how fans compared relative to their performance based on BESS Lab data.

OBJECTIVES

The objectives of this study were to:

1.) determine the best method for quantifying in-place fan capacity,
2.) use the chosen method to find the overall in-place fan capacity,
3.) determine if individual fan placement and overall layout affect fan performance, and
4.) determine if findings can be useful to future efforts to quantify ammonia emissions from tunnel-ventilated dairy barns.

These objectives will not only allow for assessment of in-place fan performance, but also may prove useful for future work where actual fan performance data is needed. For example, quantification of airborne emissions, such as ammonia, leaving a tunnel-ventilated barn can be calculated by determining the emission concentration and the in-place fan capacity. The US EPA is concerned with the amount of ammonia being discharged from animal housing facilities (Parry, 2003). A method to accurately, yet cost effectively, quantify ammonia emissions from dairy barns would be ideal.

LITERATURE REVIEW

Sufficient and uniform air exchange (ventilation) is the main concern when evaluating the effectiveness of a dairy barn ventilation system. It is important to have accurate quantification of the airflow throughout the barn. Wheeler and Bottcher (1995) found that taking measurements on the discharge side of the ventilation fan using a vane anemometer provided the most accurate results. Their research showed that taking several measurements across the fan area would account for the uneven airflow pattern and thereby accurately quantify the fan's airflow capacity.

Simmons et al. (1998) investigated the effects of fan position with respect to airflow direction and proximity to neighboring fans. They observed that placing fans at a 90° angle with respect to the primary direction of airflow decreases the effectiveness of the fan because air is required to abruptly change direction before being discharged by the fan. Air resists the change in direction because it naturally wants to remain in the direction it is traveling. Fans mounted on the sidewall with the best performance still only ran at 83% of its maximum capacity. Simmons et al. also looked at fan placement on both the endwall and sidewalls and found that placing sidewall fans no closer than 1 foot from endwall fans produced negligible flow discrepancies.

BACKGROUND

Three (3) New York State commercial dairy farms each with freestall barns that utilized tunnel ventilation were examined. The relevant characteristics of each barn are shown in Table 1.

Table 1. Barn characteristics and tunnel ventilation fan information.

Barn No.	1	2	3
Dimensions, w x l (ft)	106.5 x 400	106 x 770	62 x 352
X-sectional Area[1] (ft^2)	2,223[2]	1,908	930
No. Rows of Stall	6	6	3
Insulated Ceiling	No	Yes	Yes
Fan Type and Size	51" Vortex	60" Advantage	51" Vortex
Motor Size, Hp	1.5	2	1.5
Theor. Airflow (cfm)[3] per fan (@ 0.05" S.P. w.g.)	29,900	-- [4]	29,900
Theor. Blade Speed (RPM)	575	490	575
No. of Fans	22	20	20
Cum. Theor. Airflow (cfm)	657,800	--	598,000
Theor. Avg. Vel. (fpm)	295.9	--	643.0

[1]Perpendicular to the direction of airflow.
[2]Calculated based on area between barn floor and roof, vertical airflow baffles are ignored.
[3]Data developed by BESS Lab and listed in manufacturer's literature.
[4] Data not available.

Barn Nos. 1 and 2 were located in central New York while barn No. 3 was located in northern New York. Fans in barn Nos. 1 and 3 had discharge cones, grates, and louvers. Fans in barn No. 2 had the discharge cones and grates, but no louvers. Barn Nos. 1 and 2 had fans oriented normal to the direction of airflow while barn No. 3 had an endwall at an obtuse angle with respect to the sidewall.

The endwall elevation for barn Nos. 1 and 2 are shown in Figures 1 and 2 respectively. Each fan is identified with a number, depicting the fan position from the left side of the barn, and letter, depicting fan level (B = bottom; M = middle; T = top). The same convention was used to identify barn No. 3 fans, as shown in Figure 3.

