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Similitude Study of Airflow Characteristics and Evaluation of Ridge Vent Design for an Open Front Confinement Beef Barn

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SIMILITUDE STUDY OF AIRFLOW CHARACTERISTICS AND EVALUATION OF RIDGE VENT DESIGN FOR AN OPEN FRONT CONFINEMENT BEEF BARN

BY

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SIMILITUDE STUDY OF AIRFLOW CHARACTERISTICS

AND EVALUATION OF RIDGE VENT DESIGN FOR

AN OPEN FRONT CONFINEMENT BEEF BARN

This thesis is approved as a creditable and independent investigation by a candidate for the degree, Master of Science, and is acceptable for meeting the thesis requirements for this degree. Acceptance of this thesis does not imply that the conclusions reached by the candidate are necessarily the conclusions of the major department.

Thesis Adviser — Date

Major Adviser, Head
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Department — Date

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INTRODUCTION

Production of beef cattle in the United States has developed into a major industry with the most dynamic increase taking place since 1940. In 1968 the United States produced 109 million head of beef cattle worth over 16.2 billion dollars (21). South Dakota's economy is highly dependent upon agriculture and beef cattle production is the major source of income for South Dakota farmers. In 1964 beef cattle production was evaluated at $285,700,000, equal to 42.3 per cent of all crop and livestock income. Nationally South Dakota ranks ninth in the number of cattle on feed (6).

Predictions for the future indicate that beef cattle production will continue to increase. The U. S. Department of Agriculture has estimated that the demand for red meat would increase by over 60 per cent between 1958 and 1975 (7). Beef consumption in the United States is 80 to 85 pounds per capita per year. However, some light meat, turkey and chicken will be substituted for red meat, but due to the continuing increase in human population, the production of beef will have to increase to meet the domestic need (13).

Following the trend of the last few years, more beef cattle production is going to take place in feedlots. The feedlots will become larger and the production of beef on natural grassland will decrease partly due to the increased performance realized from using more concentrated rations as tests indicate that about 20 per cent less total feed is needed to produce one pound of gain with a concentrate-roughage ratio of 5:1 (five times more concentrate than roughage), as compared to commonly used 6:4 rations.

Better pollution control, improved environmental conditions for livestock and farm laborers and optimization of labor will hasten the trend toward beef cattle produced in confinement buildings. With more beef cattle production taking place in confinement buildings, the optimum environmental conditions for maximum performance must be defined and design criteria established. Factors such as ambient temperature, relative humidity, wind velocity, precipitation, solar radiation, type of shelter, animal density and conditions of the environmental surfaces affect the thermal environment of the beef cattle. One of the most important factors to consider in trying to achieve maximum animal production in confinement buildings is proper ventilation. The ventilation system must supply adequate oxygen, remove toxic gases and water vapor and prevent condensation and frost accumulation in the building. Several ways of ventilating open and semi-open beef barns are being employed, but there is limited information on systems which will operate efficiently and effectively under different climatic conditions.

This study was undertaken to evaluate ventilation characteristics in open front beef confinement buildings. The objectives of this study were:

1. Evaluate the effects of ridge vent design on

air flow characteristics in a model of an open front beef confinement building.

2. Determine the effect of ridge vent design on temperature in a model of an open front beef confinement building.

3. Develop prediction equations for air flow using a model of an open front beef confinement building.

REVIEW OF LITERATURE

Effects of Environment on Beef Cattle Production

Comparison of beef cattle performance in various environments and between experiments show great variation and a general trend is difficult to determine. In many cases large parts of these variations can be explained by the differences in climate.

Beef cattle are given less protection against winter climatic conditions than any other domestic livestock, Bond et. al. (3). In a literature review Nelson (17) reports that hot weather has a greater effect on beef cattle than cold weather does. Several other reports, Kelley (1), Johnson (10) and North Central Region's Publication (18), have had similar conclusions. Hellickson, Witmer and Barringer (8) found that beef cattle finished in pole type and closed environments during the winter were not significantly affected by environment even though pole barn temperature averaged 18° F colder than closed barn temperature. During the summer the beef cattle in the pole barn had significantly higher daily gains than the cattle in closed environment. The temperature in the closed environment averaged 4.3° higher than pole barn temperature. Bates et. al. (2) indicated that cattle fed in warm confinement gained from 0.07 to 0.28 pounds more per head per day than cattle in sheds. Cattle fed outside with no weather protection gained even less.

Bond et. al (3) found that muddy surfaces have a greater effect on daily gain than winter temperature that ranged from 40 to 60° F. and averaged 50° F. Cattle exposed to rain gained less than cattle exposed to wind, but cattle exposed to rain and wind gained more than cattle in muddy environments. They also reported that calves raised in Canada "in weather as cold as that experienced anywhere in existing areas of domestic livestock production" performed as well as those maintained indoors at 70° F. Another group raised with no protection from climatic conditions did not perform so well, probably because of exposure to wind and rain.

Olson and Roth (19) found, from a survey made among farmers in 27 states and four Canadian provinces, that 52 per cent of the farmers used housing for their beef cattle. For the North Central and North Eastern states, Alaska and four Canadian provinces the percentage was 56.

Schulz (23) states that in northern areas of the United States, sheds with openings to the south have adequate ventilation and that cold weather conditions have not been shown to have detrimental effects on beef cattle. Kelly (11) also reports that open and semi-open beef shelters will provide sufficient circulation of air. However, for closed and open front barns that are closed for a period of time, Kelly recommends electric fans or inlet and outlet ducts to insure circulation of air. In a survey made among 15 North Dakota farmers, Johnson (10) concluded that condensation "was not an apparent problem" in open front buildings used for inside feeding and providing access to outside yards. Johnson reported that some farmers were not satisfied with the open type ridge vent and experimented with covers over the openings to prevent rain and snow from falling into the building. However, the covers did not eliminate the problems, but instead directed snow along the roof and funneled it down into the building. Other farmers have covered part of the open wall in order to reduce inside drafts.

The Dairy Cattle Housing Sub-Committee, North Central Region (18) says that barns open to the south or east are comparatively free of condensation. However, if air circulation seems to be insufficient, they recommend providing louvered gable ends or ridge ventilators but they do not provide specific ridge vent designs. Lubinus (14) reports that condensation and frost frequently occur in cold open and semi-open confinement beef and dairy barns in the North Central United States. He has found that condensation usually occurs on the underside of uninsulated metal roofs at outside temperatures lower than 0° F. Condensation can be reduced if the air circulation maintains a temperature difference between inside and outside air of 10° F or less and if the underside of the roof is insulated with one-half inch of foam insulation or the thermal resistance value (R) of the roof is 2.5 to 3.0. For building design, Lubinus recommends insulation on the underside of the roof for buildings located where 54 hours of temperatures lower than 0^{o} F. normally are recorded from December through February (Figure 1). Burns et. al. (5) state that inadequate ventilation of dairy barns, open on the south side, during extreme periods of cold weather will cause condensation and frost on the underside of the roof. Therefore, they recommend some kind of ridge vent to improve ventilation and reduce condensation.

Model Studies of Ventilation Systems

Model studies employing simulation techniques have proved to be useful in the solution of ventilation problems.

In a similitude study of ventilation inlet configuration, Smith and Hazen (24) concluded that models of air inlets successfully predicted the prototype air flow characteristics. They further state that models can effectively describe the velocity distribution and the shape and velocity of the air jet. When the Reynolds number and the geometry of the inlets are similar in the model and prototype, geometric similarity of the jet velocity profiles were obtained. Wilson, Esmay and Persson (27) conclude in a study of nonisothermal wall jet velocities and temperature profiles that effects of buoyancy forces on the velocity profiles were negligible at velocities above 800 ft/min (13.3 ft/sec) and temperature differences of 50^{o} F. or less. Below these velocities and at the same temperature difference buoyancy forces appeared to affect air flow. Pattie and

