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Design of environmental control systems for livestock

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INTRODUCTION

Production of livestock is one of the mainstays of Iowa agriculture. According to the Iowa Annual Farm Census 16,723,000 hogs, 3,953,000 grain fed cattle and 647,000 sheep were marketed by Iowa farmers in 1967. In 1968 the number had risen to 17,329,000 hogs, 4,165,000 grain fed cattle and the number of sheep and lambs marketed had dropped to 600,000.

A survey taken by Wallaces Farmer magazine indicated that 5.5% of the respondents had raised their hogs in environmentally controlled buildings. It is estimated that as many as 50% of the pigs marketed were farrowed in environmentally controlled buildings.

The number of grain fed cattle that have been raised in environmentally controlled buildings has been limited. The author is aware of three producers in Iowa who have such facilities. In addition, there are approximately 15 producers who use open front confinement facilities for cattle feeding.

With the confinement of animals, there has been a significant increase in the need for ventilation systems to maintain an environment that is conducive to rapid growth and high feed efficiency. Many Extension publications have been written to help producers get a satisfactory ventilating system. There are several commercially designed ventilating systems.
available, however, there is an extremely wide variation in prices and performances between the systems. This is understandable since there are many differences of opinion among design engineers and researchers as to what constitutes an acceptable environment.

Many persons have felt that the animal environment needed to be controlled within a very narrow temperature range much as that required for humans. As a result their designs have called for controlling temperature within 2°F to 5°F for all probable extreme temperatures. Others have felt that temperatures varying as much as 30°F did not significantly affect feed efficiency and rate of gain as long as diurnal variation was less than 20°F. The latter was based on the premise that animals have a large capacity to adjust to their habitat and this would reduce the requirements for close control of the ventilating system. There is considerable evidence to support this contention.

The Extension Agricultural Engineer is faced with the task of evaluating the conflicting opinions and differences in research results and then recommending a ventilation system which will satisfy the individual producer. Frequently the producer will feel that the ventilation system is not acceptable because of odors, moisture buildup during cold weather,
excessive temperatures during hot weather and occasionally because the animals are not adapted to close confinement.

Most designs call for moisture control during cold weather with supplemental heat to maintain acceptable temperatures and a general disregard for the dissipation of odors. With building temperatures near 50°F acceptable odor control has usually resulted but if temperatures rose to or exceeded 70°F, odors were frequently considered intolerable by the operator. At outside temperatures above 50°F, temperature control has been used as the criterion for ventilation design, with resultant dust problems as relative humidities below 50% are frequently encountered. These problems create ill will between consumers, designers and equipment producers.

To help train Agricultural Engineers to better interpret research information, this dissertation will review pertinent literature, provide guidelines for design and develop a means of evaluating weather records to provide a basis for making judgements about design limits. Heat and moisture balance equations will be refined to better reflect actual operating conditions. As information on odors becomes available, it will become possible to design specifically for odor control. The author will outline design procedures that may be used when sufficient information is available.
OBJECTIVES OF THESIS

1. To review research relating confinement production of animals to environmental measures with particular emphasis on:
   A. the relationship of temperature and relative humidity to production and breeding efficiency, and
   B. the relationship of odor and dust to operator comfort and animal production efficiency.

2. To determine the probability of occurrence of climatic events which will adversely affect production efficiency with emphasis on swine production.

3. To organize design data for environmental control systems based on:
   A. the heat balance equation,
   B. the moisture balance equation and
   C. odor and dust control.

4. To provide guidelines for use of the equilibrium design equations.

5. To provide guidelines for objective judgements relating to environmental needs and conveniences.
EFFECT OF ENVIRONMENT ON LIVESTOCK PRODUCTION

An Overview and History of Research

The environmental design engineer is confronted with many decisions in attempting to control the environment of a confinement animal production building. He is called upon to meet the needs of the animal in regard to temperature, air currents, relative humidity, odor, light, sound, dust and other atmospheric contaminants. This may need to be done for an animal which is not adapted to confinement situations with resulting stress. When placed under stress, the animal may not perform as expected, have unsettled toilet habits which influence atmospheric moisture and pen cleanliness, become unsociable (fight) thereby affecting other animals and atmospheric dust concentrations and in the extreme situation he may even die. The engineer then is faced with the question of whether the environmental control is responsible for the problems.

Much of the research is based on controlling temperature and relative humidity for animal production. For some species odors and dust have been studied. Sight and sound effects have been approached by casual observation rather than scientific study. Limited information is available on the effects of air currents, even though many practicing engineers have
observed the effects of air movement. These movements are occasionally created to help reduce heat stress while at other times efforts are made to reduce air movement to prevent cold stress.

Many reasons exist for the abundance of information in some areas and the lack of information in others. Some insight into the problem may be gained by observing the relative difficulty of measuring the variables involved and their proportioned effects. For example, temperature is quite easy to measure and control. If temperature is maintained within proper limits, air movements and relative humidity become less significant factors.

No standards for odor level have been agreed upon. Recent developments in sensitive detecting and measuring equipment have made quantitative investigations possible, however, odor remains highly subjective. There is a wide discrepancy in observer response. Prolonged exposure to an odor desensitizes the observer to it. Some undesirable gases may not be detectable as odors. Gas toxicity may become an important design factor. Previously odors have been neglected in design, considered tolerable if temperature and relative humidity were controlled.

Dust effects and origin have been studied most extensively in poulty production. Dust has been a problem in this
industry particularly when the relative humidity of poultry buildings is low as normally occurs during summer and fall conditions. The birds tend to lose skin and feathers which become airborne, as does finely ground feed. Use of litter has increased the problem especially as the birds become more active with increased size. Very little is reported in the literature on the problem of airborne contaminants in other livestock units.

Light and sound levels have been controlled to suit the operator rather than the animals in an environmentally controlled unit. Some work has been done to study the effects of light in poultry houses with the intent of producing larger numbers of eggs by upsetting the daily clock and making the hen operate on a short day rather than the normal 24 hour day. Sound levels have been studied by casual observation which indicates that an animal becomes deaf to continuous, steady state noise unless the frequency and intensity are such that pain results. The result is that no response is attributed to sound even though some producers use changes in sound level to cause cows to become prepared for milking or call pigs to feeding equipment.
Swine Response to Environment

**Growing-finishing**

Bond (9), Hazen and Mangold (33), Bell, et al. (8), Mangold, et al. (55) have shown that most rapid gains occur when pigs are raised at temperatures between 60°F and 70°F. Feed efficiencies were also greatest in this temperature range. As temperatures were reduced, greater amounts of feed were consumed. When pigs were confined on concrete floors with wood shavings for bedding, Mangold (55) showed a reduction in feed efficiency of 0.0016 pounds of gain per pound of feed intake for every degree F in temperature drop below 57°F. Mentzer, et al. (59) found that abundant bedding provided sufficient insulation around the pig to reduce this effect, however the change in feed efficiency was not accurately measured. A 0.6 pounds of gain per pound of feed increase of bedded over unbedded open front sheds was noted.

At temperatures above 70°F, Mangold (55) noted a reduction in feed consumption with no significant reduction in feed efficiency. The feed consumption was reduced 0.005, 0.4 and 0.05 pounds per degree F increase above 72, 74 and 70°F for 30, 75 and 140 pound pigs respectively.

Bond, et al. (10) found no significant difference in daily gain between constant 70°F and a diurnal variation between 60°F
and 80°F. Feed conversion tests were too limited to check for significance although it appears that there was little difference. As the diurnal variation increased, daily gains showed significant decreases.

Kelly, et al. (45) found that providing a cooled slab beneath a shade had a variable benefit when compared to a wallow. The amount of heat transferred varied from 39 BTU/hr. sq. ft. to 61 BTU/hr. sq. ft. corresponding to slab temperatures of 65°F and 80°F. There appeared to be little relationship between heat loss to the slab and ambient air temperatures. Respiration rates and rectal temperatures remained lower for the pigs on the cooled slab as compared to those in a psychrometric chamber at ambient air temperatures of 105°F. It was also noted that all pigs lost weight at air temperatures of 105°F even though other methods of cooling were available.

Morrison, et al. (64, 65, 66) found that with air temperature change from 60°F to 85°F at 50°F dew point (relative humidity 70% to 30%) the 90-kg Duroc gilts were able to offset the decrease in sensible heat loss by doubling skin moisture loss and tripling lung loss by tripling the volume rate of respiration. At 85°F and increasing dew point from 50°F to 82°F (relative humidity 30% to 90%) skin moisture loss remained nearly constant. Lung moisture loss increased initially and then
decreased rapidly with a corresponding respiration volume increase, then decrease. It was also noted that relative humidity had little effect at temperatures below 70°F. Figure 1 compares rate of gain at various temperatures and relative humidities to that obtained at 72°F and 50% relative humidity.

Morrison, et al. (65) found wetting of the skin was beneficial in reducing skin surface temperature at temperatures between 80°F and 100°F and dew points between 48°F and 75°F. Sprinkling every 80 minutes was often enough for stress relief. Wetting of the skin was more desirable than fogging.

Roller and Goldman (76) found that 165 to 265 pound pigs could recover from short term (3 hours) exposure to an air temperature of 109°F which caused the rectal temperature to rise to 106°F and respiration rate to increase to 300 cycles per minute.

Animal body heat content was calculated as 0.83 X weight (pounds) X body temperature (degree F). Body heat content increased logarithmically during 3 hours exposure to 97°F. The body heat content approached 13,600 BTU at 97°F and 14,800 BTU after 3 hours exposure at 106°F. In both cases the body heat content was still increasing although at a lower rate for the 97°F air temperature. The capacity of the body to absorb heat may explain the ability of an animal to respond to an average
Figure 1. Ratio of weight gain for 150 lb. pigs at a given relative humidity and temperature to that at 72°F and 50% relative humidity. [Morrison (64)]
diurnal temperature rather than the high or low temperature to which the animal is exposed.