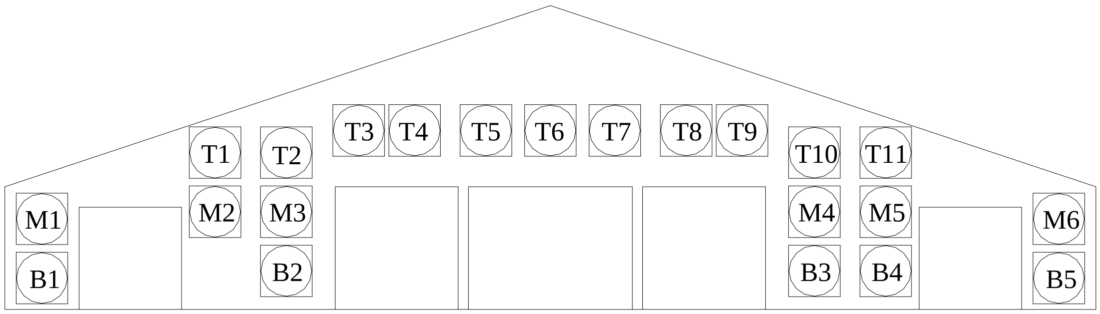

Figure 1. Endwall elevation of barn No. 1.

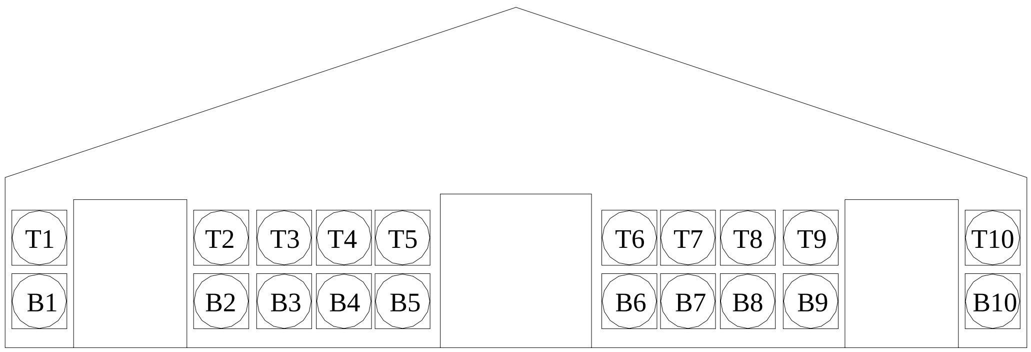

Figure 2. Endwall elevation of barn No. 2.