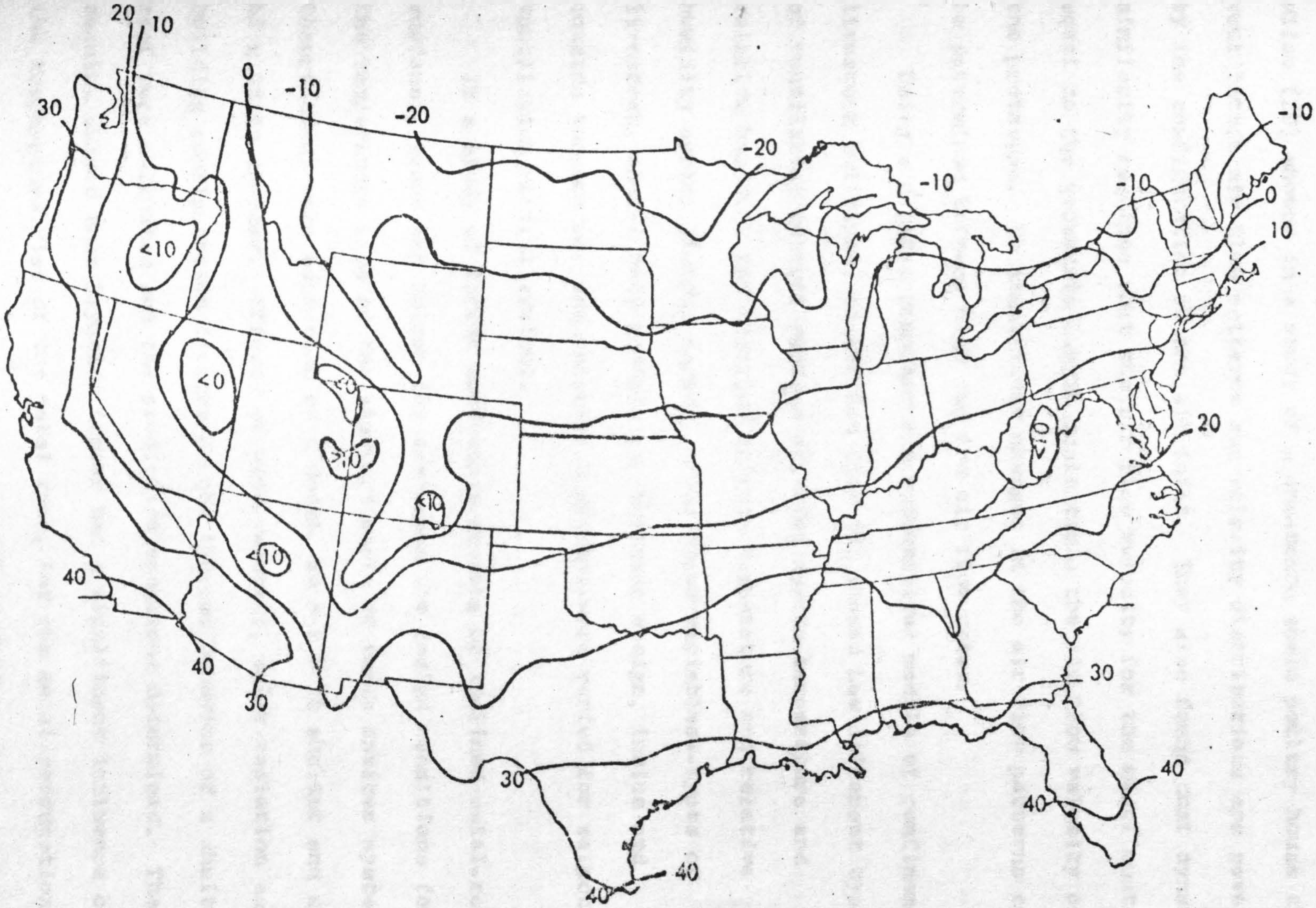

Figure 1. Probability (2½%) of Colder Winter Temperatures, December-February

Milne (20) showed in a study of a one-tenth scale poultry house that ventilation air flow patterns and velocity distributions are governed by the configuration of the air inlet. They also found that dynamic similarity requires that the air flow velocity for the model must be equal to the geometric length scale times the air flow velocity of the prototype. No significant changes in the air flow patterns could be determined between high and low air flow rates.

Using a digital computer and mathematical models of confinement livestock buildings, Hinkle and Good (9) showed how different types of ventilation control systems affected inside temperature and relative humidity for changing outside temperature and relative humidity during 24-hour periods. The input variables--types of livestock, animal heat production, building design, inside and outside temperature and relative humidity--were varied for selected ventilation control systems.

In a study of forced convective cooling of inclined metal-roof surfaces, Braud and Nelson (4) developed the design conditions for the temperature rise of the windward roofs of three shelter systems. Observations were conducted on a model, an 8 x 8 ft shelter and a 48 x 48 ft shelter. Effects of wind velocity, solar radiation and building configuration factors on the thermal behavior of a shelter roof were evaluated and the prediction equations determined. The results showed that Reynolds number had a significant influence on the temperature rise of the metal roof, but the metal corrugation size had little effect on forced convective cooling.

DETERMINATION OF PERTINENT VARIABLES

The purpose of this study was to evaluate the effects of ridge vent design on ventilation characteristics and temperature in an open front beef confinement building. A 70 ft by 96 ft pole-type barn with a capacity of 200 head of 800 to 1200 pound beef cattle was selected. Due to reduced expenses and better control of the variables, it was decided to conduct this investigation as a model study applying the principles of similitude.

The rate of air flow through the ridge vent, which essentially acts as a rectangular orifice, is affected by fluid properties, such as the ratio of inertia forces to viscous forces, the ratio of inertia forces to gravitational forces and the ratio of buoyancy forces to viscous forces. Building geometry factors, such as length and width of the outlet and length, width and height of the building also influence the rate of air flow. The rate of air flow through the ridge vent was hypothesized to be a function of the variables affecting air flow and the properties of the structural system. Assuming that the same phenomenon govern the performance in the model and prototype, a list of the pertinent variables affecting the ventilation characteristics was compiled (Table 1).

The functional relationships between the pertinent variables can be expressed as $V_o = f\,(l, w, h, s, W_o, l_o, \Delta t, r, V_w, q, N, \beta, \rho, \mu, g, c, k)$.

TABLE 1

VARIABLES AFFECTING VENTILATION CHARACTERISTICS

Variable No.	Symbol	Description	Dimensional Symbol*
1.	V_o	Velocity of outlet air	LT^{-1}
2.	l	Building length	L
3.	w	Building width	L
4.	h	Building height	L
5.	s	Ride of roof	L
6.	W_o	Width of the outlet	L
7.	l_o	Length of the outlet	L
8.	Δt	Temperature difference (Inside-Outside)	θ
9.	r	Moisture content, inside air	-
10.	V_w	Wind velocity	LT^{-1}
11.	q	Total animal heat production	L^2T^{-3}
12.	N	Animal density	$FL^{-4}T^2$
13.	β	Coefficient of thermal expansion	θ^{-1}
14.	ρ	Inside air density	$FL^{-4}T^2$
15.	μ	Dynamic viscosity of the inside air	$FL^{-2}T$
16.	g	Acceleration of gravity	LT^{-2}
17.	c	Specific heat of building materials	$L^2T^{-2}\theta^{-1}$
18.	k	Thermal conductivity of building materials	$FT^{-1}\theta^{-1}$

*L, F, T and θ are the basic dimensions of length, force, time and temperature, respectively.

Employing dimensional analysis and the Buckingham Pi Theorem (16), a set of 14 independent and dimensionless groups, π terms, (Table 2), was derived (18 variables—the 4 basic dimensions of force, length, time and temperature). The dimensionless form can be expressed as

$$\frac{y_o}{V_w} = F\left[\frac{w}{l},\frac{h}{l},\frac{w_o}{l},\frac{l_o}{l},\frac{k\ \Delta t}{NlV_w{}^3},r,\frac{qNl^2}{k\ \Delta t},\frac{l^3\rho^2\beta g\Delta\ t}{\mu^2},\frac{\rho}{N},\frac{\rho V_w l}{\mu},\frac{V_w{}^2}{gl},\frac{c\mu}{k}\right] \quad \text{Equation 1}$$

In establishing the dimensionless groups commonly used pi terms were derived whenever possible and appropriate. These include: Reynolds number (N_{RE}), which relates the ratio of inertia forces to viscous forces; Froude number (N_{FR}), which relates the ratio of inertia forces to gravitational forces and Grashof number (N_{GR}), which relates the ratio of buoyancy forces to viscous forces.

Since the functional relationship expressed in Equation 1 is general, it also applies to any other system, which is a function of the same variables. It, therefore, represents the model system and can be written as

$$\left(\frac{V_o}{V_w}\right)_m = F\left[\frac{w}{l},\frac{h}{l},\frac{s}{l},\frac{w_o}{l},\frac{l_o}{l},\frac{k\ \Delta t}{NlV_w{}^3},r,\frac{qNl^2}{k\ \Delta t},\frac{l^3\rho^2\beta g\Delta t}{\mu^2},\frac{\rho}{N},\frac{\rho V_w l}{\mu},\frac{V_w{}^2}{gl},\frac{c\mu}{k}\right]_m \quad \text{Equation 2}$$

(subscript m referring to the model). Employing the theory of models (16), π_1 equals π_{1m}, if the corresponding independent pi terms for the model and the prototype are equal. From Equations 1 and 2 $\pi_{1m} = \pi_1$, or $\left(\frac{V_o}{V_w}\right)_m = \frac{V_o}{V_w}$, if the design conditions listed in Table 3 are satisfied.

Design conditions 1 through 5 (Table 3) indicate the requirements of geometric similarity between the model and the prototype with

TABLE 2

LIST OF Pi TERMS

Pi Term No.		Description
1.	$\pi_1 = V_o/V_w$	Dependent Pi term
(Pi terms concerning building geometry constant for this study)		
2.	$\pi_2 = w/l$	
3.	$\pi_3 = h/l$	
4.	$\pi_4 = s/l$	
5.	$\pi_5 = w_o/l$	
6.	$\pi_6 = l_o/l$	
7.	$\pi_{14} = c\mu/k$	Prantl number
(Pi terms dependent upon environmental conditions)		
8.	$\pi_7 = k\ \Delta t/NlV_w^3$	
9.	$\pi_8 = r$	Moisture content in the air
10.	$\pi_9 = qNl^2/k\ \Delta t$	
11.	$\pi_{10} = l^3\rho^2\beta g\Delta t/\mu^2$	Grashof number
12.	$\pi_{11} = \rho/N$	
13.	$\pi_{12} = \rho V_w l/\mu$	Reynolds number
14.	$\pi_{13} = V_w^2/gl$	Froude number

TABLE 3

DEVELOPMENT OF DESIGN CONDITIONS

No.	Basic Equation	Design Conditions
1.	$\left(\frac{w}{l}\right)_m = \frac{w}{l}$	$\frac{w}{w_m} = n$
2.	$\left(\frac{h}{l}\right)_m = \frac{h}{l}$	$\frac{h}{h_m} = n$
3.	$\left(\frac{s}{l}\right)_m = \frac{s}{l}$	$\frac{s}{s_m} = n$
4.	$\left(\frac{w_o}{l}\right)_m = \frac{w_o}{l}$	$\frac{w_o}{w_{om}} = n$
5.	$\left(\frac{l_o}{l}\right)_m = \frac{l_o}{l}$	$\frac{l_o}{l_{om}} = n$
6.	$r_m = r$	$r_m = r$
7.	$\left(\frac{l^3\rho^2\beta g\Delta t}{\mu^2}\right)_m = \frac{l^3\rho^2\beta g\Delta t}{\mu^2}$	$\frac{\Delta t}{\Delta t_m} = \frac{1}{n^3}\left(\frac{\rho_m}{\rho}\right)^2 \frac{\beta_m}{\beta} \frac{g_m}{g}\left(\frac{\mu}{\mu_m}\right)^2$
8.	$\left(\frac{c\mu}{k}\right)_m = \frac{c\mu}{k}$	$\frac{c}{c_m} = \frac{k}{k_m} \frac{\mu_m}{\mu}$
9.	$\left(\frac{\rho}{N}\right)_m = \frac{\rho}{N}$	$\frac{N}{N_m} = \frac{\rho}{\rho_m}$
10.	$\left(\frac{qNl^2}{k\ \Delta t}\right)_m = \frac{qNl^2}{k\ \Delta t}$	$\frac{q}{q_m} = \frac{1}{n^2} \frac{N_m}{N} \frac{k}{k_m} \frac{\Delta t}{\Delta t_m}$
11.	$\left(\frac{\rho V_w l}{\mu}\right)_m = \frac{\rho V_w l}{\mu}$	$\frac{V_w}{V_{wm}} = \frac{1}{n} \frac{\rho}{\rho_m} \frac{\mu}{\mu_m}$
12.	$\left(\frac{V_w^2}{gl}\right)_m = \frac{V_w^2}{gl}$	$\frac{V_w}{V_{wm}} = n\left(\frac{g}{g_m}\right)^{1/2}$
13.	$\left(\frac{k\ \Delta t}{NlV_w^3}\right)_m = \frac{k\ \Delta t}{NlV_w^3}$	$\frac{N}{N_m} = \frac{1}{n} \frac{k}{k_m} \frac{\Delta t}{\Delta t_m}\left(\frac{V_{wm}}{V_w}\right)^3$