Gunnarson, et al. (26) found a significant difference in rates of gain when air velocity was varied at temperatures between 50°F and 70°F. With air velocities at 35 ft./min., gains were reduced by 0.50 lbs./day/pig when compared to pigs exposed to air velocities of 10 ft./min. No animal size was given in the reference cited. A linear multiple regression equation was calculated to relate average daily gain to temperatures and air velocity. The equations were:

\[
\begin{align*}
Y_{ADGM} &= 1.481 - 0.000533B - 0.287C + 0.0172D + 0.0024E \\
Y_{ADGF} &= 2.227 + 0.00192B - 0.0210C - 0.00882D + 0.00186E \\
Y_{FCCR} &= 3.643 + 0.0214B + 0.0124C - 0.0817D + 0.00945E
\end{align*}
\]

Where:

- \(Y_{ADGM}\) = average daily gain for male pigs in lb./day/pig,
- \(Y_{ADGF}\) = average daily gain for female pigs in lb./day/pig,
- \(Y_{FCCR}\) = feed conversion ratio in lb. feed/lb. gain,
- \(B\) = average liveweight,
- \(C\) = air velocity, ft./min.,
- \(D\) = air temperature degree F and
- \(E\) = floor temperature degree F.
These equations indicate that air velocity may be more important than air temperature in affecting average daily gain. The author questions the accuracy of these equations when temperatures are other than optimal for rapid growth. Mangold (55) indicates daily gain is reduced by temperatures above 75F while Gunnarson (26) shows a continued increase in gain with increasing temperature for male swine.

Beckett (7) and Morrison, et al. (66) discuss the use of "effective temperature" as a means of evaluating swine environments. This is an effort to relate the effect of relative humidity and temperature to the growth rate of swine. Beckett gives less credit to relative humidity as a factor affecting growth rate than does Morrison. Both agree that it is less important than the temperature-humidity index frequently suggested (THI = 0.55TDB + 0.2 TDP + 17.5) by the U.S. Weather Bureau for human comfort.

Where THI = temperature-humidity index,

\[ TDB = \text{temperature dry bulb} \]

\[ TDP = \text{temperature dew point}. \]

Beckett relates the relative humidity effect to the moisture removed by the lungs (Figure 2) while Morrison shows some effect from surface evaporation and is borne out by the daily gain data that he has collected \[ G(t,\phi)/G(72, 0.5) = .55 \].
Figure 2. Swine effective temperature as a function of air temperature and relative humidity. The curves are for a 150 lb. hog in a room with wall temperature equal to air temperature. Air velocity is from 20 to 30 feet per minute. [Beckett (7)]
Where \( G (t, \phi) \) = gain at any temperature, relative humidity combination and

\[ G (72, 0.5) = \text{gain at 72F and 50% relative humidity}. \]

Figure 1 shows the effect of temperature and relative humidity on the rate of gain as found by Morrison.

**Gases and odors**

Day, et al. (14) found that much of the odor in swine units was attached to moist-solid particles in the atmosphere. Gases that were detected as being associated with these airborne particles and odor were identified as carbon dioxide, hydrogen sulfide, methane and possibly ammonia. Carbon dioxide and methane are considered to be non-odorous but were included in the reference cited. No quantitative analysis was made to determine the amount of these gases present although the corrosive effects of sulfur compounds were noted on the building structure.

Merkel, et al. (60, 62) identified eighteen separate gases in the swine atmosphere. Many were identified as mercaptens, amines or aldehydes. The odors associated with swine manure were listed as amines and sulfur compounds. Again no quantitative analysis was done nor were adverse effects on the animals noted even though metal corrosion took place.
Miner and Hazen (63) showed the presence of ammonia to be below published human threshold levels (unable to be detected by humans). They further detected methylamine, ethylamine and triethylamine as part of the swine house odor complex. It was concluded that either ammonia is not a significant component of swine house odor or else the odors produced have an additive effect enabling detection at less than threshold levels.

Stombaugh, et al. (82) found that an ammonia level of 280 ppm was toxic. Levels below 150 ppm were not toxic, however, a highly significant relationship between ammonia level and reduced rates of gains were noted at 21°C (70°F). At levels below 50 ppm the pigs appeared to adjust to ammonia levels and only slightly reduced gains were noted.

Additional efforts have been made to determine the quantity of odorous gases present, however, at the time of this writing these results were not available.

**Adult Pigs**

Roller, et al. (77) showed a significant reduction in embryo survival as sows were exposed to higher temperatures (above 80°F). Only dry bulb temperatures appeared to affect ovulation rate and embryo survival while the wet bulb and wet bulb-dry bulb interaction had some effect on daily feed
consumption. They also indicated that heat stress becomes critical at dry bulb temperatures above 92°F unless some means of external cooling (sprinkling, wallows, etc.) is provided.

Merkel and Hazen (61) studied the uses of cooled air blowing onto the face of a lactating sow to increase comfort at 90°F ambient air temperatures. No statistically significant differences were noted in the feed consumption, however, visual inspection showed greater comfort for cooled sows and also that the litters gained more rapidly when the sows were breathing cooled air.

Mahoney, et al. (53) noted a significant decrease in viable embryos for gilts stressed in a heat chamber. A temperature of 105°F for 17 hours per day and a 90°F temperature for the remaining 7 hours was "excessive". Apparently sows were in danger of dying. When exposed to 102°F and 90°F for 17 and 7 hours respectively, either prebreeding or 1 to 15 days post breeding, there was a significant decrease in survival rate and smaller embryos.

**Beef Cattle Response to Environment**

Research information concerning desirable environmental characteristics for optimum performance of beef animals is quite limited. Some of the reason for this is the practice of providing little or no shelter for beef animals except for
calving during severe weather. In addition, these animals will shed and regrow a protective hair coat as the seasons change, therefore requiring less environmental control. Personal observations on mechanically ventilated confined beef feeding operations indicate that during the summer hot spots in the structure are common; also, when the outside temperature drops below 10°F, fogging occurs where no supplemental heat is provided.

Ittner, et al. (42) have shown that shade for beef cattle improves performance in the feedlot. Morrison, et al. (67) also indicates that heat stress affects performance at temperatures above 70°F. High temperature stresses masked the effect of crowding, however, crowding stresses were evident at temperatures below 70°F when pen space was less than 40 sq. ft. per animal.

Hudek (41) notes a fog problem during cold weather as well as a nasal discharge and cough. Also noted was an increase in hydrogen sulfide when the manure pits under a slotted floor were agitated. Two animals died during agitation of the pits. Less than 5 ppm hydrogen sulfide in the air was noted during normal operation but increased to 250 ppm within 60 seconds of the onset of agitation.
Mendel, et al. (58) states that it is the average temperature which determines stress in beef cattle as long as the humidity is low. Feed intake was uniform throughout the day and night for continuously cooled animals, whereas animals exposed to high daytime temperatures did not begin eating until the peak heat of the day had passed. They found that cooling for 6 or 12 hours permitted weight gains only slightly below those for continuous cooling.

Dairy Cattle Response to Environment

Dairy cows have been sheltered from cold weather to prevent freezing of teats even though no production losses were evident with temperatures above 10°F for Holsteins and 30°F for Jerseys. Weirsma, et al. (89, 90, 91), Stott (83), Stott and Williams (84) and Welchert, et al. (86, 87) showed that at temperatures above 75°F breeding efficiency declined. Hahn and McQuigg (27) and Hahn and Osborne (28, 29) also showed milk production decline with temperatures above 75°F.

Wiersma and Hahn studied methods of relieving heat stress to provide more efficient milk production and higher breeding efficiencies. Breathing cooled air provided for some relief from heat stress even though complete relief was not afforded unless the entire space was cooled. The effect of high temperatures was noted almost immediately on milk production, but
had a delayed effect on breeding efficiency. A three-month lag was noted between heat stress and breeding problems.

Hahn, et al. (27, 28, 29, 30) related the probability of milk production losses to cost of providing heat stress relief. The temperature-humidity index (THI), defined earlier, was related to decrease in milk production above a THI of 70. The decrease in milk production was described as: 

\[ \text{MDEC} = -2.370 - 1.736\text{NL} + 0.0274(\text{NL}) \times (\text{THI}) \]

Where MDEC = decrease in milk production (lbs./cow) and NL = normal level (lb./cow).

A THI of 70 or greater was expected to occur for two to three weeks annually, with equipment and operating costs, base 1968, only the southern 1/3 of the United States could justify total cooling and then only for animals producing more than 70 lbs. of milk per day. This area would change as the price of milk and equipment costs varied.

Hodges and Farmer (35) were unable to detect significant differences in semen quality of dairy bulls by reducing radiant heat load under shade. Their evaluation was that the cooled roof either had no effect on semen quality or else that time lag obscured the effects.

Light (48) advised that veal calves be housed at temperatures between 55°F and 65°F at 3 days of age and this be lowered
to 45F and 50F for 3 week old calves. He also indicates a tendency toward detrimental effects if the relative humidity rises above 83% coupled with ammonia concentration greater than 40 ppm.

Poultry Response to Environment

Layers

Hobgood and Jasha (36) and Gleaves (22) found that maximum production was achieved at 57F. At temperatures between 45F and 75F egg production was at least 95% of the 57F level. Hobgood and Jasha (36) also noted that at temperatures above 70F hens began to lose weight and show an increase in respiration rate. Temperatures above 100F were considered lethal with many birds dying.

Howes, et al. (40) showed that heat stress was cumulative. Equal depression of egg production resulted from 64 days at 100F, 108 days at 90F, and 224 days at 80F. Periodic cooling to temperatures below 80F helped to reduce the depression in egg production. Short exposures to temperatures above 90F could not be completely compensated for by equivalent exposure time to temperatures between 50F and 65F.

Palmer and Clayton (71) demonstrated wind and temperature effects on layers. By subjecting them to wind at low temperatures, increased air motion affected the comfort of layers.
Above 95°F very little cooling was taking place with air velocities of 1600 fpm. At 85°F and 90°F panting was reduced with increased velocity of air. At 45°F with air velocities of 800 fpm there was a rapid body temperature drop indicating that the chicken was unable to maintain homeothermy.

Howes, et al. (37, 39) recommends that layers not be kept at temperatures below 55°F for extended periods and that for optimum feed efficiency they be kept between 60°F and 80°F.

Baker, et al. (4, 5) indicate that in addition to removal of respired moisture from the birds the manure, if left in the building, will dry from an initial moisture content of 77% (wet basis) to 35% (wet basis). This will increase the total water vapor production to 42 lbs. per 100 (3 lb.) hens per day. For summer, cooling is recommended to maintain temperatures within the range of 55°F to 85°F with relative humidity between 60 and 80%. Ammonia concentrations can be kept below damage levels at 70°F and 70% relative humidity with 1/3 cfm of ventilation air per pound liveweight.