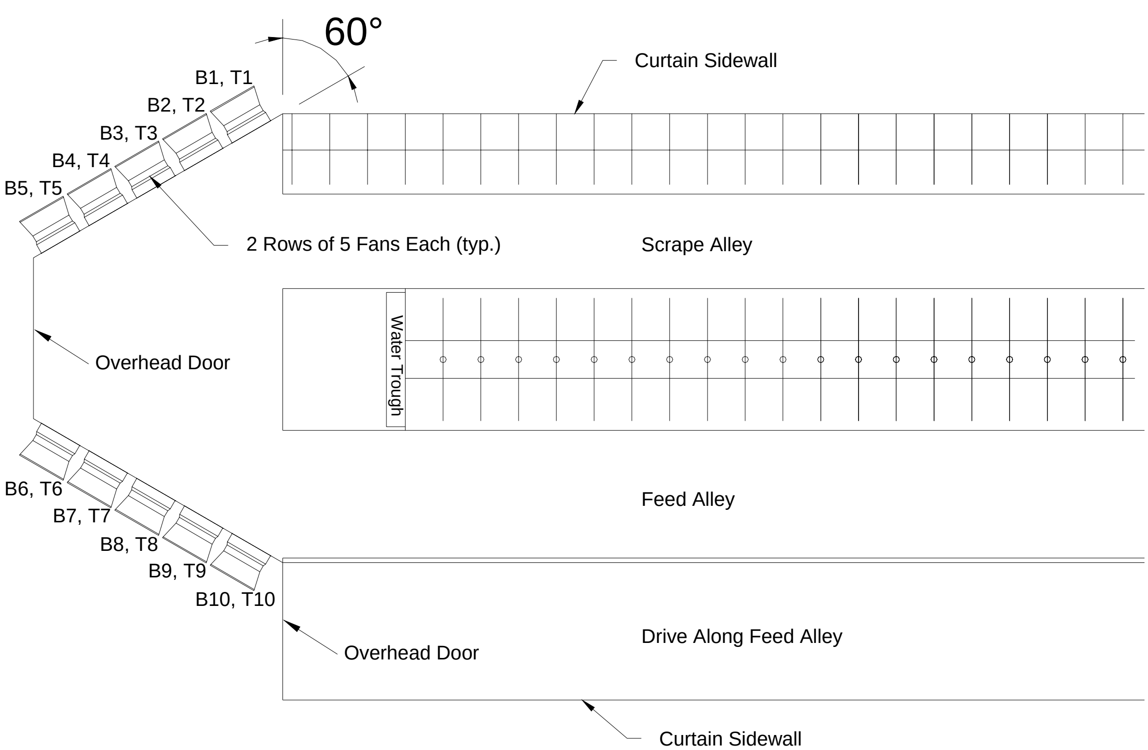

Figure 3. Plan view of barn No. 3.

PROCEDURE

In-place fan capacity analysis included static pressure measurements for each barn, and fan blade rotation and air discharge measurements for each fan. A digital manometer (Dwyer, model #475) was used to measure static pressure. Static pressure measurements were taken at approximately halfway down the length of each barn. Fan blade rotational speed was measured using a non-contact tachometer (Monarch, model #3T179). Air velocity was measured using a recording vane anemometer (Extech, model #451126). For safety reasons, all air velocity readings were taken on the discharge side of the fan.

<u>Method Comparison</u>

To quantify in-place fan capacity, two methods, the average velocity method and contributing area method, were tried and compared as a pilot study to evaluate which one should be used based on field sampling time and accuracy. The procedure for the average velocity method is to make several evenly spaced air discharge speed measurements for a single fan, average them, and multiply this value by the discharge cross-sectional area to obtain an overall volumetric flow rate through the fan. With the contributing area method, air velocity measurements are taken at specific predetermined positions on the fan face. This value is then multiplied by its contributing area to give a flow rate for that particular section of the fan. The sum of all of the individual sections across the fan provide a total volumetric flow rate for that fan.

The average velocity method assumes that each air velocity measured plays an equal role in the overall performance of the fan. The contributing area method takes into consideration the fact that each area of the fan discharges a different amount of air by weighting the air velocity with the appropriate contributing area according to its position on the discharge face.

After analysis of the pilot study data (data not shown), air velocity was not found to be consistent across the discharge face of each fan and each area measured does not contribute equally to the overall fan discharge (Figure 4); therefore the contributing area method was determined to be the better method for this study.

<u>Preliminary Measurements</u>

The radial distance that the vane anemometer was placed from the fan's center as well as the size of the contributing area was calculated from the preliminary measurements for each contributing area. A group of four to five fans for each barn were chosen for these preliminary measurements according to location in order to provide an accurate sampling of all fans. Factors taken into account included distance from barn openings, as well as proximity to neighboring fans. At least one fan from each vertical level was chosen. Measurements were made across four axes: the horizontal axis, vertical axis, and diagonally at a 45-degree angle across the fan face starting from the upper left and again starting from the upper right. Measurements were taken for 10-second intervals every two inches along each of the four axes described (there were approximately 25-35 intervals for each axis). The anemometer recorded data points every second. The

10 data points for each two-inch position were then averaged together. These data points were then plotted using Excel and a graphical analysis was made.

The graphs obtained from the preliminary analysis data resembled a sinusoidal curve as shown in Figure 4b. The curve was sectioned off as shown and an average value for each section was then calculated from the corresponding two-inch position points. The curve sections shown in Figure 4b determine the different circular area boundaries shown in Figure 4a. The average value for each section was then matched to a position on the sinusoidal curve, and the corresponding distance from the center was obtained. This method was used for each of the remaining sections of the curve for each of the fans under preliminary analysis. After 9 (or 13 for barn No. 2) values were obtained for each of the fans tested, an average value was taken for each position across each fan. These new 9 values (or 13 for barn No. 2) and corresponding areas were used for the actual fan analysis.