$n = 1/l_m$ being the geometric length scale. Design condition 6 requires that the moisture content in the air for the model and prototype be the same. The requirement of design condition 7 determines the temperature scale: $\frac{\Delta t}{\Delta t_m} = \frac{1}{n^3}$, if the same fluid is used in the two systems. For $n = 20$, this design condition could not be satisfied with the laboratory facilities and equipment available for this experiment. By distorting π_{10} such that $\pi_{10m} = \alpha_1 \pi_{10}$ (α_1 a distortion factor), the temperature scale equals unity ($\frac{\Delta t}{\Delta t_m} = 1$), if $\alpha_1 = n^3$, and with $\pi_{10m} = n^3 \pi_{10}$ the experiment can be conducted using the existing facilities. Design condition 8 indicates the same material may be used in the model and prototype since $c_m = c$ and $k_m = k$. The animal density (lbm/ft^3) scale, $\frac{N}{N_m} = 1$ is obtained from design condition 9. When this requirement is substituted into design condition 10, the animal heat production (Btu/hr•lb) scale becomes $\frac{q}{q_m} = \frac{1}{n^2}$. The total animal heat production (Q, Btu/hr) is related to q, N and V (Volume) as follows: $Q = qNV$. The heat production scale becomes $\frac{Q}{Q_m} = \frac{q}{q_m}\frac{N}{N_m}\frac{V}{V_m} = \frac{1}{n^2} 1\, n^3 = n$ and $Q_m = \frac{Q}{n}$. When the heat production of cattle (1) is used, this requires that Q_m=28,800 Btu/hr (8400 watts). In order to prevent overheating of the model, π_9 was distorted, so that $\pi_{9m} = \alpha_2\, \pi_9$ (α_2 a distortion factor) and the animal heat production scale is $\frac{q}{q_m} = \frac{1}{n}$, for $\alpha_2 = n$. The total heat production for the model then becomes $Q_m = \frac{Q}{n^2}$ 1440 Btu/hr (421 watts).

The wind velocity scale can be obtained from design condition 11 or 12, $\frac{V_w}{V_{wm}} = \frac{1}{n}$ or $\frac{V_w}{V_{wm}} = n^{\frac{1}{2}}$, respectively. Design condition 11 is determined from the Reynolds number (N_{RE}) and 12 from the Froude number (N_{FR}). Which pi term will have the greatest influence on the air flow cannot be determined before tests are conducted. Assuming that Reynolds number is the most important, the wind velocity scale is $\frac{V_w}{V_{wm}} = \frac{1}{n}$. Design condition 13 will then be distorted in the following manner: $\pi_{7m} = \alpha_3 \pi_7$, and the animal density scale becomes unity ($\frac{N}{N_m} = 1$), for $\alpha_3 = \frac{1}{n^2}$.

If Froude number is assumed to be the dominant pi term, the wind velocity scale will be $\frac{V_w}{V_{wm}} = (n)^{\frac{1}{2}}$. In this case design condition 13 will be distorted so that $\pi_{7m} = \alpha_3' \pi_7$, and the animal density scale equals unity for $\alpha_3' = n^{5/2}$.

EXPERIMENTAL METHODS AND PROCEDURE

The design of the model system was conducted assuming the same material and fluid would be used in the model and prototype. The geometric length scale was arbitrarily selected to be 20, which established the length and width of the model at 57.6" and 42.0", respectively. The model, open on the south side (Figures 2 and 3), was constructed according to the design conditions listed in Table 3. Trusses, purlins and poles were made of wood and the sides and roof were constructed of 26 gauge galvanized sheet metal. An eave inlet was provided along the north side of the building.

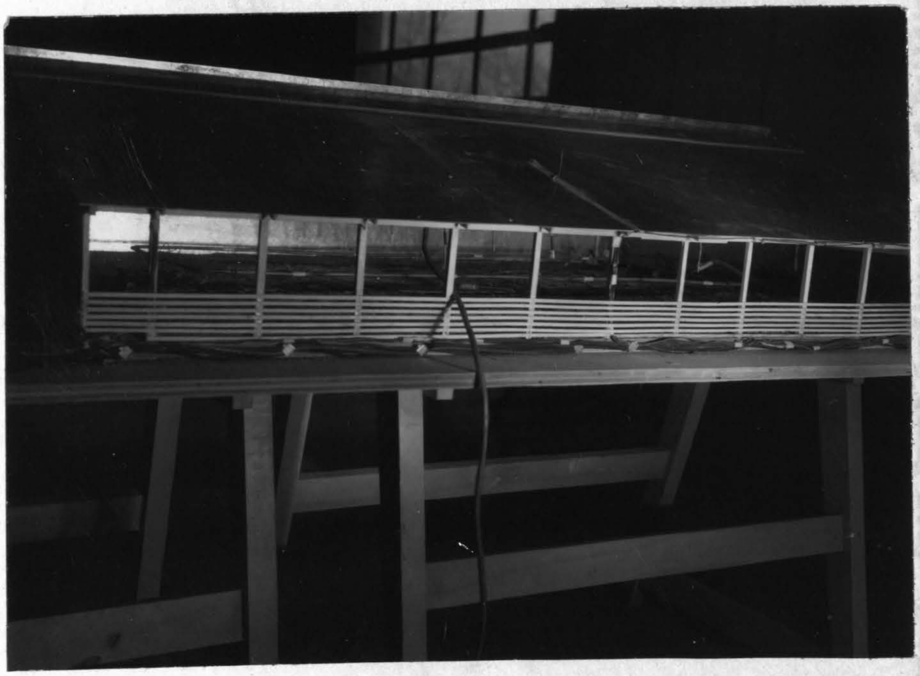

Figure 2. Model of Open Front Confinement Building, South Side

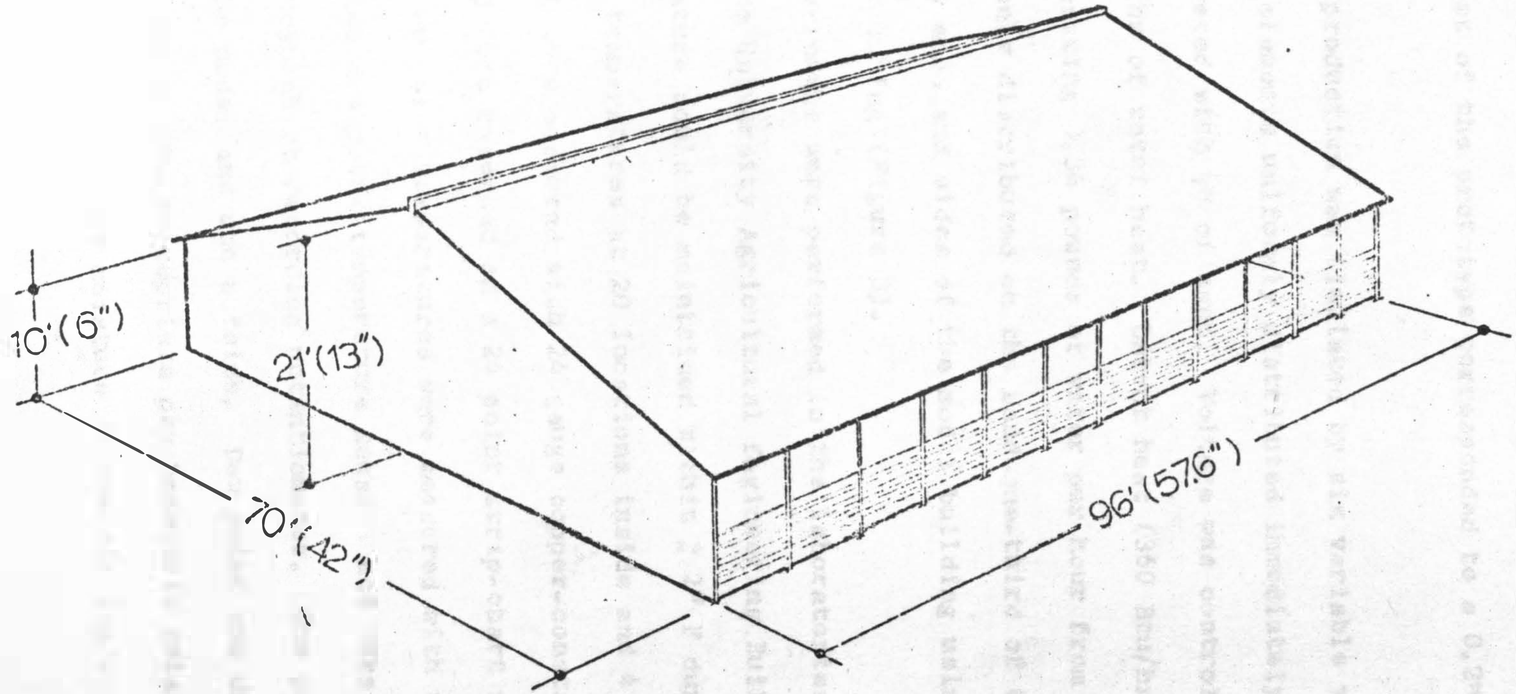

Figure 3. Open Front Confinement Beef Building. Numbers in Parenthesis Refer to the Model Dimensions.