Broilers

Howes and Grub (38) found that birds raised at 50°F to 60°F had the highest rate of gain, but feed efficiency was greatest
at 75°F to 85°F. Broilers raised in the higher temperatures had the highest dressing percentage, the least feathers, blood and digestive tracts. They produced less dust due to inactivity.

Griffin, et al. (24) found that chicks could be grown at cyclic temperatures, within limits, without seriously affecting overall performance except that those near market age did not adjust well to a sudden change from a low temperature weekly cycle to a higher temperature. Chicks raised with a daily cyclic temperature varying 30°F showed no significant difference in feed efficiency, rate of gain or mortality from those raised in a constant temperature equivalent to the mean of the cyclic temperature.

McCune, et al. (56, 57) found that in a ventilated broiler house insulation increased the amount of moisture removed over a comparable non-insulated house. Insulation was necessary to prevent freezing without supplemental heat in Eastern Texas. An increase in temperature also brought about an increase in overall feed efficiency. During the summer time it was possible to maintain inside temperatures 7°F cooler in houses with evaporative coolers than in similar houses using natural ventilation. No difference in feed efficiency was noted nor did there appear to be any effect on quality of birds produced. The average daytime temperature in the
uncooled houses during the test periods varied from 85F to 88F. At the same time the average daily maximum temperature was 90F to 93F. During the sixty day test period the maximum outside air temperature reached 97F with the average outside air temperature never exceeding 93F. On the second test the maximum outside air temperature reached 100F on three different days while the average outside air temperature remained below 95F on those same days. There were at least two days at lower temperatures between each peak which may have modified the temperature effect. No records on relative humidity were cited.

Dust and odor effects

Grub, et al. (25), Koon, et al. (46) and Koon, et al. (47) found that the dust collected on filters was composed of skin debris, feed and broken feathers. Koon (45, 47) indicates that for layers raised on litter, the dust production increases as the temperature rises from 60F to 70F then levels off between 70F and 80F and begins to decline at temperatures above 80F. All tests were made with a constant relative humidity of 60%. Grub (25) showed similar results for layers, however, with broilers raised on wire, dust production increased with age with no temperature effects visible. Broilers raised on litter increased dust production with age except that after
seven weeks of age at 90°F dust production began to decrease.

Grub also found that more dust was produced during hours of illumination than during hours of darkness. Litter materials were significantly different in the amount of dust produced as well as the age of the litter. Dust production from litter decreased as air moisture increased. Up to 80 mg dust per bird per day was observed for broilers at 6 weeks of age and 60 mg per bird per day for layers. It is expected that dust production is directly related to bird activity.

Wolfe, et al. (92) exposed turkeys to varying ammonia and dust levels to determine the effect on airsacculitis. Ammonia was varied between 0-8 ppm and 20-30 ppm while dust concentrations were between 0.1 - 0.4 mg/ft.³ and 0.6 - 1.0 mg/ft.³. No significant difference was noted in mortality between the various tests or interactions. However, the incidence of airsacculitis more than doubled between the low and high dust concentrations.

Ludington, et al. (52) studied the release of gases and odors from diluted and undiluted chicken manure. More ammonia was produced from the diluted manure than from the undiluted. Also the diluted manure indicated a higher level of bacteriological action, indicating that some bacteriological decomposition was taking place. Gases soluble in water were
absorbed in greater quantities in the diluted manure than in the undiluted manure, however, after a period of time the water becomes saturated and less of the water soluble gases are captured by the water.

Miscellaneous Information Pertaining To Animal Physiology and Ventilation

Hinkle (34) lists four areas in which work needs to be done to supply more accurate data for ventilation system design. There are heating and cooling load requirements, (2) system and equipment energy, (3) logic, the combination of instantaneous load calculation with weather data and building operation schedule, (4) field validation to refine and validate energy calculations. These data should provide information for dynamic solutions to ventilation design. A dynamic solution should be more realistic in terms of the actual animal environment moment to moment, than as the design by equilibrium method frequently used.

Phillips (74) prepared a computer program to size the supplemental heating equipment in poultry houses (Connecticut). He found that some minimum supplemental heat in the brooding house was required between the 169th day of the year and the 225th. The greatest amount of supplemental heat was required beginning with the 300th day of the year and continuing on
through the 40th day of the next year. The effect of space allowance is shown in Figure 3. With a constant insulation of $U = 0.046 \text{ BTU/hr. sq. ft.}$, the supplemental heat per brooding period varied from 2000 BTU/bird to 5000 BTU/bird as the ventilation rate varied from 0.4 to 1 air change per hour. This is shown in Figure 4.

Phillips and Bartok (75) studied the heat requirements for animal buildings. They suggest that 10 years of data be used in determining the weather conditions that are applicable. They propose that calculations be made for minimum weather conditions in winter as well as average conditions over the 10-year period. They indicate that the building has some capacity to serve as a heat sink to respond more nearly to the average of a diurnal temperature cycle rather than following the maximum and minimum outside condition. They suggest that solar radiation be considered particularly when calculating operating costs. More realistic information is needed to accurately determine the amount of free water that must be removed by evaporation from within agricultural buildings.
Figure 3. Effect of bird density on heating needs for various brooding period starting dates. [Phillips (74)]
Building size: 40' X 200' X 8'
Number of birds: 8000
Date started: Dec. 3
Insulation: $U = 0.046$
Brooding period: 63 days

Figure 4. Effect of infiltration on total heat required per bird during the coldest period of the year. [Phillips (74)]
Summary of Animal Response to Environment

All classes of animals respond to their environment by regulating feed intake, panting, exercise and adjusting water intake. In general the following responses can be expected from varying environmental conditions:

**High temperature**

- High temperatures reduce feed consumption, decrease breeding efficiency, milk production and rate of gain.

**Low temperature**

- Low temperatures reduce feed efficiency, increase feed consumption, promote huddling of pigs and poultry and reduce rate of gain.

**Air velocity**

- Increased air movement can be expected to reduce the adverse effects of high temperatures up to approximately 100°F, while magnifying the adverse effects of low temperatures.

**Relative humidity**

- Relative humidity below 50% increases dust production while above 80% fog may form during cold outside conditions and temperature effects are magnified at high temperatures.

**Dust**

- Dust has caused problems with heating and cooling equipment and controls, carries some of the odor associated with the specific animal being raised and caused or magnified diseases of the respiratory system.
Odors appear to be most objectionable to the manager although some gases have been shown to be toxic to the animals being produced. Ammonia and hydrogen sulfide have been shown to have detrimental effects. At 50 ppm or greater ammonia reduces rate of gain for swine and at 200 ppm is lethal. For turkeys 30 ppm increased disease. Other gases have not been studied in sufficient detail to determine their effects.

Recommendations for Control of Animal Environment

The engineer can design an environmental control system with some assurance that he will be able to meet the animal's requirements for temperature and relative humidity. He can be less confident in establishing dust tolerance levels until more information is available and he will need to estimate the tolerance to various gas and odor concentrations. From the available information the author has established the following guidelines by animal stage for all classes of livestock and poultry:

1. Maternity and nursery stage.

   Temperature: 80°F to 95°F. Decrease 5°F each week until conditions for growing-finishing stage are reached. Short term exposure to lower temperatures are not harmful as long as no freezing takes place.
Relative humidity: 50% to 80%

Dust: no visible dust.

Odors and other gases: below threshold levels for odors, oxygen at least 16%, carbon dioxide less than 1%.

In most cases it is desirable to keep the mother at a temperature lower than is required for the newborn. Zone heating or cooling may be used to satisfy this condition.

II. Growing-finishing stage.

Temperature: 50°F to 80°F.

Relative humidity: 50% to 80%.

Dust: 0.1 mg/ft.³.

Odors and other gases: less than 50 ppm for ammonia, less than 5 ppm for hydrogen sulfide, criteria for other objectionable gases unknown, oxygen at least 16% by volume, carbon dioxide less than 1% by volume.

III. Mature stage (breeding and producing animals).

Temperature: 35°F to 75°F.

Relative humidity: 50% to 80%.

Dust: 0.4 mg/ft.³.

Odors and other gases: less than 50 ppm for ammonia, less than 5 ppm for hydrogen sulfide, criteria for
objectionable gases unknown, oxygen at least 16% by volume, carbon dioxide less than 1% by volume.

The temperature ranges have been selected for rapid growth, prevention of death in newborn animals and tolerance to heat and cold of mature animals. The upper temperature limits would be expected to minimize the effects of relative humidity and air currents. The lower temperatures would prevent freezing of water pipes and troughs for mature animals. For newborn and growing animals the lower temperature limits would insure good feed efficiency and also minimize the chilling effect of air currents.

Relative humidity limits were selected to prevent condensation from forming on insulated walls and to help limit the amount of airborne dust present inside the facility, rather than because of any effect that has been shown on animal response within the temperature limits previously established. With temperatures in excess of those used for optimum production, the relative humidity levels will need to be lowered 15% for each degree increase in temperature above 80F.

The dust and odor levels are somewhat arbitrary and may need to be revised as new information becomes available. Some difficulty may be encountered in attempting to maintain low
dust levels at moderate temperatures (65°F to 75°F) with relative humidities near or below 50%. Based on Wolfe's work with turkeys a short term exposure (less than 2 days) to dust concentrations as high as 1.0 mg/ft. would not be expected to adversely affect performance. Longer periods of exposure may result in increased respiratory problems and as a result decrease performance (rate of gain, milk production or breeding efficiency). Information on the rate of dust production is necessary before this can be used as a controlling design parameter.

Upper limits for gas concentrations are recommended from measurements made in existing systems where no deleterious effects have been observed. Higher levels may be tolerable, however, present practices have been able to maintain concentrations at or below these levels. Before a design can be based on gas tolerances, more information needs to be available on the rate of production, the rate of diffusion and the ultimate distribution of the gases to be controlled.
PERFORMANCE OF ENVIRONMENTAL CONTROL SYSTEMS

Karhnanak and Aldrich (44) found that temperature variation in a farrow to finish building for swine was no more than 5°F despite a 50 degree range in outside temperatures when space heaters were used for maintaining comfortable temperatures at low outside temperatures. Ventilation rates were 30 cfm per sow and litter for cold weather and 600 cfm per litter, or approximately 80 cfm per pig, in warm weather. Relative humidity varied from 37% to 75%. Air velocities at floor level had to be reduced by closing the perimeter slot inlet in an exhaust type ventilation system. Odor was considered objectionable due to the large volume of air moving over the liquid manure storage area.