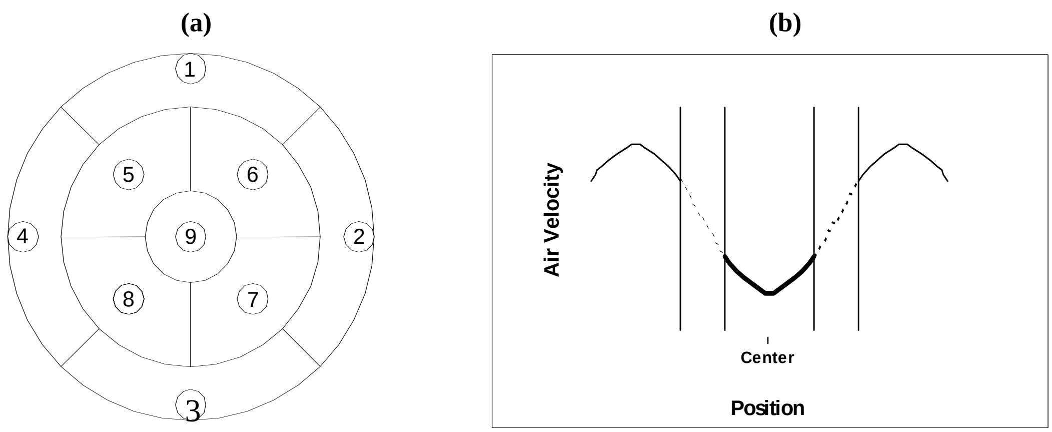

Figure 4. The central, inner, and outer areas (a) correspond to the air velocity vs. position graph (b). The position is the placement across the diameter of the fan. The outermost peaks (thin solid line) represent regions 1-4. The dotted lines represent regions 5-8. The central trough (thick solid line) represents region 9.

The preliminary data for barn Nos. 1, 2, and 3 are shown in Table 2.

Table 2. Preliminary data measurements.

Fans in Barn Number	Region	Position from Edge (in)	Contributing Area (ft^2)
1	Inner	11.25	2.991
	Outer	26.25	1.841
	Center	30	0.3068
2	Inner	12	3.338
	Outer	24	0.9163
	Center	33	0.4418
3	Inner	13.75	4.057
	Outer	22.5	0.8181
	Center	25	0.1364

The data collection positions found from the preliminary analysis vary because they are affected not only by fan type and size, but also by fan location and position on the endwall. The volume of air that enters each fan before being discharged varies across the inlet of the fan. This variation can be attributed to obstructions placed at the fan inlet, or from an air deficit due to the placement of the fan on the endwall. Therefore, the optimum position for data collection for each fan will differ from barn to barn, as well as from fan to fan. The same fans placed in different barns will experience varying airflows, and will consequently yield differing preliminary profiles. However, once preliminary positions are determined for a particular barn, they should not vary from day to day. Only the magnitude of the values should change.

For the actual fan analysis, air velocity readings were taken for 30 seconds in each of the 9 (or 13 for barn No. 2) contributing areas for each fan. The anemometer recorded a data point every second. Volumetric flow rate for each fan was found and an overall barn flow rate was calculated. Analysis of the overall in place performance of the fan was done through comparison with the manufacturer (theoretical) data.

RESULTS

The following data was collected using the contributing areas method and analyzed. Table 3 shows the months that the testing was carried out. All measurements were taken during the summer months so as to keep the fan environment as consistent as possible and minimize any data variations due to the surroundings.

Table 3. Fan testing schedule by month.

Barn No.	Trial 1	Trial 2	Trial 3
1	July 2002	July 2002	June 2003
2	August 2002	August 2002	June 2003
3	August 2002	July 2003	July 2003

Fan Capacity Data

Barn No. 1
The volumetric output and fan blade rotational speed for each fan in barn No. 1 is shown in Table 4 for three trials. The average volumetric output for all fans in trial No. 1 was 25,099 cfm, 26,133 cfm for trial No. 2, and 22,512 cfm for trial No. 3. The static pressures for trials Nos. 1-3 were 0.04 and 0.06, 0.05 s.p.w.g., respectively. The average rotational speed for trial Nos. 1 and 2 was 560.5 and 560.6 respectively. No fan speed measurements were taken for trial No. 3.

Table 4. In-place fan performance for barn No. 1.