The four ridge vents (Figure 4) were constructed of galvanized sheet metal and were reduced according to design conditions 4 and 5. The 4" ridge outlet of the prototype corresponded to a 0.2" outlet for the model.

Animal heat production was simulated by six variable voltage electric heating elements uniformly distributed immediately above the floor and covered with ½" of sand. Voltage was controlled to provide 1440 Btu/hr of total heat. Latent heat (360 Btu/hr) was produced by evaporating 0.34 pounds of water per hour from the sand. The water was evenly distributed on the sand one-third of the distance from the ends and sides of the model building using perforated copper tubing (Figure 3).

All the experiments were performed in the laboratories of the South Dakota State University Agricultural Engineering Building, where the temperature could be maintained within $\pm$ 2^{o} F during all tests. Dry bulb temperatures at 20 locations inside and 4 locations outside the model were measured with 24 gauge copper-constantan thermocouples and were recorded on a 24 point strip-chart recording potentiometer. Dew point temperatures were measured with two thermo-electrically cooled dew point temperature sensors and were recorded on a dual channel strip-chart recording potentiometer. One probe was located inside the model and one outside. Dew point and dry bulb temperatures along with the appropriate psychrometric relations were used to determine the amount of moisture in the air (π_8).

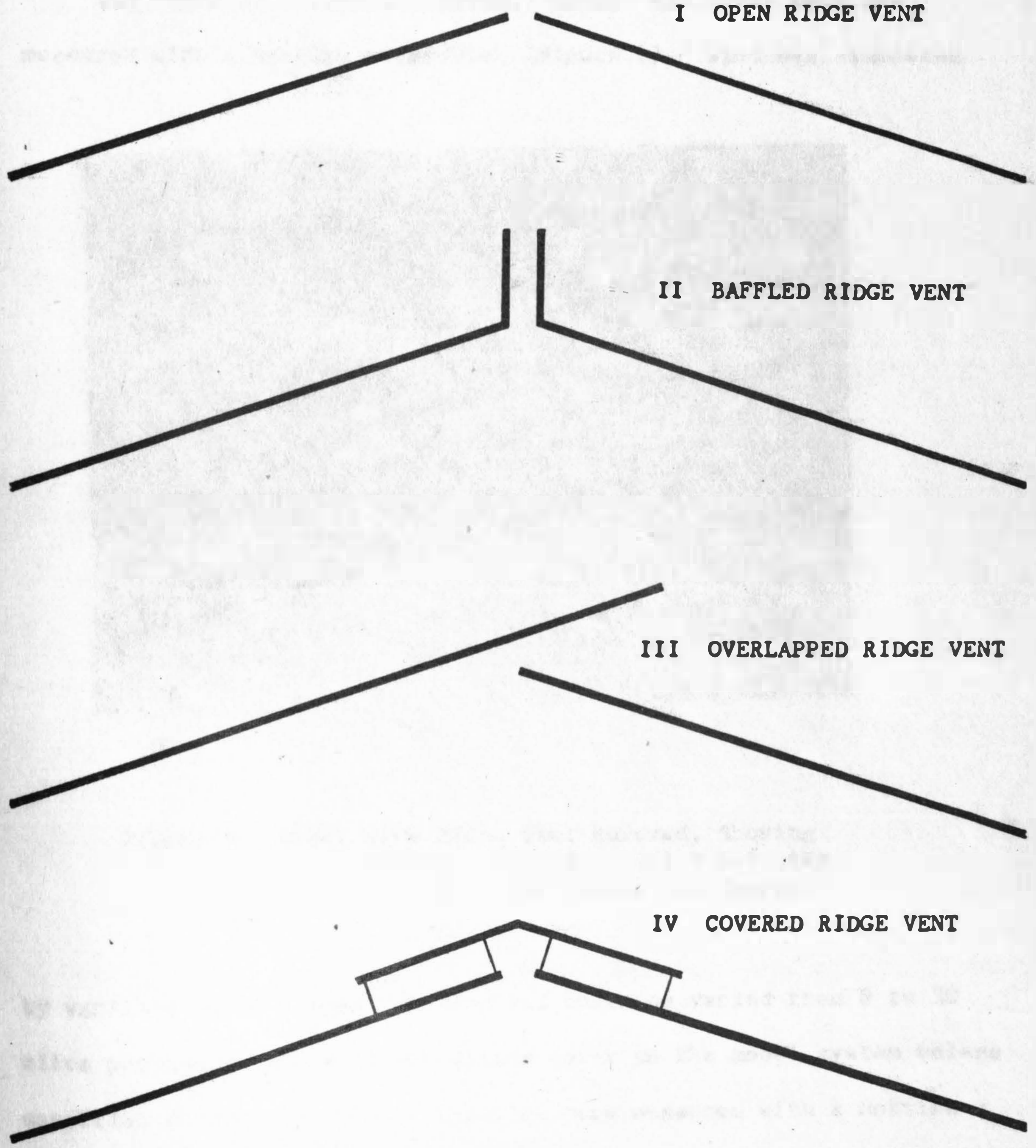

Figure 4. Types of Ridge Vents

The velocity of the air moving through the ridge vent was measured with a hotwire anemometer (Figure 5). Wind was simulated

Figure 5. Model with Ridge Vent Removed, Showing Location of Hotwire Probe for Outlet Velocity Monitoring; Note Dewpoint Sensor and Thermo-Couple.

by variable speed vane-axial fans and could be varied from 0 to 30 miles per hour. All wind velocities refer to the model system unless specified otherwise. Wind velocities were measured with a hotwire anemometer for velocities from 10 to 30 miles per hour, and a vane anemometer for velocities from 0 to 10 miles per hour. Simultaneous observations were made of wind and outlet velocities and dewpoint and drybulb temperatures during all tests. The instrumentation and the physical arrangement of the model are shown in Figure 6.

Figure 6. Model of Beef Confinement Building and Monitoring Instruments. From Left to Right: Model, Anemometer (Wind Velocity), Voltage Transformer, Dewpoint Temperature Recorder and Selector Switch (Above), Hotwire Anemometer (Outlet Velocity) and Digital Voltmeter (Output of Hotwire Probe)

The initial tests were performed from September 1 through September 3, 1971, with 15 observations conducted on each ridge vent at wind velocities from 0 to 30 miles per hour. One dependent and seven independent pi terms were calculated from the open type ridge vent data at 14 wind velocities.

A second series of tests with wind velocities varying from 0 to 14 miles per hour was conducted on September 28, 1971, and on October 17, 1971. During all tests, pi terms describing the material and geometry of the model were held constant, while pi terms describing environmental conditions varied with drybulb and dewpoint temperature and wind and outlet velocities. From the data obtained, one dependent and seven independent pi terms were calculated at 14 wind velocities for each type of ridge vent.

Multiple linear regression analyses were performed on the data obtained from the initial tests, on both logarithmic transformed and nontransformed pi terms. Pi terms from the second series of tests were analyzed using multiple linear regression analyses on logarithmic and nontransformed data. Polynomial regression analysis were used to relate outlet velocity to wind velocity, and for relating temperature difference (inside-outside) to wind velocity. The calculated regression coefficients were tested by F tests using analyses of variance. Slopes and intercepts for the air flow prediction equations were tested by F tests using analyses of covariance.

RESULTS AND DISCUSSION

The relationships between outlet and wind velocities for the open, baffled, overlapped and covered ridge vents determined from the initial tests, revealed a similar trend for all four ridge vents (Figure 7). Therefore, only data from the open ridge vent was further analyzed to determine whether Reynolds number or Froude number should be used to define the wind velocity scale. Regression analyses of transformed (log base e) data and nontransformed data were performed to evaluate exponential relations of the form

$$\pi_1 = C\ \pi_7^{b_1}\ \pi_8^{b_2}\ \pi_9^{b_3}\ \pi_{10}^{b_4}\ \pi_{11}^{b_5}\ \pi_{12}^{b_6}\ \pi_{13}^{b_7} \qquad \textbf{Equation 3}$$

(multiplicative model) and linear relations of the form

$$\pi_1 = C'+b'_1\pi_7+b'_2\pi_8+b'_3\pi_8+b'_4\pi_9+b'_5\pi_{10}+b'_6\pi_{11}+b'_7\pi_{12} \qquad \textbf{Equation 4}$$

(additive model).

Based on the multiplicative model, only Froude number significantly influenced the dependent pi term (π_1). Using the additive model, the dependent pi term was found to be significantly influenced by moisture content, $\frac{k\ \Delta t}{N1V_w{}^3}$ and Froude number (Table 4).