Schulte, et al. (79) found that whenever heavy vapors were introduced under slotted floors they began to rise through the slots. He concluded that some of the air introduced into the building was entering the pit and carrying the pit gases out at some other point.

Mangold (54) found that a high level of ventilation was required to keep odors below an objectionable level. It was speculated that more frequent cleaning would reduce the volume of air required to that required for moisture control. This reduction in air volume would result in a 25% reduction in air conditioning equipment.
Mangold (54) found that 1/3 ton of refrigeration capacity was required per pig during the summer to maintain an air space temperature of 60°F if odors and moisture were controlled. It was recommended that the fan be kept operating at all times to prevent stagnation and thermal stratification. The use of a mixing duct helped to reduce stratification and temperature variation. With a 20°F temperature differential between inside and outside air temperature a maximum of 3°F variation was experienced. With a 40°F differential, a 5°F temperature variation was experienced.

Esmay (18) recommends that in buildings less than 40 feet in width an exhaust type system with a continuous slot inlet on the opposite wall be used to provide positive cross flow ventilation. The slot should be one to two inches wide and draw the air in from the attic for winter operation. For summer operation it is recommended that the slot be six inches wide and the air be drawn directly from outside. The incoming air velocity through the slot inlet should be limited to 500 feet per minute or less.

Esmay suggests that axial flow or propeller type of fans be used for livestock ventilation systems. He cites their efficiency at low pressure differentials and the self-cleaning characteristic of narrow blades as the basis for this
recommendation. He recommends that shutters installed on fans be placed on the intake side of the fan to prevent the shutters from freezing. The shutter blades should be adjustable to increase fan efficiency.

Rollo, et al. (78) describes the use of porous curtain wall materials as air intake devices. Generally one or more walls are made of a porous material and the air flow is controlled by means of fans in a conventional "solid wall". The large surface area of the wall would be expected to supply ventilation without drafts by eliminating high velocity air currents. The advantage of this system was cited as a minimum of heat loss. With the use of baffle walls, the influence of wind direction and velocity would be eliminated. Most woven materials tested had from 80 to 150 cfm of air movement per square foot of surface area when exposed to a static pressure differential of 0.2 inches of water.

Dixon and Lampman (16) studied air patterns in environmentally controlled poultry houses. They found that it was possible to maintain temperatures with a 3 to 5 degree variation even though outside temperatures varied as much as 15°F. It was also possible to maintain a 3 degree temperature difference between floor and ceiling temperatures. Desirable air flow characteristics were obtained with exhaust fans
located in one wall and a slot inlet in the opposite wall. An adjustable baffle was used to prevent cold air from dropping directly onto the floor during cold weather operations or to permit the air to drop directly on the floor during warm weather operations. Air flow characteristics with these units are described in Figures 5, 6 and 7.

Pattie (72) showed that conductive heat loss could be eliminated by insuring that ventilation air entered by infiltration through porous insulation. He also found that porous permeable ceilings allowed moisture to escape even though air was moving in through the insulation. An overall heat transfer coefficient for the porous permeable ceiling was estimated at 0.08 BTU per hour per square foot degree F. The ratio of heat loss per pound of vapor loss was found to be 85% of the estimated heat loss per pound of vapor loss in a conventional ventilating system.

Pattie (73) discussed the use of permeable wall material to allow odorous gases and water vapor escape from the building. The advantage of this would be dissipation of water vapors without the heat loss associated with increased mechanical ventilation. At an animal density of 10 sq. ft. per 125 pound pig and 60F inside temperature, 70% relative humidity, outdoor conditions 10F, 100% relative humidity, it would
Figure 5. General pattern of air circulation with wall fan and slot inlet in opposite walls.
Figure 6. Enlargement of a portion of Figure 5 with inlet adjusted for cold weather to jet the incoming air at ceiling level into the house for maximum blending and diffusion of outside and inside air.
Figure 7. Enlargement of a portion of Figure 5 showing inlet baffle adjusted to direct incoming air to the floor.
require a heat transmission coefficient less than 0.398 BTU per hour ft. sq. degree F and a permeability greater than 296 perms.

Furry and Hazen (21) studied the use of models to approximate ventilation distribution phenomenon. They found that the number of air changes and the geometry of the structure were the most important parameters in such a study. Neither Froude nor Reynolds number appeared to impose requirements for modeling of the ventilation dilution phenomena. Low magnitude time scales were recommended to permit satisfactory operation of the monitoring instrumentation.

Weller, et al. (88) used a scale model to determine the air flow characteristics of a ventilation system. A well-defined air jet was formed when air is drawn into a building through a slot inlet. Maximum air velocity, velocity fluctuation and turbulence was contained in this jet resulting in good mixing with air inside the building.

A high correlation was derived between velocity and dispersion, indicating a direct relationship between increased velocity and the amount of mixing that takes place. In the areas outside of the air jet there was no definite pattern of air mixing except for the space directly below the slot inlet. This area was characterized by a complete lack of mixing.
activity. The lack of mixing or turbulence in this area would indicate inadequate ventilation.

Smith and Hazen (81) studied the ventilation inlet to determine the value of models in analyzing inlets. Their study showed that with equivalent Reynolds number and geometric similarity, a geometricly similar jet velocity profile was obtained in models and prototype. To obtain a flow range comparable to that used in animal production buildings a transitional turbulent flow was encountered. This did not provide for maximum effectiveness in mixing of air.

Cramer, et al. (12) compared floor and air temperatures in enclosed buildings and in open front swine buildings. On the open front buildings the inside temperature varied as the outside temperature varied except that it remained 10°F warmer on the inside. With the enclosed building, the air temperature remained at the minimum designed temperature except where the outside temperature dropped below anticipated design conditions. The floor temperatures in the total enclosed building varied from 75°F in November to a low of 55°F by April. The corresponding temperatures for the open front building were 65°F to 45°F.

Odors at a level objectionable to the operator were noticed, however, no adverse effects on the pigs were observed.
The only gas detected was ammonia at levels of 15 to 20 ppm in an air sample taken at 12 inches above the floor. This is below usual human threshold levels.

Esmay and Zindel (20) compared the environmental control in poultry houses with different amounts of wall insulation. They found that greater ventilation was allowable without supplemental heat in the insulated house with a 30°F temperature differential across the wall versus an uninsulated house. The house having a U value of 0.241 BTU per hour sq. ft. degree F allowed 3/8 cfm per bird of ventilation compared to 5/8 cfm for a house having a U value equal to 0.115 BTU per hour sq. ft. degree F.

They also observed that fan operation was highly dependent on outside air temperatures with the thermostat control operating the fans. Duration of fan operation was highly dependent on fan capacity with the inside temperature lowering slightly as the fan operated and increasing slightly with the fan shut off.

Aldrich, et al. (1) studied environment in broiler houses in Pennsylvania. They found that a high relative humidity helped to reduce dust levels in the air, however, it did not suppress it completely. Temperature and air distribution patterns indicated that an exhaust ventilation system with a
slot inlet was doing an acceptable job for winter operation. Dust accumulations on controls considerably reduced the accuracy of control desired. The control units (sensors, controllers and fans) were the weak link in the total system designed.

Isothermal lines plotted across the building show a 10°F variation horizontally at chick height and the same variation vertically. The coolest place on the floor was directly below the slot inlet. Relative humidity increased from a low of approximately 55% when the chicks were introduced to approximately 80% at 56 days of age. This reflected an increase in moisture content of the litter in the house.

A weekly mean minimum and maximum temperature was used to design the ventilating system. The inside design temperatures began at 85°F when the chicks were introduced and remained there until the chicks were three weeks of age reducing each week at 5°F. The recorded temperatures inside the house varied from a low of 60°F to a maximum of 85°F. For approximately the first half of the period the inside temperatures were below the design condition even though the outside temperatures were lower than the weekly mean. Part of this variation could be due to the dust accumulation on the fan controls. Driggers (17) found that fan spacing had no significant effect on the performance of the exhaust type ventilating system. He also
found that the theoretical design will nearly equal the actual conditions that exist in the poultry house. A 4F temperature differential was experienced when using 6.5 cfm per bird of ventilation rate. Part of this ventilation was used to remove the heat from solar radiation. Good mixing of the air was noticed as was evidenced by fluctuation in air movement. Litter moisture contents also remained quite stable indicating satisfactory performance with respect to moisture removal.

Esmay, et al. (19) found that when evaporative cooling takes place in the poultry house (moisture evaporated from the litter), there was less increase in the temperature of the ventilation air as it moved through the house than when no evaporative cooling takes place. The ratio of sensible to latent heat of the air varied from 0.5 to 0.2. The variation was partially explained by differences in outside relative humidity and the proximity to cleaning date. In houses with 3.5 cfm per bird a 5F air temperature rise was experienced on days with outside relative humidities below 50% as the air moved through the house.

Summary of Environmental Control Systems

Satisfactory air distribution can be obtained in a livestock facility by means of exhaust fans and slot inlets. The slot inlets should be designed for air velocities of less than
500 feet per minute. This is a compromise between fan performance and turbulences within the jet of air coming through the slot inlet. At 500 feet per minute the pressure drop across the slot inlet is approximately 1/8" water. At velocities above 500 feet per minute there is an increased pressure drop across the slot inlet causing decreased fan performance and below 500 feet per minute the flow becomes laminar with little or no mixing taking place. Most turbulence and air mixing takes place within the jet of air entering through the inlet. Some stagnation occurs directly below baffled inlets. This area frequently has the lowest temperatures in the building with gradual warming as the air moves toward the exhaust. Temperature variations within the building can be limited to 10°F at chick height. As the air moves from the inlet to the outlet, it replaces the gases already in place including those under slotted floors. Fan spacing has little effect on the performance of the ventilation system.

Some consideration has been given to the use of porous permeable walls and ceilings as inlets. For satisfactory results the thermal conductivity of the material must be very low and the permeability extremely high. Such materials are not presently available. One method of operating a porous permeable wall system would be to increase ventilation rates
sufficiently to remove all moisture produced within the building and adding supplemental heat to maintain temperature, however, this would be decreasing one of the major advantages of such a system, that of reduced ventilation rates to conserve heat.