	Trial 1		Trial 2		Trial 3
Fan	CFM	RPM	CFM	RPM	CFM
B1	23,138	560	22,589	560	19,897
B2	23,115	562	23,466	562	22,107
B3	24,004	560	23,988	560	21,124
B4	21,969	561	23,316	562	21,017
B5	22,535	539	23,110	539	21,425
M1	25,949	562	26,897	562	23,336
M2	26,429	563	27,373	563	24,320
M3	26,294	561	27,468	561	24,738
M4	25,294	563	25,774	563	24,427
M5	22,688	561	26,126	560	23,848
M6	24,250	561	25,189	562	23,560
T1	26,254	566	27,090	565	22,429
T2	25,938	560	26,478	560	24,106
T3	27,717	562	27,466	562	22,364
T4	27,775	561	26,864	562	22,879
T5	25,578	560	27,885	561	22,099
T6	26,401	562	26,888	563	21,785
T7	26,124	562	27,276	560	20,475
T8	25,214	563	27,690	564	22,590
T9	26,243	558	27,507	560	22,650
T10	24,877	564	26,786	563	22,028
T11	24,381	560	27,691	560	22,049

Barn No. 2
The individual volumetric output and fan blade rotational speed for each fan in barn No. 2 is shown in Table 5 for trial Nos. 1-3. The average output for fans and static pressure in barn No. 2 were 37,262 cfm @ 0.05 s.p.w.g., 36,018 cfm @ 0.06 s.p.w.g., and 38,492 cfm @ 0.08 s.p.w.g. for trials 1 through 3, respectively. Fan T1 was not measured because there was no grate attached to the discharge cone of the fan. The lack of a grate prevented the accurate, steady placement of the anemometer and presented a safety hazard. The average rotational speed was 431.9 RPM for trial No. 1 and 429.0 RPM for trial No. 2. Fan rotational speed was not checked for trial No. 3.

Table 5. In-place fan performance for barn No. 2.

	Trial 1		Trial 2		Trail 3
Fan	CFM	RPM	CFM	RPM	CFM
B1	41,401	463	34,996	428	38,732
B2	43,899	472	44,025	471	40,611
B3	22,005	326	19,011	323	41,570
B4	43,481	473	44,430	472	41,126
B5	42,369	470	42,543	470	36,953
B6	42,045	473	43,079	470	39,897
B7	43,673	463	42,318	462	40,222
B8	19,611	323	13,208	446	36,163
B9	35,402	422	35,717	410	38,315
B10	35,533	420	32,937	397	34,536
T1	--	470	--	467	--
T2	28,584	371	30,035	372	43,449
T3	30,621	373	29,435	351	39,027
T4	42,627	472	42,558	471	41,026
T5	42,355	460	41,483	470	31,971
T6	41,547	465	43,155	462	41,136
T7	35,021	397	31,397	360	38,206
T8	42,324	465	41,661	448	37,581
T9	35,566	400	33,201	379	37,971
T10	39,905	460	39,146	451	32,859

Barn No. 3

The volumetric output and average fan blade rotational speeds for each fan in barn No. 3 are shown in Table 6. The average volumetric output for trial Nos.1 through 3 respectively were 27,181 cfm, 27,604 cfm, and 27,612 cfm. The static pressure for trial No.1 was 0.08 and for trial No. 2 was 0.05 s.p.w.g. The average blade rotation for trial No. 1 was 539.1 RPM and 536.2 RPM for trial No. 2. Fan rotational speed and static pressure were not measured for trial No. 3.

DISCUSSION

It was observed that wind played a large role in how the fans performed. Depending on the direction of the wind, measurements showed that fan discharge can either be enhanced or impeded. For example, fan capacity for barn No. 3 was measured on one day when wind velocity was 300-400 ft/min. Data analysis showed that the fan efficiency for this trial dropped 12% lower than its theoretical in-place value and this data was subsequently discarded. For this analysis, we defined fan efficiency as the fan's theoretical value (BESS lab data) minus the measured in-place value divided by the theoretical value. All data reported herein were collected when little or no wind was present (less than 20 ft/min). Air temperature will also affect fan discharge rates due to the change in air density. However, because this density difference is relatively small except over very large temperature changes, this discrepancy was neglected. All measurements were made within a 15°F temperature range.

Table 6. In-place fan performance for barn No. 3.