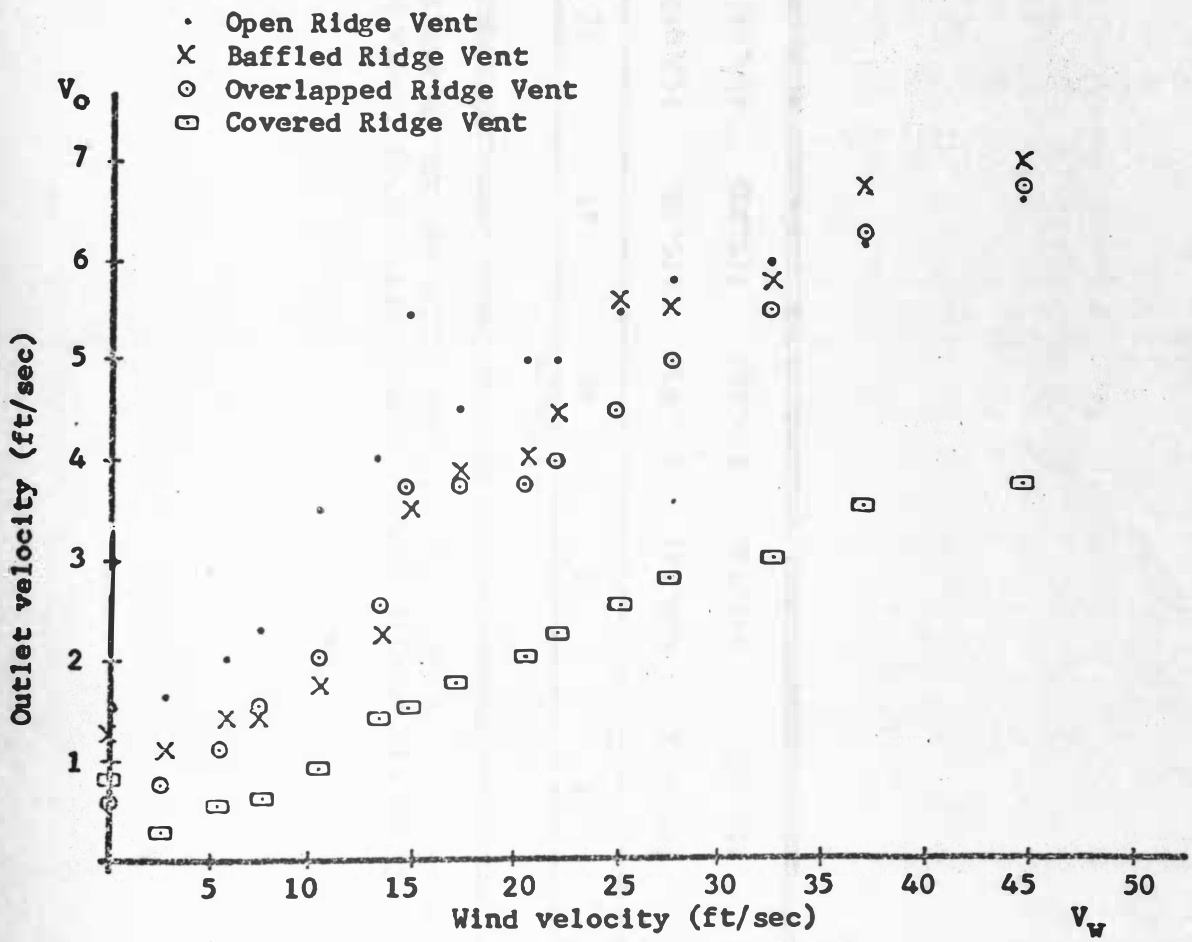

Figure 7. The Effect of Wind Velocity on Outlet Velocity, Initial Tests.

TABLE 4

ORDER OF APPEARANCE OF Pi TERMS AND COEFFICIENTS OF DETERMINATION (R^2%) (WHOLE NUMBERS INDICATE ORDER OF APPEARANCE IN THE STEP-WISE MULTIPLE LINEAR REGRESSION ANALYSIS)

Type of Relationship	Pi Terms						
	7	8	9	10	11	12	13
Linear	2*(6.8%)	1*(86.0%)	4(0.4%)	5(0.0%)	6(0.0%)	7(0.0%)	3*(4.7%)
Multiplicative	5(0.2%)	2(3.0%)	3(2.1%)	6(0.1%)	7(0.0%)	4(0.3%)	1*(88.3%)

*Significant at the 5% level.

From these results the wind velocity scale was determined from Froude number as $V_w/V_{wm} = (n)^{\frac{1}{2}}$ and another series of tests was conducted in which the wind velocity was selected to vary from 0 to 14 miles per hour.

The relationships between outlet velocity (V_o) and wind velocity (V_w) for the four types of ridge vents as determined from the second series of tests are shown in Figure 8 (variables and pi terms refer to the model unless otherwise specified). Polynomial regression analyses relating outlet velocity to wind velocity for wind velocities from 0 to 20.58 ft/sec, revealed linear relationships for the overlapped and covered ridge vents with R^2 values of 98.5 and 92.8%, respectively, a cubic relationship ($R^2 = 99.5\%$) for the open ridge vent and a quartic relationship ($R^2 = 92.1\%$) for the baffled ridge vent.

Analyses of these regression equations (Table 5) revealed that linear regression lines could be fitted to the quartic and cubic equations with only a small loss in the amount of variation predicted. Therefore, outlet velocity was related to wind velocity using linear relationships for all ridge vents. Basically the linear relationships can be expressed as $V_o = a + bV_w$ or

$$\frac{V_o}{V_w} = \frac{a}{V_w} + b \qquad \text{Equation 5}$$

Since Equation 5 indicates that the dependent pi term (V_o/V_w) is linearly and inversely related to wind velocity (V_w), relationships

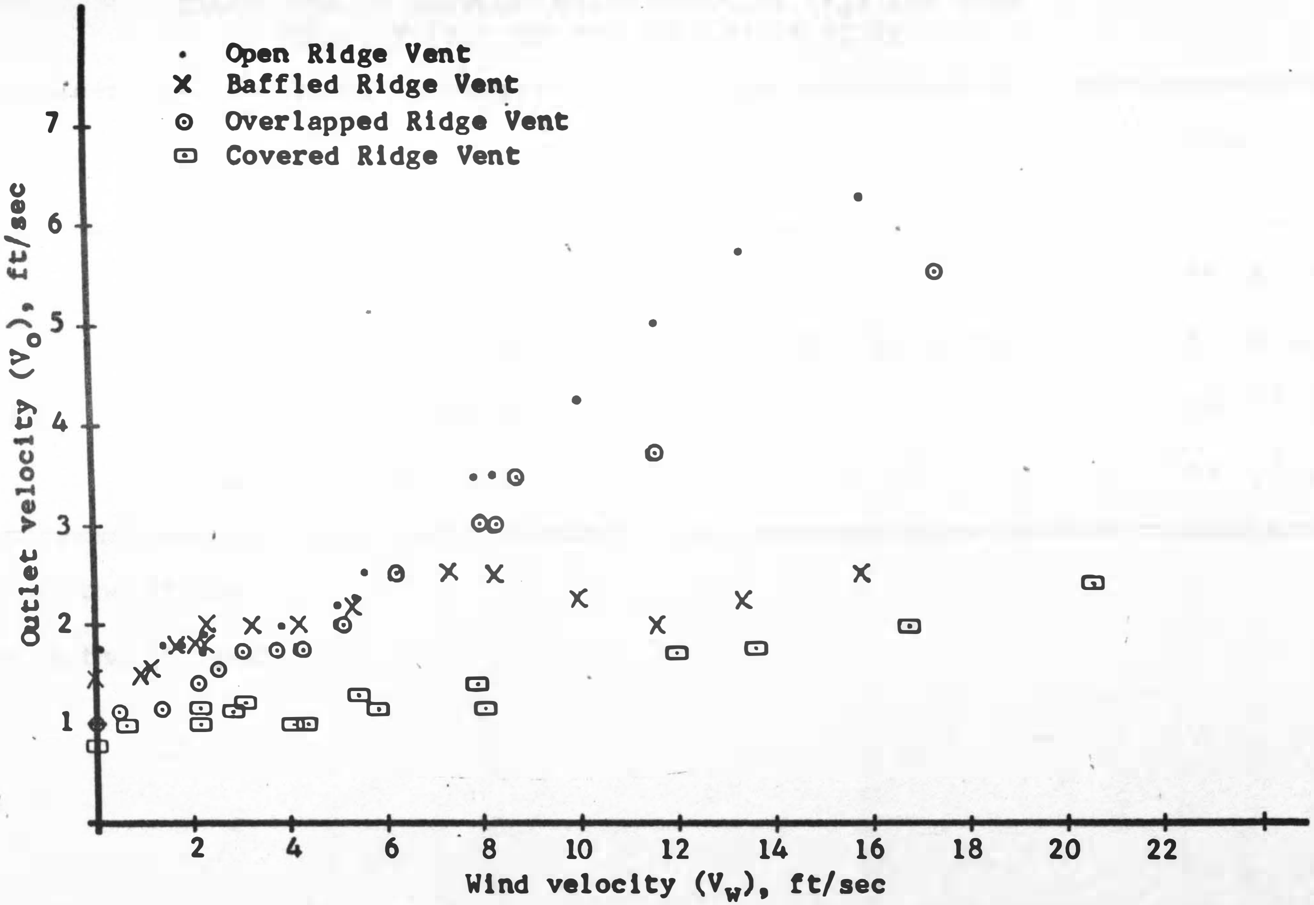

Figure 8. The Effect of Wind Velocity on Outlet Velocity, Second Series of Tests.

TABLE 5

RELATIONSHIPS BETWEEN OUTLET VELOCITY (V_o) AND WIND VELOCITY (V_w) FOR THE FOUR RIDGE VENTS

Type of Ridge Vent	Equation	Coefficient of Determination
I Open	Cubic $V_o=1.90-0.193V_w+0.066V_w^2-0.0023V_w^3$	** $R^2=99.5\%$
II Baffled	Quartic $V_o=1.40+0.147V_w+0.026V_w^2-0.052V_w^3+0.0002V_w^4$	* $R^2=92.1\%$
III Overlapped	Linear $V_o=0.84+0.26V_w$	** $R^2=98.5\%$
IV Covered	Linear $V_o=0.84+0.07V_w$	** $R^2=92.8\%$

* Significant at the 5% level.

**Significant at the 1% level.

involving the reciprocal of Reynolds number ($\frac{\mu}{\rho V_w l}$) and the reciprocal of Froude number ($\frac{\sqrt{gl}}{V_w}$) are suggested.

Regression analyses expressing the relationships between the dependent pi term and the reciprocal of both Reynolds and Froude numbers provided linear equations (Tables 6 and 7, respectively) which were highly significant for all four ridge vents.