Environmental control systems operate satisfactorily as long as outside conditions remain within the limits of the original design. Odors objectionable to the operator may be occasionally experienced, but no adverse effects on the livestock have been documented. Hydrogen sulfide and ammonia levels are below human threshold levels under normal system operation, although the hydrogen sulfide concentrations increased to toxic levels when under floor liquid manure storage pits are agitated. This indicates that many gases may be held in solution in the manure storage pit and released when agitated. Additional ventilation may be required when under floor storage pits are emptied or else the manure removal must be accomplished at a time when no animals are present in the facility.

The effect of an oxidation ditch type of under floor manure storage area on the building environment is partially unknown at this time. It is expected to reduce the level of toxic and odorous gases and increase the amount of carbon
dioxide and water vapor released into the building. Additional research is required.

Most of the undesirable gases are associated with the anaerobic decomposition of manure, consequently any system which rapidly separates the animal from its manure and then either removes the manure from the building or stores the manure aerobically will largely reduce this problem.

Dust accumulation on the controls will reduce the sensitivity of thermostats and humidistats used to control ventilation fans and heaters. This increases the length of time that the equipment is on and off with a resulting greater variation in temperature and humidity conditions. The result may be a failure to maintain satisfactory conditions for production and possibly death under extreme conditions.

Evaporation of liquids may be used to reduce the temperature within the building, however, changes in outside relative humidity will seriously affect the amount of cooling that takes place. The engineer needs to carefully evaluate the enthalpy of the incoming air if evaporative cooling is to be used to insure satisfactory operation of the system. In areas with low relative humidities evaporative cooling may be an inexpensive way to reduce heat stress, while in areas with high relative humidities very little cooling can be expected.
BUILDING EFFECT ON ENVIRONMENTAL CONTROL SYSTEMS

Dingle (15) found that the skin temperature on the outside of roof sections was frequently below outside air temperatures on calm clear nights even though there was an outward flow of heat from the building structure. There was also a wide difference between the north and south slopes of the roof in terms of the skin temperatures. Direct solar radiation was found to increase the insulating value of any roofing material when net heat flow was outward. Diffuse radiation warmed the skin surface only at the same rate as the outside air temperature increased when the net heat flow was outward.

Dale (13) studied the effect of roofing material on the temperature experienced in the attic portion of the building. In a ten day study with asphalt shingles the following conditions were found:

- average outside air temperature = 85.9°F,
- average outside surface temperature = 135.3°F,
- average inside surface temperature = 117.5°F,
- average inside air temperature = 111.1°F and
- average outside air velocity = 6.6 mph.

The use of white painted galvanized steel increased the reflection of solar radiation with a resulting increase in heat transfer coefficient (U value). On a north sloping roof
section the painted white galvanized steel had a U value of 1.324 BTU/sq. ft. hr. degree F while aluminum had a U value of 0.820 BTU/sq. ft. hr. degree F. Standard procedures for determination of U values would have shown that both of these materials had the same U value and this would indicate that some consideration must be given to absorption or reflection of solar radiation when calculating heat loss or gain in building construction.

Walker (85) compared the predicted temperatures in green houses to those actually obtained. He found that the predicted temperature based on energy balance were less than 2.5F different than those actually observed. The difference had a random variation and that there was no statistical difference between the two values. He concluded that the energy balance relationship would permit the prediction of temperatures within ventilated green houses when solar and thermal radiation were considered. During periods of high solar radiation and high ventilation rates, the heat loss to the ground could be neglected.

Harman (32) found that a solar collector was able to raise the temperature of the incoming air on a sunny day. The increase in heat allowed for a higher ventilation rate through the building and as a consequence a greater moisture removal
by the ventilation system. He also noted that there was a significant thermal lag of the building which reduced the rate at which the inside temperature increased compared to the rate at which the outside air temperature increased.

No benefit was noted from the solar collectors on a cloudy day. In addition considerable negative radiation was experienced at night. No heat loss value was assigned to the negative radiation.

Neubauer and Cramer (69) studied the effect of building shape on interior temperature. They found that a cubical shaped building was the coolest followed by the cylindrical shaped building, quonset type and dome. A north south orientation appeared to be the coolest for a rectangular building. A gable roofed building with an east west orientation was slightly warmer and a tall cube was considerably warmer. All models were constructed of the same material and painted a dull black. The heat capacity of the shell or contents as well as insulation and infiltration were omitted from the discussion. It was mentioned that on hot sunny days a black surface had a skin temperature of 86°F higher than ambient while white surfaces had temperatures 23°F above ambient.

Jones (43) studied the effect of color on the absorbed heat. She found a difference of 12°F between colors of equal
lightness varying only in hue. Colors containing green and yellow absorbed the most heat while long wave length red and yellow red were cooler and the short wave length purple and blue were the coolest hues. With colors described as whitish or very light, the reflectance was 72.5%. Their mean temperature rise above ambient air temperature was 35°F or 43% of the rise for black surfaces.

It was concluded that middle wave length hues absorbed more heat than those hues which approach either infrared or ultraviolet. A 10°F temperature difference was noted between two colors of the same hue in saturation except that one could be distinguished as being lighter than the other.

Bond, et al. (11) lists four types of radiant energy.
They are: (a) a direct beam solar energy from the sun; (b) a diffuse sky radiation that has been scattered, reflected and diffused out of the original beam; (c) atmospheric radiation emitted by particles or gases in the atmosphere and (d) emitted and reflected energy from surrounding terrestrial objects.
Of the total radiation on a horizontal surface, about 59% was direct beam energy, 9% was diffuse sky energy and 32% was radiation of long wave lengths. Removal of direct beam solar radiation reduced the radiation on the horizontal surface by approximately 50% under low humidity clear sky conditions.
ASHRAE [(3) pp. 467-512] states that steady state analysis is the basis for most design of animal ventilation systems. The required air flow is related to generation rate and building loss as: \[ \dot{m}(X_e) + q = \dot{m}(X_i) + [(X_i) - (X_o)] / R \]

Where \( \dot{m} \) = mass flow rate of air,

\( X \) = the concentration of the quantity considered,

\( q \) = the generation rate of the material within the system,

\( R \) = the resistance to transfer through the system boundaries,

\( e \) = the conditions in the air entering the system,

\( i \) = the conditions in the air within the system and

\( o \) = the conditions in the air surrounding the system (outside).

The rate of transfer through the walls is usually neglected except for heat transfer. Condensation within the wall cavity should be considered under conditions where significant condensation occurs as may be found under high relative humidity conditions inside the building and low temperature and relative humidity outside the building.

For calculation of cooling needs, it is suggested that the load be divided into sensible (change in temperature) and latent heat (moisture content) loads. Solar radiation needs
to be included and is given as \((455 \pm 2\%) \text{ BTU/hr. ft.}^2\) on a surface normal to the sun's rays. The variation goes from 428 BTU/hr. ft.\(^2\) on June 22 to 457 BTU/hr. ft.\(^2\) on Dec. 22. The total shortwave radiation (that which heats the structure) is given as:

\[
I_t = I_{\text{on}} \cos \theta + I_d + I_r
\]

Where \(I_t\) = total shortwave radiation,
\(I_{\text{on}}\) = direct normal radiation,
\(\theta\) = angle of incidence between solar rays and a line perpendicular to the surface,
\(I_d\) = diffuse sky radiation and
\(I_r\) = radiation reflected from surrounding surfaces.

A heat balance at a sunlit surface gives the heat flux into the surface as:

\[
\frac{q}{A} = \alpha I_t + h_o (t_o - t_s) - E\Delta R
\]

Where \(\frac{q}{A}\) = heat flux into the surface,
\(\alpha\) = absorbtance of the surface for solar radiation,
\(I_t\) = total solar radiation incident on the surface (BTU/hr. ft.\(^2\)),
\(h_o\) = coefficient of heat transfer by radiation and convection at the outer surface (BTU/hr. ft.\(^2\) degree F),
\(t_o\) = outdoor temperature degree F,
\(t_s\) = surface temperature degree F,
\(E\) = emittance of the surface and
\[ \Delta R = \text{the difference between the longwave radiation incident on the surface from the sky and surroundings, and the radiation emitted by a black body at outdoor temperature (BTU/hr. ft.}^2\).}

In an effort to reduce the calculations required to determine the effect of solar radiation an equivalent surface temperature or sol-air temperature \( (t_e) \) is given as:

\[ t_e = t_o + \alpha I_t / h_o - E \Delta R / h_o . \]

For horizontal surfaces subjected to sky radiation only \( \Delta R \) is about 20 BTU and with \( E \approx 1 \) and \( h_o \approx 3.0 \), this term is about \(-7F\). For horizontal surfaces it is common practice to assume \( \Delta R = 0 \). These values have been calculated for light colored surfaces \((\alpha / h_o = 0.15)\) and the maximum value likely to occur \((\alpha / h_o = 0.30)\). These are reproduced in part on Table 1. These sol-air temperatures can be adjusted to any other air temperature cycle simply by adding or subtracting the difference between the desired air temperature and the values given in Table 1. The sol-air temperature for other latitudes can be calculated whenever the total solar radiation is known.

The steady state heat gain is given as: \( Q = UA(T) \).

Where \( Q = \text{total heat gain}, \)

\[ U = \text{heat transmittance of the wall section and} \]
\[ \Delta T = \text{temperature difference across the wall.} \]

To use the sol-air temperature with the steady state heat equation, the exterior air film resistance is not included in the calculation of the U value and the temperature differential is the difference in temperature between the sol-air and inside temperatures.

Table 1. Sol-Air Temperatures for July 21, 40 deg. North Latitude

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</tr>
<tr>
<td>24</td>
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<td>77</td>
</tr>
</tbody>
</table>

Palmatier (70) suggests that the psychrometric chart can be used to visualize pertinent processes, solve specific problems and signal potentially dangerous conditions. ASHRAE [(2) pp. 179-198] outlines the basic processes on the psychrometric chart. These processes are given in Figure 8.
Figure 8. The psychrometric chart showing the 3 basic processes that air undergoes during conditioning for use in animal housing.
CALORIMETRIC DATA FOR DESIGN OF VENTILATION

Longhouse (49) and Longhouse, et al. (50, 51) provide heat and moisture design data for poultry housing. These figures are reproduced as part of ASAE data: 249.2 and are reproduced as Table 2 for convenience of the reader.