	Trial 1		Trial 2		Trial 3
Fan	CFM	RPM	CFM	RPM	CFM
B1	25,931	538	24,267	534	23,863
B2	27,220	545	28,931	542	27,293
B3	27,903	546	26,525	542	27,320
B4	20,154	515	24,279	511	25,247
B5	25,737	516	27,261	512	27,509
B6	27,406	540	27,470	539	30,415
B7	25,089	538	27,161	535	25,700
B8	25,850	537	26,975	533	25,379
B9	28,378	540	29,363	538	28,488
B10	25,539	543	24,366	540	24,611
T1	29,940	541	28,786	538	28,757
T2	29,695	545	30,294	542	30,075
T3	29,476	543	29,547	541	28,580
T4	30,062	548	30,536	545	29,812
T5	28,972	540	29,044	539	29,229
T6	30,152	543	28,716	540	30,094
T7	29,745	540	29,084	539	29,492
T8	28,779	539	26,718	535	28,522
T9	28,324	544	29,165	542	28,837
T10	19,258	540	23,594	536	23,016

Barn No. 1

The calculated average air velocity through barn No. 1 was 243.3 ft/min. This value is based on the average volumetric flow rates from each trial. This air velocity is 82.2% of the value predicted when using BESS lab data for fans running under similar static pressure conditions. The theoretical output for a 51” Aerotech Vortex fan with these attachments is 29,900 cfm at a static pressure of 0.05 s.p.w.g. The theoretical rotational speed for the Vortex fan was 575 RPM. The fan speed for trial Nos. 1 and 2 were 97.5% of the maximum. These fans were well maintained and according to the rotational speed measurements, the fans were running properly for the first two trials. The discrepancies in the flow rates could be due to the slight differences between the measured and theoretical static pressures. This difference was 0.02 inches of water for trial Nos. 1 and 2. Discrepancies could also, in part, be due to the cleanliness of the fans at the time of measuring. Prior to trials 1 and 2, the tunnel fan blades, housing, and discharge cones had been thoroughly cleaned. When trial 3 had been conducted (one year later), however, there had been no recent maintenance of the tunnel fans.

The data showed that fan location also played a large role in air discharge rate of a fan. The fans placed the highest on the barn endwall (T1-T11) had the highest efficiencies (average = 26,045 cfm, 27,332 cfm, and 22,314 cfm for trials 1, 2 and 3 respectively), while the fans at cow level (B1-B5) had the lowest efficiencies (average = 22,052 cfm, 23,293 cfm, and 21,114 cfm for trials 1, 2, and 3 respectively). This was most likely due to the fans on the highest level having the least amount of competition for air converging towards the fan bank. There were no obstructions in front of the highest fans, unlike the ground level ones that had animals, stalls, and gates blocking airflow. Also, the high level fans were able to draw air from above (no ceiling) and share air with the fans from below in addition to the air at their level. The low level fans can draw some air from the fans above, and air at their level, but are blocked by the ground from drawing air from below.

Fans on the left side of barn No.1 (fan Nos. T1-T2 averaged 25,383 cfm for trial Nos. 1, 2, and 3; fan Nos. M1-M3 averaged 25,867 cfm, for trial Nos. 1, 2, and 3; fan Nos. B1-B3 averaged 22,604 cfm for trial Nos. 1, 2, and 3) performed better than the right side (fan Nos. T10-T11 averaged 24,636 cfm for trials 1, 2, and 3; fan Nos. M4-M6 averaged 24,573 cfm for trials 1, 2, and 3; fan Nos. B4-B5 averaged 22,229 cfm for trials 1, 2, and 3). This may be due to the connecting barn that leads to the naturally ventilated milking center located half way down the left side of the barn. This large opening was not fully sealed, and therefore the fans draw air from this area in addition to the inlet endwall of the barn. This decreased the airflow through the barn. A summary of the averaged data, by trial for barn No. 1, is shown in Table 7.

Table 7. Summary table for barn No. 1.