TABLE 6

FUNCTIONAL RELATIONSHIPS FOR ($\frac{V_o}{V_w}$) AND THE RECIPROCAL OF REYNOLDS NUMBER ($\frac{\mu}{V_w \rho l}$)

Type of Ridge Vent	Equation	Coefficient of Determination
I Open	$\frac{V_o}{V_w} = 0.26+32.28\frac{\mu}{V_w \rho l}$	** R^2=94.8%
II Baffled	$\frac{V_o}{V_w} = 0.12+36.55\frac{\mu}{V_w \rho l}$	** R^2=98.2%
III Overlapped	$\frac{V_o}{V_w} = 0.22+24.78\frac{\mu}{V_w \rho l}$	** R^2=98.8%
IV Covered	$\frac{V_o}{V_w} = 0.05+24.03\frac{\mu}{V_w \rho l}$	** R^2=99.5%

**Significant at the 1% level.

TABLE 7

FUNCTIONAL RELATIONSHIPS FOR $(\frac{V_o}{V_w})$ AND THE RECIPROCAL OF FROUDE NUMBER $(\frac{\sqrt{gl}}{V_w})$

Type of Ridge Vent	Equation	Coefficient of Determination
I Open	$\frac{V_o}{V_w} = 0.32 + \frac{1.08}{V_w}\sqrt{gl}$	** $R^2 = 95.2\%$
II Baffled	$\frac{V_o}{V_w} = 0.57 + \frac{1.68}{V_w}\sqrt{gl}$	** $R^2 = 78.1\%$
III Overlapped	$\frac{V_o}{V_w} = 0.26 + \frac{0.84}{V_w}\sqrt{gl}$	** $R^2 = 98.5\%$
IV Covered	$\frac{V_o}{V_w} = 0.07 + \frac{0.84}{V_w}\sqrt{gl}$	** $R^2 = 92.8\%$

**Significant at the 1% level.

The coefficients of determination for the open, baffled, overlapped and covered ridge vents based on the reciprocal of Froude number were 95.2, 78.1, 98.5 and 92.8%, respectively. Using the reciprocal of Reynolds number the R^2 values were 94.8, 98.2, 98.8 and 95.5% for the open, baffled, overlapped and covered ridge vents, respectively.

The standard errors of estimate based on the reciprocal of Reynolds number were 0.063, 0.053, 0.054 and 0.026 for the open, baffled, overlapped and covered ridge vents, respectively.

For all but one ridge vent the equations based on the reciprocal of Reynolds number predicted a greater amount of variation in the dependent pi term than did the reciprocal of Froude number. In that one exception there was very little difference in the amount of variation predicted. Therefore, the relationship based on one over Reynolds number was selected to predict the ratio of outlet velocity to inlet velocity for the model. The improvement in prediction based on the reciprocal of Reynolds number as compared to the reciprocal of Froude number can be explained in part by the variation in Reynolds number with air density and wind velocity, whereas Froude number only varies with wind velocity. Reynolds number also varies with viscosity, but the range of viscosity (0.0128 to 0.0126 16/sec°ft) was negligible for this study.

A plot of the linear regression equations based on the reciprocal of Reynolds number (Table 6) is shown in Figure 9.

At wind velocities up to 15.83 ft/sec the ratio of V_o to V_w was highest for the open ridge vent, while the ratio of V_o to V_w was least for the covered ridge vent at wind velocities up to 20.58 ft/sec. At zero wind velocity the outlet velocities for the open, baffled, overlapped and covered ridge vents were 1.75, 1.40, 1.00 and 0.75 ft/sec, respectively, while at a wind velocity of 8.0 ft/sec they were, respectively, 3.50, 2.50, 3.00 and 1.15 ft/sec. Analyses of covariance revealed highly significant differences in slopes and lines for the four ridge vents. However, it should be noted (Figure 9) that the ratio of outlet velocity to wind velocity increased for all ridge vents as the reciprocal of Reynolds number increased. Individual comparisons of the most similar looking

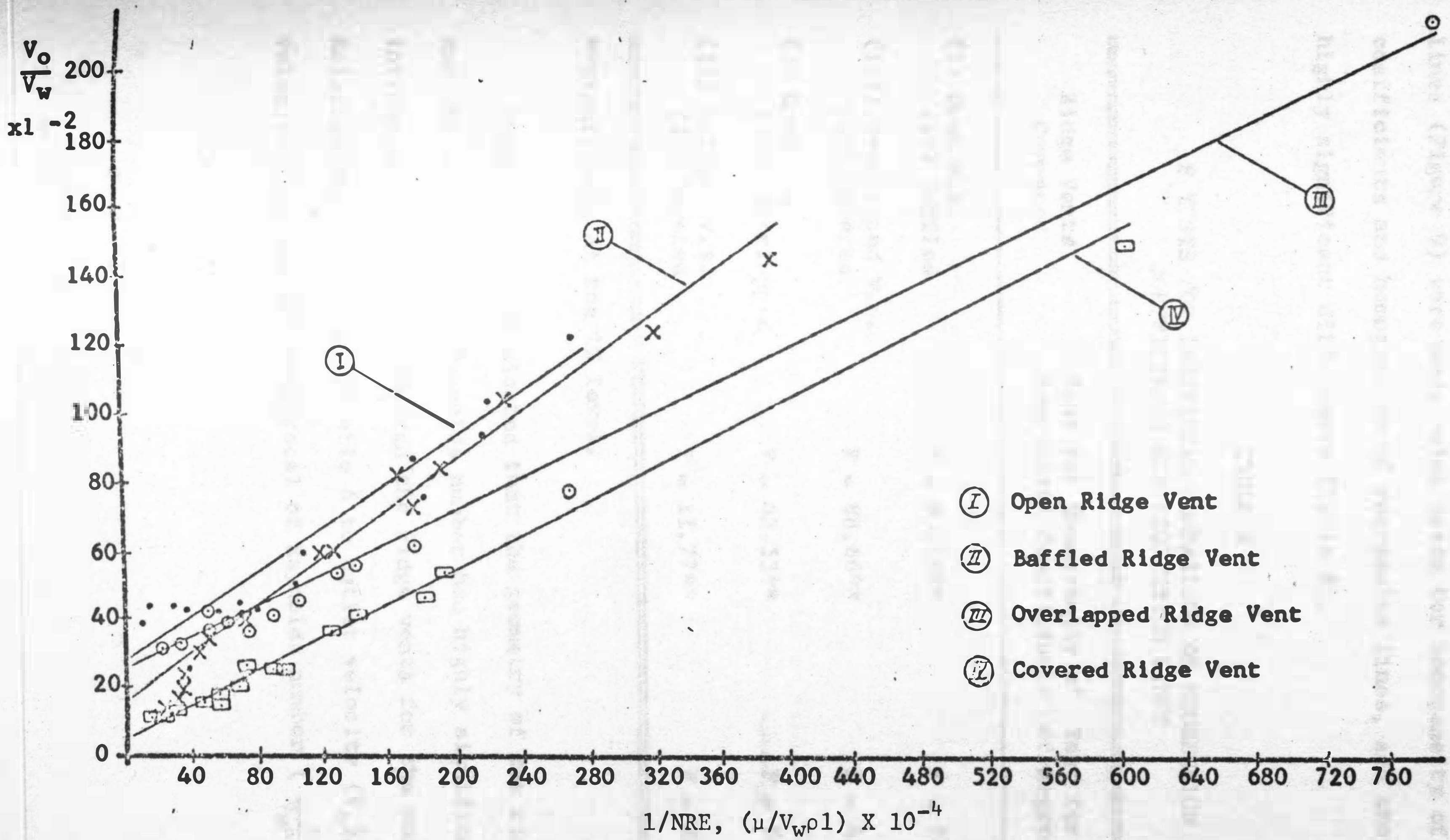

Figure 9. Linear Regression Equations Based on the Reciprocal of Reynolds Number.

lines (Figure 9) were made using tests for homogeneity of regression coefficients and homogeneity of regression lines, and they revealed highly significant differences (Table 8).

TABLE 8

F TESTS FOR INDIVIDUAL COMPARISON OF REGRESSION COEFFICIENTS AND REGRESSION LINES

Ridge Vents Compared	Test for Homogeneity of Regression Coefficients	Test for Homogeneity of Regression Lines
(I) Open v.s. (II) Baffled	F = 83.96**	F = 86.30**
(III) Overlapped v.s. (IV) Covered	F = 88.66**	F = 45.00**
(I) Open v.s. (III) Overlapped	F = 42.53**	F = 56.32**
(II) Baffled v.s. (IV) Covered	F = 11.77**	F = 275.70**

**Significant at the 1% level.

These analyses indicated that the geometry of the ridge vents and the reciprocal of Reynolds number had highly significant influences on air flow through the ridge vents for the model system. Relating the equations in Table 6 to outlet velocity (V_o), wind velocity (V_w) and the reciprocal of Reynolds number ($\frac{\mu}{V_w \rho l}$), the prediction equations for air flow through the ridge vents become

Open Ridge Vent	$V_o = 32.28 \frac{\mu}{\rho l} + 0.26\ V_w$
Baffled Ridge Vent	$V_o = 36.55 \frac{\mu}{\rho l} + 0.12\ V_w$
Overlapped Ridge Vent	$V_o = 24.78 \frac{\mu}{\rho l} + 0.22\ V_w$
Covered Ridge Vent	$V_o = 24.03 \frac{\mu}{\rho l} + 0.05\ V_w$

Polynomial regression analyses relating temperature differences to wind velocities from 0 to 20.58 ft/sec revealed the relationships included in Table 9 and shown in Figure 10. Temperature differences ranged from 28.7° F for the overlapped to 21.3° F for the covered

TABLE 9

RELATIONSHIPS BETWEEN TEMPERATURE DIFFERENCE (Δt), °F, AND WIND VELOCITY (V_W), FT/SEC, FOR THE FOUR RIDGE VENTS

Type of Ridge Vent	Equation	Coefficient of Determination
I Open	Cubic $\Delta t=26.0+0.33V_W-0.230V_W^2+0.0096V_W^3$	**$R^2=97.4\%$
II Baffled	Cubic $\Delta t=23.8-0.68V_W-0.135V_W^2+0.0075V_W^3$	**$R^2=99.3\%$
III Overlapped	Quadratic $\Delta t=29.0-2.24V_W+0.055V_W^2$	* $R^2=90.3\%$
IV Covered	Quadratic $\Delta t=23.4-1.42V_W+0.028V_W^2$	+ $R^2=90.5\%$

+ Significant at the 10% level.

* Significant at the 5% level.