Table 2. Average Hourly Moisture Production of 1000 4 lb. (1.8 kg) White Leghorn Laying Chickens at Various Temperatures

<table>
<thead>
<tr>
<th>Temperature</th>
<th>Respired</th>
<th>Defecated</th>
<th>Total*</th>
</tr>
</thead>
<tbody>
<tr>
<td>deg F</td>
<td>deg C</td>
<td>lb</td>
<td>kg</td>
</tr>
<tr>
<td>25</td>
<td>-3.9</td>
<td>6.3</td>
<td>2.9</td>
</tr>
<tr>
<td>35</td>
<td>1.7</td>
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<tr>
<td>45</td>
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<tr>
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</tr>
<tr>
<td>95</td>
<td>35.0</td>
<td>20.0</td>
<td>9.1</td>
</tr>
</tbody>
</table>

*Includes drinking water wasted by hens estimated at 10% of water consumed from 25 to 80 °F (-3.9 to 36.7 °C) and 15% at 95 °F (35.0 °C).

Similar data are available for most classes of farm livestock and those for swine are being reproduced in part as Table 3 for the convenience of the reader. For the complete listing of heat and moisture production data for swine, beef, dairy and sheep, the reader is referred to ASAE data: 249.2. These data are published in their entirety in the Agricultural Engineering Yearbook published annually.
Table 3. Heat and Moisture Production in Psychrometric Chamber for Swine at Various Temperatures

<table>
<thead>
<tr>
<th>Weight of hogs lbs.</th>
<th>40F</th>
<th></th>
<th></th>
<th>60F</th>
<th></th>
<th></th>
<th>80F</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Total BTU per hr.</td>
<td>Sens. BTU per hr.</td>
<td>Latent BTU per hr.</td>
<td>Total BTU per hr.</td>
<td>Sens. BTU per hr.</td>
<td>Latent BTU per hr.</td>
<td>Total BTU per hr.</td>
<td>Sens. BTU per hr.</td>
<td>Latent BTU per hr.</td>
</tr>
<tr>
<td>50</td>
<td>429</td>
<td>308</td>
<td>121</td>
<td>358</td>
<td>206</td>
<td>152</td>
<td>339</td>
<td>93</td>
<td>246</td>
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<tr>
<td>100</td>
<td>589</td>
<td>438</td>
<td>151</td>
<td>470</td>
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<td>183</td>
<td>427</td>
<td>141</td>
<td>286</td>
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<tr>
<td>150</td>
<td>729</td>
<td>548</td>
<td>181</td>
<td>567</td>
<td>355</td>
<td>212</td>
<td>502</td>
<td>181</td>
<td>321</td>
</tr>
<tr>
<td>200</td>
<td>860</td>
<td>649</td>
<td>211</td>
<td>657</td>
<td>419</td>
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<td>571</td>
<td>223</td>
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<td>250</td>
<td>984</td>
<td>747</td>
<td>237</td>
<td>741</td>
<td>483</td>
<td>258</td>
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<td>366</td>
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<td>1104</td>
<td>848</td>
<td>256</td>
<td>821</td>
<td>550</td>
<td>271</td>
<td>697</td>
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<td>951</td>
<td>269</td>
<td>900</td>
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<td>364</td>
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<tr>
<td>400</td>
<td>1335</td>
<td>1063</td>
<td>272</td>
<td>976</td>
<td>706</td>
<td>270</td>
<td>813</td>
<td>468</td>
<td>345</td>
</tr>
</tbody>
</table>
Hahn, et al. (31) suggests the use of weather data as an aid in determining the value of shelter for livestock and for selecting the design conditions for those shelters. Using Des Moines, Iowa weather data showing that temperatures were below freezing for 72% of the month of January, he felt it was imperative that some form of shelter be used for all classes of livestock. With the wind from the NW to NNW over 60% of the month, an open front shed should be open to the East.

He also suggests that consideration of the frequency distribution of weather extremes be considered. The third and sixth highest temperatures were suggested for design of air conditioning systems. Again using data for Des Moines, Iowa, he estimates the probability of a summer having 6 days above 98.5F as being 0.20 or recurring once in 5 years on the average.

Becker and Alyea (6) and Green (23) have prepared tables of probability for temperatures in Wyoming and Arizona respectively. These tables list the probability of the maximum temperatures being at or below a specified level. An example of such a table is reproduced in part as Table 4.
Neild and Myers (68) have prepared a series of probabilities of hot and cold days for various locations in Nebraska. They list the percentage chance of having N or more hot or cold days per week for each week of the year. An example of this is included for Alliance, Nebraska in part as Table 5.

Table 4. Empirical Probabilities of Observing Maximum Temperatures Less Than, or Equal to the Specified Values at Parker, Arizona [Nov. 1893-Dec. 1932]

<table>
<thead>
<tr>
<th>Elev. (ground): 350</th>
<th>Period of Record: 11/93 - 12/32</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum Temp. (F)</td>
<td>May</td>
</tr>
<tr>
<td>76</td>
<td>2.6</td>
</tr>
<tr>
<td>78</td>
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<td>80</td>
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<td>82</td>
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<tr>
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Table 5. Percentage Chance of Having N or More Hot Days Per Week at Alliance, Nebr.

Period of Record: Sept. 1896 to Dec. 1963

<table>
<thead>
<tr>
<th>Days/Week</th>
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<th>Jn14</th>
<th>Jn21</th>
<th>Jn28</th>
<th>J010</th>
<th>J112</th>
<th>J119</th>
<th>J126</th>
<th>Au02</th>
<th>Au09</th>
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<th>J112</th>
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<th>Days/Week</th>
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<tr>
<th>Days/Week</th>
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<th>Jn14</th>
<th>Jn21</th>
<th>Jn28</th>
<th>J010</th>
<th>J112</th>
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<th>J126</th>
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</table>
Shaw and Waite (80) have listed the weather bureau normals for maximum and minimum temperatures for various locations in Iowa by month. They then provide a table of adjustments to convert these to selected quantities for the temperatures to be equal to or below the determined value. The adjustment table and normal temperatures for Des Moines, Iowa are reproduced as Table 6.

Table 6. Weather Bureau Normals for Des Moines, Iowa

<table>
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<tr>
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</thead>
<tbody>
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<td>33.9</td>
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<td>85.4</td>
<td>77.6</td>
<td>66.4</td>
<td>47.5</td>
<td>34.9</td>
</tr>
</tbody>
</table>

| Minimum Temperature | 12.5 | 15.9 | 25.8 | 39.1 | 50.7| 61.2 | 65.6 | 63.6 | 54.1  | 43.0 | 28.3 | 18.1 |

Adjustment in °F to be made to normal temperature to obtain mean monthly temperature values which have selected probabilities of occurring.

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<th>B</th>
<th>C</th>
<th>D</th>
<th>E</th>
<th>F</th>
<th>G</th>
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<td>-9.0</td>
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<td>-7.4</td>
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<td>+4.7</td>
<td>+4.4</td>
<td>+4.0</td>
<td>+3.7</td>
<td>+3.4</td>
<td>+3.0</td>
<td>+2.7</td>
<td>+2.3</td>
<td></td>
</tr>
<tr>
<td>90...9.6</td>
<td>+9.0</td>
<td>+8.3</td>
<td>+7.7</td>
<td>+7.0</td>
<td>+6.4</td>
<td>+5.8</td>
<td>+5.1</td>
<td>+4.5</td>
<td></td>
</tr>
<tr>
<td>95...12.3</td>
<td>+11.5</td>
<td>+10.7</td>
<td>+9.8</td>
<td>+9.0</td>
<td>+8.2</td>
<td>+7.4</td>
<td>+6.6</td>
<td>+5.7</td>
<td></td>
</tr>
</tbody>
</table>
After the probability of the occurrence of an event has been obtained, the engineer must determine what risk of failure he and his client will tolerate. In evaluating this choice he must consider the consequences of failure and act accordingly. If death results from system failure, a very high probability of success must be used. A lower probability of success can be used where failure results only in decreased performance. No attempt to control a variable needs to be made if it will not affect performance.

With probabilities listed by month as in Table 4, it is easy to see the annual trend of temperatures. The author suggests that the 68 percentile maximum July temperature be used as a design high temperature and that the design be checked at the 95 percentile level to determine the degree of control lost. Using Parker, Arizona as an example, the 68% temperature is given as 110°F and the 95% is 116°F. In a normal year, July would have 10 days out of control, June would have 6 days, May would have 1 day, August would have 7 days, and September would have 3 days for a total of 27 days out of control. Growing animals respond to the average diurnal temperature, therefore if the cooling system were to lower the night time temperature 6°F below the upper performance limit, no reduction in production would be expected.
A similar procedure can be used to determine the design temperature for heating equipment. It is suggested that a 33% chance of failure be incurred for the minimum January temperature. This would result in temperature below design on 25 to 30 days during the winter season. Checking the 5% level reveals that the control would be lost by 6°F to 20°F. A 20°F loss of control may result in frozen water pipes and death of newborn animals so a lesser chance of failure would be tolerated. In areas where more than 10°F difference occurs between the 5% and 33% levels, it is suggested that a temperature 10°F warmer than the 5% level be used. This would insure that plumbing would not freeze, however, newborn animals may have their growth rate reduced for a short period. Huddling of the animals would reduce any stress caused by low temperatures.

Probability data presented in a form similar to that in Table 5 are useful by indicating the distribution and duration of the temperature of concern along with the probability of more temperatures of similar magnitude. One of the disadvantages of this form is the difficulty in deciding what design temperature should be used. The author suggests that the design temperature be selected by totaling the probabilities for 3 or more days per week for any 5 week period. If the total exceeds 100, the anticipated design temperature is too low and
the next higher temperature should be tried. This procedure would provide for the system to reach its design limits for at least 3 days for 5 consecutive weeks in 2 years out of ten or for 3 days each year.

Before selecting the outside design temperature by the above procedure, the probability for single day temperatures 6°F higher should be checked. If the total of the 1 day probabilities for 10 consecutive weeks is less than 100, the selected outside temperature is satisfactory and the system will be out of control by more than 6°F only 10 days in 10 years. If the 6°F higher temperature probability total exceeds 100, determine the temperature at which the total is 100 or less and use a temperature 6°F lower as the design temperature. This will result in the equipment being used to its fullest extent less frequently, but will help insure that animal mortality will not be a problem.