Variable	Trial 1	Trial 2	Trial 3
Ave. Output for All Fans (cfm)	25,099	26,133	22,512
Fan Efficiency (percent)	83.9	87.4	75.3
Ave. Static Pressure (in. w.g.)	0.04	0.06	0.05
Ave. Fan Speed (rpm)	561	561	--
Percent Theoretical Fan Speed	97.5	97.5	--

Barn No. 2

The calculated average air velocity through barn No. 2 was 390.5 ft/min. The manufacturer was unable to provide theoretical data for the fans operating with discharge cones and grates in place, but without shutters, therefore, overall efficiency of the fans could not be calculated. Based on the theoretical blade rotation value for the fans (490 RPM), the fans ran at an average of 12% less than the theoretical. Two fans in particular, B3 and B8, ran at less than 70% of the theoretical. Visual observation showed that the fans were not properly maintained on a regular basis. This is a major reason why several of the fans were not performing to their potential. Figure 5 shows a direct correlation between airflow and fan rotational speed. Improving blade rotation would correct the discrepancies in airflow, as was shown in trial No. 3 data. (Fans were serviced by the farm between trial Nos. 2 and 3.)

Fan location does not seem to play as large a role with barn No. 2 as it did with barn No. 1 (top level average = 37,550 cfm for trials 1, 2, and 3; bottom level average = 36,994 cfm for trials 1, 2, and 3). Barn No. 2 had an insulated ceiling that limits the area from which the upper fans can draw air. Horizontal position also does not appear to play a role in fan discharge. Unlike barn No. 1, the milking center associated with barn No. 2 is tunnel-ventilated, resulting in a nearly static air condition within the barn connecting the two together. This condition results in little chance for additional air source for the tunnel ventilation fans in barn No. 2. A summary of the averaged data, by trial, is shown in Table 8.

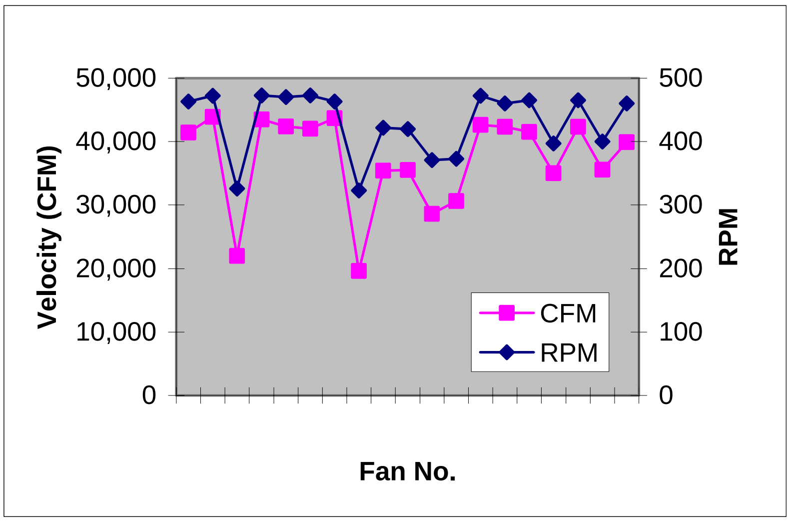

Figure 5. Barn No. 2 velocity and fan blade rotation comparison.

Table 8. Summary table for barn No. 2.

Variable	Trial 1	Trial 2	Trial 3
Ave. Output for All Fans (cfm)	37,262	36,018	38,492
Fan Efficiency (percent)	--	--	--
Ave. Static Pressure (in. w.g.)	0.05	0.06	0.08
Ave. Fan Speed (rpm)	434	429	--
Percent Theoretical Fan Speed	88.7	87.6	--

Barn No. 3

The calculated average air velocity through barn No. 3 was 591 fpm. This value was based on using the average value for the average volumetric flow rates measured in trial Nos. 1 through 3. The air velocity was 90.4% of the value using theoretical fan capacity, (643 fpm) predicted from the airflow rates from the manufacturer’s data and cross-sectional area of the barn. The calculated average volumetric flow rate through the barn was 549,320 cfm compared to the theoretical value of 598,000 cfm. This barn has the same make and model fans as barn No. 1. The fan blade rotational speeds for trial Nos. 1 and 2 are both approximately 94% of the theoretical.