**Significant at the 1% level.

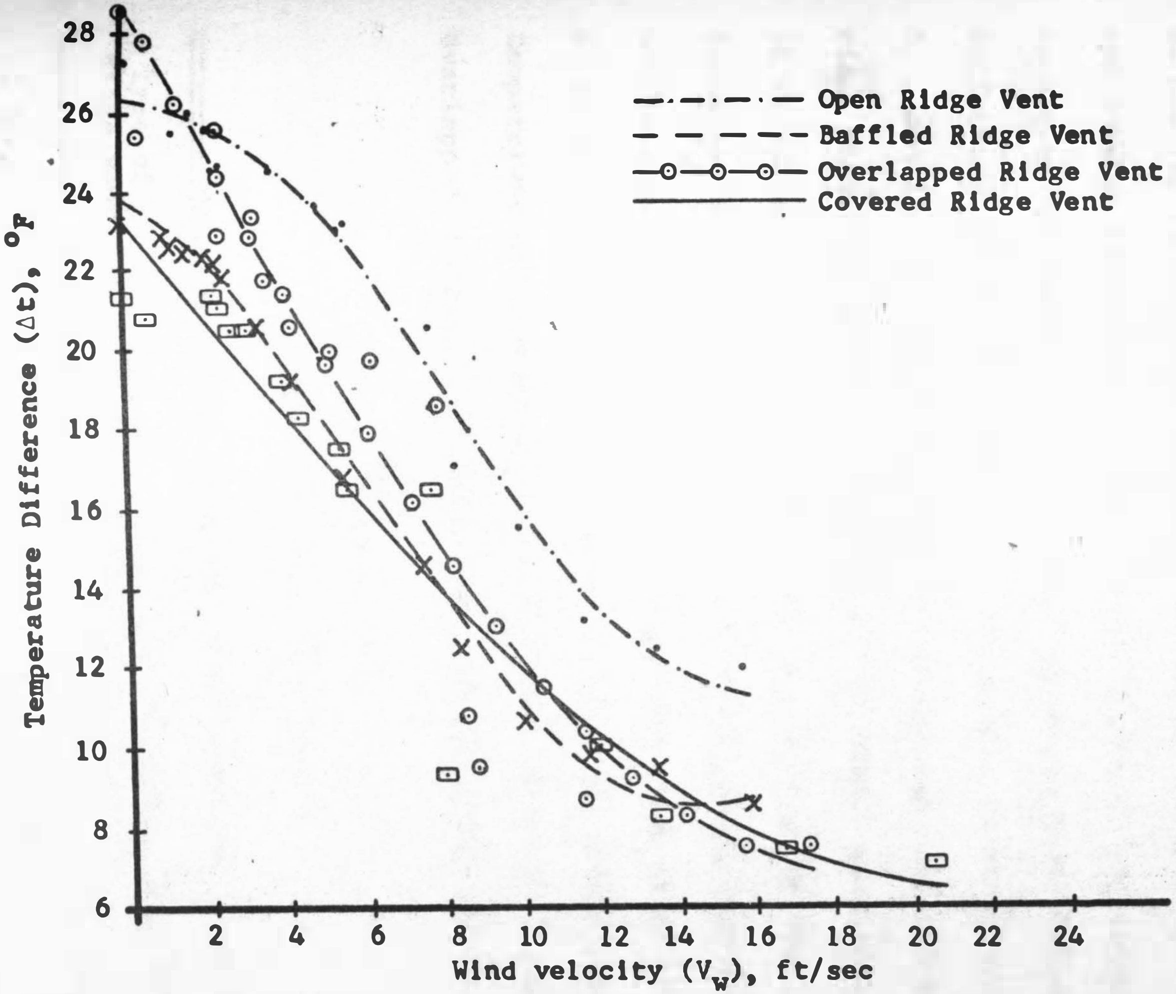

Figure 10. The Effect of Wind Velocity on Temperature Difference, Second Series of Tests.

ridge vent at zero wind velocity. At a wind velocity of 12 ft/sec the temperature differences were 13.2, 9.9, 8.9 and 10.0° F for the open, baffled, overlapped and covered ridge vents, respectively.

Analyses of variance showed that the equations were significant at the 1%, 1%, 5%, and 10% level for the open, baffled, overlapped and covered ridge vents, respectively. Using the equations developed to predict the effect of wind velocity on temperature difference (Table 9), temperature differences at three wind velocities--0, 5 and 10 miles per hour--were calculated for the four types of ridge vents (Table 10). Temperature differences ranged from 11.6° F at 10 miles per hour to 26.0° F with no wind for the open ridge vent. Predicted temperature differences at zero wind velocity for the open, baffled, overlapped and covered ridge vents, respectively, were 26.0, 23.8, 29.0 and 23.4° F. At a wind velocity of 10 miles per hour the temperature differences were 7.8, 8.5, 8.6 and 11.6° F for the overlapped, baffled, covered and open ridge vents, respectively.

TABLE 10

PREDICTED MODEL TEMPERATURE DIFFERENCES (Δt), °F

Type of Ridge Vent	Wind Velocity (miles per hour)		
	0	5	10
I Open	26.0	19.8	11.6
II Baffled	23.8	14.5	8.5
III Overlapped	29.0	15.5	7.8
IV Covered	23.4	15.0	8.6

The largest temperature difference was noted for the open ridge vent at all but zero wind velocity. It was also noted that the open ridge vent had the highest rate of air flow through the ridge vent. This combination was contrary to what was expected and indicates non-uniform ventilation distribution and areas of turbulence in the model building. Evidence of non-uniform ventilation distribution was also noted when temperature profiles in the model were evaluated. Studies using smoke candles revealed evidence of air flow through the eave inlet, along the underside of the roof and out the ridge vent. However, no definite patterns were established for the different types of ridge vents.

Since the pi terms associated with the fluid properties varied simultaneously in this study and there was distortion present, no conclusive test could be performed relating air flow in the model to air flow in the prototype. The effects of distortion need to be further analyzed by studying models of different sizes, as stated by Murphy (16), or the obtained pi terms need to be verified on a prototype to determine the functional relationships of the pi terms for the prototype. However, this does not invalidate the results determined for the model.

CONCLUSIONS

The following conclusions were indicated by this study:

1. Ridge vent design has a highly significant effect on outlet velocity.
2. Linear relationships developed between the reciprocal of Reynolds number predicted 94.8, 98.2, 98.8 and 99.5% of the variation in the ratio of outlet velocity to wind velocity, respectively, for open, baffled, overlapped and covered ridge vents. The standard errors of estimate for the open, baffled, overlapped and covered ridge vents were 0.063, 0.058, 0.054 and 0.026, respectively.
3. Linear relationships predicting the ratio of outlet velocity to wind velocity from the reciprocal of Froude number gave highly significant results, but generally predicted less variation than did the relationships based on one over Reynolds number.
4. The open ridge vent had the greatest air flow and the covered ridge vent the least air flow at corresponding wind velocities.
5. Temperature difference was greatest for the open type ridge vent at all but zero wind velocity.

6. Third degree polynomial regression equations based on wind velocity predicted 97.4 and 99.3% of the variation in temperature difference between inside and outside air, for the open and baffled ridge vents, respectively. Quadratic equations based on wind velocity accounted for 90.3 and 90.5%, respectively, of the variation in temperature difference for the overlapped and covered ridge vents.
7. The simultaneous variation of several pi terms associated with the fluid properties, prevented obtaining functional relationships between air flow in the model and in the prototype. However, this does not invalidate the results determined for the model.

SUMMARY

The trend in beef cattle production is toward increased use of confinement buildings to improve environmental conditions, better control pollution and facilitate farm labor. However, design information for adequate ventilation systems is often lacking. This is especially true concerning design of ridge vents for open front confinement buildings used under widely varying climatic conditions. Therefore, a model study of the effect of ridge vent design on ventilation characteristics was conducted.

Employing the principles of similitude, 14 dimensionless groups (pi terms) were formed, describing the fluid properties and the building geometry of a model of an open front confinement building. Two series of tests involving 119 observations on the effects of ridge vent design and wind velocity on outlet velocity and difference between inside and outside air temperature were conducted during the fall of 1971. Statistical analyses of the pi terms were performed using regression analyses, analyses of variance and analyses of covariance.

The results indicated that ridge vent design has a highly significant effect on outlet velocity and that the ratio of outlet velocity to wind velocity was linearly related to the reciprocal of Reynolds number. Standard error of estimate and coefficient of determination for the open, baffled, overlapped and covered ridge

vents were, respectively, 0.063 and 94.8%, 0.058 and 98.2%, 0.054 and 98.8%, and 0.026 and 99.5%. Linear relationships predicting the ratio of outlet velocity to wind velocity based on the reciprocal of Froude number generally predicted less variation in the dependent pi term than did the reciprocal of Reynolds number.

Temperature differences between inside and outside air were related to wind velocity using a polynomial regression analyses, and indicated cubic relationships for the open and baffled ridge vents and quadratic relationships for the overlapped and covered ridge vents. Simultaneous variation of the pi terms associated with the fluid properties prevented obtaining functional relationships between air flow in the model and in a prototype.

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APPENDICES

APPENDIX A

LIST OF SYMBOLS

LIST OF SYMBOLS

a, = Intercepts in regression equations

b_i, b_i' = Sample partial regression coefficient for the i^{th} variable

C, C' = Coefficients in the mathematical models

c = Specific heat of building materials, Btu/lb ^{o}F.

g = Acceleration of gravity, ft/sec^2

h = Building height, ft.

k = Thermal conductivity of building materials, Btu-in/hr·ft^2·^{o}F.

l = Building length, ft.

l_o = Length of ridge vent outlet, ft.

m = Subscript, designates the model system

N = Animal density, lb/ft^3

N_{GR} = Grashof number

N_{FR} = Froude number

N_{PR} = Prantl number

N_{RE} = Reynolds number

n = Geometric length scale

Q = Heat production per hour, Btu/hr.

q = Total animal heat production, Btu/lb·hr.

R = Regression coefficient

r = Moisture content of inside air

s = Rise of roof, ft.

Δt = Temperature difference (inside-outside), ^{o}F.

V_o = Velocity of outlet air, ft/sec.