The same procedure is suggested for cold temperatures, except that 10°F out of control is tolerated. The reason for the greater tolerance to cold is the proximity of the inside design temperatures to stress. Except for animals less than 1 month of age, the inside design temperatures are based on feed efficiency rather than true stress where the animal is in danger of dying. Since there is more spread between the
design temperature and the stress temperature at low temperatures than at high temperatures, a greater degree of control loss can be tolerated for periods of a day or less as the case may be where probabilities are based on maximum or minimum temperatures. The probabilities may need to be re-evaluated as the various species become acclimated to continuously controlled environments. It may be that the animals will become less tolerant to temperature variations as there is less variation to cope with.
SUMMARY

The response of livestock to their environment and to changes in that environment has been quite well documented with respect to temperature and relative humidity. The effects of dust and odors along with other components of the atmosphere have been less well documented, although some progress has been made in that direction in recent years. The notable exception is poultry housing where dust effects have been studied quite extensively. Light and sound effects have not been studied in any detail except to note that continuous steady state sound has little effect except where pain results. Production of eggs may be stimulated by changing the duration of light and dark periods although more study needs to be done in this area.

Environmental control systems in use in livestock facilities can be expected to operate in accordance with design expectations. Properly adjusted slot inlets can be used to direct incoming air along the ceiling for maximum mixing and warming before reaching animal level or down the wall for minimal mixing and warming before reaching the animals. Some disagreement exists on the amount of turbulence or mixing taking place directly under the slot inlet. Air flow into the fan appears to be uniform from all areas as long as no
restrictions are placed in the vicinity of the fan. Temperatures vary across the building with the lowest temperature at the floor beneath the slot inlet and the highest temperature at the ceiling near the fan. Temperature variations of 3F to 5F have been measured with time and also between the floor and ceiling of ventilated buildings.

Pressure ventilation systems have not been discussed in any of the literature cited. The author is aware of two such systems in operation. Where it has been used, it appears to be operating satisfactorily. One of the major disadvantages of such a system is the difficulty of predicting the distribution of the air from a duct if there is only a fan at the end of the duct. As a result there is usually some readjustment that must be made to the duct outlets after the system has been installed. In an effort to insure good distribution, some systems use a small fan at each outlet of the duct. This has worked very well in connection with zone cooling systems. It is the opinion of the author that a pressure ventilation system is more desirable than an exhaust type system in a building which is not reasonably air-tight, one that has numerous openings that are not easily closed, such as might be encountered in an old building which is being partly renovated for confinement livestock production. Since the location of
openings through which air may enter is not controlled. The use of the pressure system causes the warm air to be exhausted through such openings with little effect on the animals. During cold weather operations exhaust openings may freeze shut.

Models of air handling systems will predict air flow patterns. The design of the models must be based on the number of air changes per unit of time and geometric similarity.

Heat flow through building materials can be predicted if all heat sources are identified. Solar radiation will increase the surface temperature above that of the surrounding air and needs to be considered in the design of cooling systems. Light colored surfaces will reduce the effect of solar radiation. Negative radiation at night may cause the surface to become cooler than air temperature and may need to be considered in the design of heating systems. Sol-air temperatures have been calculated to show the effect of positive radiation. Building shape and orientation can be expected to influence the interior temperature by varying the percentage of the surface area exposed to direct and diffuse solar radiation.

Calorimetric heat and moisture production data are available for swine and poultry with less complete information available for beef, dairy or sheep. These data can be used
to approximate the heat and moisture produced in production design.

Probability data are available in various forms for all areas of the United States. The interpretation of this information has been left to the discretion of the engineer involved. For livestock production the author has suggested that the 68% probability for maximum July temperature be used for design of cooling and the 33% probability for minimum January temperature for heating system design be used. These values need to be checked against the 95% probability to determine the degree of control loss.

Design Factors the Engineer Can Use With Confidence

In setting design limits for the environment that livestock will be exposed to, the engineer can be quite confident that a desirable response will be obtained if the following conditions are met: (1) temperature between 50°F and 80°F, lowering 10°F as the animal matures, (2) relative humidity, no effect at temperatures between 50°F and 80°F except that condensation on walls and ceiling may be a problem above 80% relative humidity and dust problems increase below 50% relative humidity, (3) air velocities will have little effect at temperatures near 70°F to 80°F but can be used to reduce heat stress at temperatures between 80°F and 95°F and should be kept below 50 fpm below 70°F to reduce cold stress.
The steady state design equations for heat and moisture balance can be used with confidence if the heat balance includes the sol-air concept and the ability of the building to absorb or give off heat is considered. Insulation values for most materials in use have been determined. Moisture resistance for some materials has been evaluated if the material is considered to have any significant value as a vapor barrier. Solar radiation effects have been documented sufficiently to enable the engineer to use this information in the overall design.

Design Factors the Engineer Can Use With Caution

The selection of outside design temperatures remains an area of judgement. Probability data can be a valuable tool in reinforcing his convictions, but cannot be used blindly. Each area needs to be evaluated with respect to its own environmental peculiarities including the temperature-relative humidity combinations that may be expected and diurnal temperature variation.

Some information is available on animal response to various gaseous concentrations. Information on threshold levels for humans can be used as a guide to system design, but should not be used to indicate an animal response. There are some indications that long term exposure to 50 ppm of ammonia will
not affect production even though the operator may find this objectionable. In some cases 250 ppm of hydrogen sulfide has been cited as causing mortality of beef animals.

Dust levels should be kept low to prevent interference with controls, fans and feeding equipment. Its effect on livestock is not known in sufficient detail to permit the engineer to be confident that any dust concentration will not result in respiratory infections or otherwise cause production losses. Increases in airborne dust is closely related to the activity of the animals. There is some indication that dust levels vary inversely with relative humidity.

The amount of dust in the air at any one time may be controlled by using sufficient air to transport the dust out of the structure as rapidly as it is produced by reducing dust production, or by trapping it in dust filters. With the small amount of information available on the causes of dust, only an educated guess can be made on methods for reducing the dust production. Some methods that may be tried are: (1) covering all feeding equipment to contain this dust within the conveyors and feeders, (2) eliminate all litter and bedding, (3) use liquid feed and (4) maintain high relative humidities (around 70%).

Air distribution remains an area of judgement for the engineer. Air handling systems can be designed with some
assurance that proper distribution will be achieved with slot
inlet-exhaust fan systems and with some types of pressure
systems. If any variations are to be used, there must be some
on site adjustment and calibration or a model must be built to
determine whether adequate distribution will be accomplished.

Factors About Which There is Insufficient Information

The Extension Agricultural Engineer needs more informa-
tion about the rate of odor and dust production and the com-
position of each. He needs to know what causes odor to be
produced and what can be used to control the odor and dust
other than to transport it out of the facility. With the
emphasis on pollution control in the environment, the Exten-
sion Agricultural Engineer also needs to know what treatment
of discharge air will meet the requirements of persons down-
wind from the facility for freedom from odor and dust.

More information is needed with regard to animal behavior
and its effect on the environment. Can an animal be induced
to contribute to a more easily controlled environment by fact-
ors within that environment, such as directing a cool air
stream in a area to encourage defecation within an area that
is cleaned immediately after the animal defecates in the area.
Can carbon dioxide levels be used to encourage resting rather
than fighting or scratching in the litter without affecting performance.

Of the obviously needed research projects to improve design parameters, the following priorities are suggested.

1. The rate of toxic gas production and tolerance levels before production is reduced.

2. The sources of odors in livestock facilities.

3. The causes and rate of dust production and methods for controlling dust.

4. Heat and moisture production under actual production conditions, eq. slotted floors, partially slotted floors and bedded floors.

5. Methods for improving air distribution to prevent stagnation areas within the building during hot weather.

6. Improvement of the calorimetric heat and moisture production for beef, dairy cattle and sheep.
PRACTICAL APPLICATION OF THE RESEARCH INFORMATION

The Extension Agricultural Engineer needs to evaluate the research results, compare them with his own experiences and those of his clientele, evaluate the economics of the situation, design an operable environmental control system and provide advice regarding the successful management of the system. He may also need to serve as a trouble shooter and to offer suggestions on methods to improve existing systems.

Some of the decisions that must be made in designing and operating a system can best be brought to the foreground by the use of an example. The solution is a four step process involving the definition of the problem, gathering pertinent information, evaluating the quality of the information available and the design of the system. The example that follows does not include all of the aspects of the problem.

Problem

Design the environmental control system to maintain comfort zone conditions to raise 500 pigs from 70 lbs. to 200 lbs. The building is 36' X 125' with a 7'6" ceiling and attic above. The long axis of the building has an east-west orientation. The building is clad with 28 gauge white enameled steel, lined with 1/2" exterior grade plywood and has 3" fiber
glass insulation in the sidewalls and ceiling. No bedding is to be used and floors are partially slotted. The facility is to be located near Des Moines, Iowa.

Step 1. Define the problem

In the definition of the problem the Extension Agricultural Engineer determines what information he is being asked to supply. In this problem he must (a) decide under what conditions the pigs will respond to produce a pound of gain for the least expenditure for feed and environmental control system operation, (b) the heat, moisture, odor and dust production rate at those conditions, (c) the climatic variables of the location, (d) the effect of the animals and the climatic conditions upon the environment to be controlled, (e) the size of heating, cooling, and ventilation equipment and whether it is needed, (f) the method of heating, cooling, ventilating and the air handling system to be used and (g) determine whether the insulation is adequate.

Step 2. Collect the information

(a) Under what conditions do pigs respond with the highest feed efficiency and rate of gain? From Morrison's data (Figure 1) it appears that 80% of optimum gain can be obtained at 80°F and 80% RH. If the pigs are cooled to 64°F at night,
the average would be 72, the optimum temperature for gain. This should result in little or no reduction in rate of gain or feed efficiency according to Bell, Bond, Hazen, and Mangold. At the other end of the scale, Mangold states that at 57°F feed efficiency begins to decrease. A 0.0016 lbs. feed/lb. gain per degree F drop would result in the consumption of an additional 0.0112 lbs. feed/lb. of gain at 50°F. For 500 pigs gaining 1.6 lbs./day this would require an additional 8.96 lbs. feed/day to get the same gain. At 2¢/lb. of feed this would cost 17.9¢/day or 0.035¢/pig day in feed. This cost may or may not be less than the cost for heating to maintain the higher temperature. As a starting point the assumption will be that the feed cost is less than the heating cost and 50°F will be used as the lower temperature. It may need to be re-evaluated when the heating equipment is sized.