There were no trends between top and bottom row fans. This was due to the insulated ceiling and lack of large obstructions placed in front of the fans. These fans were brand new when trial No. 1 was conducted, and thoroughly cleaned by the farm prior to the 2003 tunnel-ventilation season. Several factors contributed to the high efficiency of these fans. This barn was designed to specifically function as a tunnel barn. It was not converted from a naturally ventilated facility, as were barn Nos. 1 and 2. The walls contain minimal air leaks and the parlor walkway was sealed with a plastic drop curtain. In order to achieve the desired airflow, Barn No. 3 needed more fans than could fit on the flat endwall. Because of the findings from Simmons et al., (1998) the option for placing fans on the sidewall at a 90-degree angle to the airflow was rejected. Angling the endwall allowed the desired number of fans to be installed without forcing the air to make abrupt turns before discharge. The angled endwalls funneled the air into a smaller area allowing for a more gradual, smoother discharge of the air. This design significantly decreases the amount of air hitting the flat walls, therefore the air is able to remain streamlined as it approaches the fan inlet. Flat endwalls positioned normal to the flow pulling air towards the end of the barn discharge a portion of the air out, while the rest hits the wall and must change direction before being discharged. The air that hits the wall and must be redirected towards the fan inlet and thereby decreases the overall performance efficiency of the fan. Angling the endwall actually improved the performance of the fans. The 2 fans (B10 and T10) positioned closest to the overhead door of the feed alley were less efficient than the others. Because the air approaching these fans comes at an obtuse angle from the feed delivery alley, the air tends to overshoot these first 2 fans and be discharged from the next 8 fans along the endwall. Simmons, et al. stated that fans are frequently positioned with the intent to minimize the amount of dead air spaces rather than decreasing performance losses. This design does leave a pocket of dead air along the rear overhead door positioned between the 2 sets of fans. However, this is not a drawback because air exchange is not needed in this area. Barn No. 3 is also a three-row barn (versus Nos. 1 and 2 that are six-row barns). This smaller width allows for more efficient airflow. A summary of the data for barn No. 3 is shown in Table 9. Table 10 compares actual vs. theoretical values, based on using data average for all three trials.

Table 9. Summary table for barn No. 3.

Variables	Trial 1	Trial 2	Trial 3
Ave. Output for All Fans (cfm)	27,181	27,604	27,612
Fan Efficiency (percent)	90.9	92.3	92.3
Ave. Static Pressure (in. w.g.)	0.08	0.05	--
Ave. Fan Speed (rpm)	539	536	--
Percent Theoretical Fan Speed	93.7	93.2	--

Table 10. Comparison of actual vs. theoretical values for various parameters based on average values.

	Barn No. 1		Barn No. 2		Barn No. 3	
	Actual	Theor.	Actual	Theor.	Actual	Theor.
Avg. Air Velocity (fpm)	243.3	295.9	390.5	--	590.7	643.0
Avg. Airflow per Fan (cfm)	24,581	29,900	37,257	--	27,466	29,900
Average Total Airflow (cfm)	540,787	657,800	745,149	--	549,320	598,000
No. Room Vol. Exchanges	0.61	0.74	0.51	--	1.68	1.83
Avg. Blade Speed (RPM)	560.6	575	430.5	490	537.5	575

CONCLUSIONS

Based on the data collected from the three tunnel ventilated barns, the following conclusions were made:

1.) The contributing area method was determined to be the best means for quantifying airflow.
2.) The theoretical and in-place fan capacities are not the same and this should be taken into account when designing a tunnel-ventilated barn.
3.) Placing fans at a 60-degree angle with respect to the direction of airflow did not appear to significantly affect fan performance.
4.) The fans perform best when placed away from obstructions and higher off the floor.
5.) Regular fan maintenance is important.
6.) Regular fan maintenance ensures that fans perform closer to their maximum potential at all times.

REFERENCES

Wheeler, E.F., R. Bottcher. 1995. Evaluating Mechanical Ventilation Systems, Evaluating
Livestock Housing Environments.

Shearer, J.K., D.K. Beede, R.A. Bucklin, and D.R. Bray. 1991. Environmental Modifications to Reduce Heat Stress in Dairy Cattle. *Agri-Practice*, Volume 12, No. 4, July/August, 1991.

Simmons, J.D., I.E. Hannigan, and B.D. Lott. 1998. Flow Rate Variations Resulting from Placement of Large Fans in Tunnel Ventilated Broiler Houses. 1998 *ASAE Annual Int'l Meeting* 1998.

Parry, R. (2003) Personal Contact. US EPA, Office of Water.