V_w = Wind velocity, ft/sec.

w = Building width, ft.

W_o = Width of ridge vent outlet, ft.

X = Independent variable

Y = Dependent variable

α_i = Distortion factor for the i^{th} independent pi term

β = Coefficient of thermal expansion, $^{o}F^{-1}$

δ = Distortion factor for the dependent pi term

μ = Population mean in the statistical analysis

μ = Dynamic viscosity of the air, lb/sec•ft.

π_i = i^{th} pi term (dimensionless group)

ρ = Air density, lb/ft^3

APPENDIX B

OUTLET AND WIND VELOCITY, OUTSIDE AND INSIDE TEMPERATURE, AND OUTSIDE AND INSIDE DEW POINT TEMPERATURE

TABLE 1

OPEN RIDGE VENT, SEPTEMBER 28, 1971

Outlet Velocity V_o(ft/sec)	Wind Velocity V_w(ft/sec)	Avg. Outside Temp t_o(^oF)	Avg. Inside Temp t_i(^oF)	Outside Dew Point Temp t_{id}(^oF)	Inside Dew Point Temp t_{od}(^oF)
1.75	0.0	81.5	108.7	52.0	70.0
1.80	1.47	82.0	107.4	52.0	70.0
1.80	1.87	82.7	108.6	52.0	70.0
1.90	2.17	82.4	108.0	52.0	70.0
1.70	2.23	83.0	107.4	52.0	70.0
2.00	2.30	82.3	106.8	52.0	70.0
2.00	3.95	82.0	106.5	52.0	67.0
2.20	5.02	81.7	105.4	53.0	67.0
2.25	5.40	81.6	104.6	53.0	67.0
2.50	5.62	80.8	103.9	53.0	67.0
3.50	7.92	80.8	101.3	53.0	66.0
3.50	8.33	80.8	97.6	53.0	65.0
4.25	10.00	80.9	96.4	53.0	62.0
5.00	11.67	80.7	93.9	54.0	60.0
5.75	13.33	80.2	92.7	54.0	60.0
6.25	15.83	80.3	94.3	54.0	60.0

TABLE 2

BAFFLED RIDGE VENT, SEPTEMBER 28, 1971

Outlet Velocity V_o(ft/sec)	Wind Velocity V_w(ft/sec)	Avg. Outside Temp t_o(°F)	Avg. Inside Temp t_i(°F)	Outside Dew Point Temp t_{od}(°F)	Inside Dew Point Temp t_{id}(°F)
1.40	0.0	83.5	103.6	54.0	74.0
1.50	1.03	82.2	105.0	54.0	74.0
1.55	1.25	82.0	104.6	54.0	74.0
1.80	1.72	81.9	104.4	54.0	73.0
1.80	2.10	82.0	104.3	54.0	73.0
1.75	2.30	81.4	103.5	54.0	73.0
2.00	2.40	81.2	103.0	54.0	73.0
2.00	3.33	81.2	101.7	54.0	73.0
2.00	4.17	80.8	99.9	54.0	71.0
2.20	5.45	80.4	97.2	54.0	68.0
2.50	7.33	80.1	94.6	54.0	66.0
2.50	8.33	80.2	92.6	54.0	65.0
2.25	10.00	80.3	91.0	54.0	64.0
2.00	11.67	80.5	90.4	54.0	64.0
2.25	13.33	79.8	89.3	54.0	62.0
2.50	15.83	79.7	88.3	54.0	60.0

TABLE 3

OVERLAPPED RIDGE VENT, OCTOBER 17, 1971

Outlet Velocity V_o(ft/sec)	Wind Velocity V_w(ft/sec)	Avg. Outside Temp t_o(^oF)	Avg. Inside Temp t_i(^oF)	Outside Dew Point Temp t_{od}(^oF)	Inside Dew Point Temp t_{id}(^oF)
1.00	0.0	79.8	108.5	64.0	78.0
1.15	0.53	81.7	107.0	64.0	78.0
1.15	1.47	81.4	107.7	64.0	78.0
1.40	2.25	81.7	107.3	64.0	78.0
1.55	2.67	81.4	104.3	64.0	78.0
1.75	3.17	81.0	104.4	64.0	78.0
1.75	3.83	81.2	102.9	64.0	78.0
1.75	4.33	80.9	101.5	64.0	78.0
2.00	5.20	80.9	100.9	64.0	78.0
2.50	6.33	80.7	100.4	64.0	78.0
3.00	8.00	80.3	98.9	64.0	78.0
3.00	8.33	80.2	91.0	64.0	74.0
3.50	8.83	80.1	89.7	64.0	72.0
3.75	11.67	80.3	89.2	64.0	70.0
5.50	17.43	80.1	87.7	64.0	69.0

TABLE 4

COVERED RIDGE VENT, OCTOBER 17, 1971

Outlet Velocity V_o(ft/sec)	Wind Velocity V_w(ft/sec)	Avg. Outside Temp t_o(^oF)	Avg. Inside Temp t_i(^oF)	Outside Dew Point Temp t_{od}(^oF)	Inside Dew Point Temp t_{id}(^oF)
0.75	0.0	82.1	101.4	64.0	83.0
1.00	0.67	81.1	102.0	64.0	82.0
1.15	2.08	80.8	102.3	64.0	85.0
1.00	2.23	81.0	102.2	64.0	84.0
1.15	2.83	81.0	101.4	64.0	84.0
1.20	3.17	80.8	101.4	64.0	84.0
1.00	4.08	80.9	100.0	64.0	84.0
1.00	4.25	80.8	98.9	64.0	84.0
1.25	5.42	80.5	98.4	64.0	84.0
1.15	5.75	80.4	97.0	64.0	82.0
1.40	7.92	80.3	96.9	64.0	80.0
1.15	8.00	80.4	89.7	64.0	73.0
1.75	11.92	80.2	90.2	64.0	73.0
1.75	13.67	80.2	88.4	64.0	72.0
2.00	16.67	80.2	87.5	64.0	70.0
2.40	20.58	80.1	87.2	64.0	70.0

APPENDIX C

STATISTICAL ANALYSIS

TABLE 5

ANALYSIS OF COVARIANCE FOR TEST OF HOMOGENEITY OF REGRESSION COEFFICIENTS AND REGRESSION LINES

Ridge Vent	df	Total SS	Reduction in SS Due to	df	Residual SS
I Open	14	0.977	0.926	13	0.051
II Baffled	14	2.404	2.360	13	0.044
III Overlapped	13	2.943	2.907	12	0.036
IV Covered	14	1.689	1.680	13	0.009
Residuals from indiv. regressions		8.013	7.873	51	0.140
Totals for single regressions		8.780	7.653	54	1.127
Difference for homogeneity				3	0.987

Test for homogeneity of regression lines: $F(3.51) = \frac{0.987/3}{0.140/51} = 119.85^{**}$

Test for homogeneity of regression coefficients: $F(3.51) = \frac{.220/3}{0.140/51} = 26.71^{**}$

**Significant at the 1% level.

TABLE 6

INDIVIDUAL COMPARISON OF REGRESSION LINES AND REGRESSION COEFFICIENTS FOR BAFFLED AND COVERED RIDGE VENTS

Ridge Vent	Total SS	Reduction in SS Due to	df	Residual SS
II Baffled	2.404	2.360	13	.044
IV Covered	1.689	1.680	13	.009
Residuals from indiv. regressions	4.093	4.040	26	.053
Totals for one regression (II+IV)	4.631	4.016		.615
Difference for homogeneity			1	.562

Test for homogeneity of regression lines: $F(1,24) = \frac{.562}{.053/26} = 275.70^{**}$

Test for homogeneity of regression coefficients: $F(1,24) = \frac{4.040-4.016}{.053/26} = 11.77^{**}$

**Significant at the 1% level.

TABLE 7

INDIVIDUAL COMPARISON OF REGRESSION LINES AND REGRESSION COEFFICIENTS FOR OPEN AND OVERLAPPED RIDGE VENTS

Ridge Vent	Total SS	Reduction in SS Due to	df	Residual SS
I Open	.977	.926	13	.051
III Overlapped	2.943	2.907	12	.036
Residual from indiv. regressions	3.920	3.833	25	.087
Totals for single regression (I+III)	3.923	3.685		.283
Difference for homogeneity			1	.196

Test for homogeneity of regression lines: $F(1.25) = \dfrac{.196}{.087/25} = 56.32^{**}$

Test for homogeneity of regression coefficients: $F(1.25) = \dfrac{3.833-3.685}{.087/25} = 42.53^{**}$

**Significant at the 1% level.

TABLE 8

INDIVIDUAL COMPARISON OF REGRESSION LINES AND REGRESSION COEFFICIENTS FOR OPEN AND BAFFLED RIDGE VENTS

Ridge Vent	Total SS	Reduction in SS Due to	df	Residual SS
I Open	.977	.926	13	.051
II Baffled	2.404	2.360	13	.044
Residual from indiv. regressions	3.381	3.286	26	.095
Total for one regression (I+II)	3.386	2.976		.410
Difference for homogeneity			1	.315

Test for homogeneity of regression lines: $F_{(1.26)} = \frac{.315}{.095/26} = 86.30^{**}$

Test for homogeneity of regression coefficients: $F_{(1.26)} = \frac{3.286-2.976}{.095/26} = 83.96^{**}$

**Significant at the 1% level.

TABLE 9

INDIVIDUAL COMPARISON OF REGRESSION LINES AND REGRESSION COEFFICIENTS FOR OVERLAPPED AND COVERED RIDGE VENTS

Ridge Vents	Total SS	Reduction in SS Due to	df	Residual SS
III Overlapped	2.943	2.907	12	.036
IV Covered	1.689	1.680	13	.009
Residual from indiv. regressions	4.632	4.587	25	.045
Total for one regression (III+IV)	4.915	4.747		.126
Difference for homogeneity			1	.081

Test for homogeneity of regression lines: $F_{(1.25)} = \frac{.081}{.045/25} = 45.00^{**}$

Test for homogeneity of regression coefficients: $F_{(1.25)} = \frac{4.747-4.587}{0.045/25} = 88.66^{**}$

**Significant at the 1% level.