At temperatures below 70°F Morrison shows little effect for relative humidities and at 80°F the difference in rate of gain between RH of 30% and 80% is 15% of optimum. Since condensation on building walls may be a problem at RH above 80% this will be used as the upper limit. Dust may be a problem at RH below 50%. With little information available on the effect of dust and gases except for toxic levels, these will be established below toxic levels. The comfort zone then will
be defined as:

- **temperature**: 50 to 80°F,
- **relative humidity**: 50 to 80%,
- **dust**: less than 0.1 mg/ft.³,
- **ammonia**: less than 25 ppm,
- **hydrogen sulfide**: less than 2 ppm and
- **oxygen**: approximately 16%.

(b) Since the floors are partially slotted and no bedding is used, it is expected that moisture production will be similar to that in a psychrometric chamber. As the pigs grow, total heat output increases, but it is expected that the building will never be completely filled with 200 lb. nor with 70 lb. pigs. A more realistic loading would be partially filled with pigs of two or more different weights. The worst conditions would be the larger pigs during hot weather and the smaller pigs during the cold weather. For this problem it will be assumed that the building is 1/2 full of 175 lb. pigs and 1/2 full of 115 lb. pigs. If the client specified that the building would be full of pigs of the same weight or only partially filled at times, the calculations would be adjusted accordingly. Completely filled with heavy pigs increases cooling load. Completely filled with lightweight pigs or partially filled would likely require the addition of supplemental heat.
The heat produced is given by Bond as:

<table>
<thead>
<tr>
<th>Sens.</th>
<th>Lat.</th>
<th>Total</th>
<th>Sens.</th>
<th>Lat.</th>
<th>Total</th>
</tr>
</thead>
<tbody>
<tr>
<td>115 lb. pig</td>
<td>380</td>
<td>170</td>
<td>550</td>
<td>150</td>
<td>300</td>
</tr>
<tr>
<td>175 lb. pig</td>
<td>490</td>
<td>210</td>
<td>700</td>
<td>200</td>
<td>335</td>
</tr>
</tbody>
</table>

The heat produced by 500 pigs is:

<table>
<thead>
<tr>
<th>50F (BTU/hr.)</th>
<th>80F (BTU/hr.)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total</td>
<td>312,500</td>
</tr>
<tr>
<td>Latent</td>
<td>95,000</td>
</tr>
<tr>
<td>Sensible</td>
<td>217,500</td>
</tr>
</tbody>
</table>

The latent heat can be converted to lbs. of moisture by dividing the latent heat of water vaporized at body temperature (1043 BTU/lb.). This yields a water production of 91 lbs./hr. at 50F and 152 lbs./hr. at 80F.

No figures are available on the rate of dust and odor production, therefore a best estimate needs to be made. The operation of systems designed for moisture and temperature appear to keep dust and odors below those shown to be harmful even though the operator has objected to the odor levels. The assumption will be made that odor and dust levels will be tolerable if moisture and temperature are maintained within the comfort zone as defined in Step 2a.
(c) Using Shaw and Waite's data for the Des Moines area, the average July maximum is 88.3F. Adjusting this to the 75% probability requires adding 2.0F and 4.9F for the 95% probability level. Since the 95% level is only 5F higher than the average maximum temperature, there is little advantage in adjusting to the 68% level as suggested in the text. To simplify calculations a temperature of 89F will be used rather than 88.3F. The precision of operating controls also limits the accuracy to which temperatures can be controlled, therefore little accuracy is lost by the rounding of the number to 2 significant digits.

The average January minimum temperature is given as 12.5F. For January temperatures Des Moines is in Zone C and to correct for the 33% level requires an adjustment of approximately -5F. At the 5% level an adjustment of -10.7F is needed. Since there is less than 10F difference between the 33% and the 5% level, the 33% level of 7F will be used.

The average wet bulb temperature for an 89F dry bulb temperature at Des Moines, Iowa is given as 75F. This corresponds to a RH of 50%. The average wet bulb temperature for a 7F dry bulb temperature is 6F corresponding to a RH of 80%. The design temperatures and relative humidities to be used are:
minimum temperature: 7°F and 80% RH and  
maximum temperature: 89°F and 50% RH.

(d) Determine the effect of the animals and the climate conditions upon the environment to be controlled, and the heat transmission coefficients for the building. The U value is defined as the reciprocal of the sums of the resistances or 

\[ U = \frac{1}{\left(\frac{1}{f_i} + \frac{L}{k} + \frac{1}{C} + \frac{1}{f_o}\right)} \]

Where \( f_i \) = conductance of a still air film,
\( L \) = thickness of the material,
\( k \) = conductance of one inch of material,
\( C \) = total conductance of a material and
\( f_o \) = conductance of an air film in 15 mph wind.

The heat transmission coefficients given in ASHRAE (2) are used to calculate the U value.

<table>
<thead>
<tr>
<th>material</th>
<th>k</th>
<th>L</th>
<th>C</th>
<th>R</th>
</tr>
</thead>
<tbody>
<tr>
<td>outside air film:</td>
<td>6.00</td>
<td></td>
<td>.166</td>
<td></td>
</tr>
<tr>
<td>28 gauge steel:</td>
<td></td>
<td></td>
<td>negligible</td>
<td></td>
</tr>
<tr>
<td>3&quot; fiberglass insulation:</td>
<td>.27</td>
<td>3.0</td>
<td>11.210</td>
<td></td>
</tr>
<tr>
<td>1/2&quot; exterior plywood:</td>
<td>.80</td>
<td>.5</td>
<td>.625</td>
<td></td>
</tr>
<tr>
<td>inside air film:</td>
<td></td>
<td></td>
<td>1.65</td>
<td>.606</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>12.607</td>
<td></td>
</tr>
</tbody>
</table>

With an outside air film \( U = 1/12.61 = 0.0794 \). Without the outside air film \( U = 1/12.44 = 0.0804 \).
Heat flow through the roof ceiling combination is given as $U_c = (U_r)(U_{ce})/[U_r + (U_{ce}/n)]$. Assume the ratio of roof area to ceiling area is 1.05 corresponding to a roof slope of 4:12.

Where $U_r = U$ of roof,

$U_{ce} = U$ of ceiling and

$n = \text{ratio of roof to ceiling area.}$

The $U_r$ value is calculated as:

<table>
<thead>
<tr>
<th>material</th>
<th>$R$</th>
</tr>
</thead>
<tbody>
<tr>
<td>outside air film</td>
<td>.166</td>
</tr>
<tr>
<td>steel</td>
<td>.000</td>
</tr>
<tr>
<td>inside air film</td>
<td>.606</td>
</tr>
</tbody>
</table>

$$U_r = 1/0.722 = 1.29 \text{ or } 1.65 \text{ with no air film outside.}$$

The $U_{ce}$ value is calculated as:

<table>
<thead>
<tr>
<th>material</th>
<th>$R$</th>
</tr>
</thead>
<tbody>
<tr>
<td>inside air film</td>
<td>.606</td>
</tr>
<tr>
<td>3&quot; insulation</td>
<td>11.210</td>
</tr>
<tr>
<td>1/2&quot; plywood</td>
<td>.625</td>
</tr>
<tr>
<td>inside air film</td>
<td>.606</td>
</tr>
</tbody>
</table>

$$U_{ce} = 1/14.047 = 0.071. \quad U_c = (1.29)(0.071)/[1.29 + (0.071/1.05)] = 0.0675. \quad \text{Without the air film } U_c = 0.0680.$$ Since there is such a small difference in the $U$ value no
differentiation will be made between the two and the following U values will be used:

- walls \( U = 0.080 \)
- ceiling \( U = 0.068 \).

(e 1) Calculate the heat and moisture balance for the system. The total heat produced is the sum of all heat sources. This may be expressed as:

\[
Q_t = Q_a + Q_e + Q_v + Q_b + Q_s + Q_f.
\]

Where:

- \( Q_t \) = total heat produced,
- \( Q_a \) = heat produced by the animals,
- \( Q_e \) = heat produced by the mechanical equipment within the building,
- \( Q_b \) = heat gained or lost through the building walls and ceiling,
- \( Q_v \) = heat gained or lost through the ventilation air,
- \( Q_s \) = heat gained or lost through the heating or cooling equipment and
- \( Q_f \) = heat lost through the floor, is negligible after building reaches operating conditions.

For the purpose of this example the heat produced by the mechanical equipment will be neglected since there is insufficient information available to determine what equipment may be operating. In addition the heat produced by the animals overshadows all other heat sources except that lost through the
ventilation air and the heating or cooling equipment. For steady state operation there is no heat stored or removed from the structure or contents, therefore $Q_t = 0$. This leaves 2 terms ($Q_s$ and $Q_v$) to be evaluated. As a first try the supplemental heat can be set equal to zero leaving only the heat gain or loss through the ventilation air to be calculated. For cold temperatures this should work since the air brought into the building is colder than the air inside and the animals would be expected to produce more sensible heat than is lost through the walls and ceiling. Then $Q_v = Q_a - Q_b$.

Where $Q_b = U A \Delta T$, and

$$Q_v = 217,500 - 0.080 (43)(2576) - 0.068 (43)(4500)$$

$$= 195,460 \text{ BTU/hr}.$$

The ventilation air must remove 195,460 BTU of sensible heat and 95,000 BTU of latent heat for a total of 290,460 BTU/hr. The sensible enthalpy of air at 50F and 58% RH as shown on the psychrometric chart is 11.8 BTU/lb. and at 7F and 80% RH is 1.4 BTU/lb., therefore 18,780 lbs. of air/hr. must be moved to keep the building cool. However, this does not insure that adequate ventilation exists for removal of water vapor. A check on the water vapor holding capacity of the air shows 0.0046 lbs. of water/lb. of air at 50F and 50% RH and 0.0009 lbs. of water/lb. of air at 7F and 80% RH. A moisture balance
Figure 9. The psychrometric chart as used in the example problem.
equation would be \( W_t = W_p + W_e - W_v \).

Where \( W_t \) = total water stored by the system,

\( W_p \) = water produced by the animals,

\( W_e \) = water introduced by evaporation of liquids and

\( W_v \) = water removed by the ventilation system.

If no water is to be stored and the evaporation water is
considered to be included with the water produced by the ani­
mals, the \( W_p = W_v \). The water removed by the ventilation air
is \( W_v = M(W_i - W_o) \).

Where \( M \) = mass of air moved,

\( W_i \) = water content of air in the system and

\( W_o \) = water content of air introduced.

Then \( W_v = 18,780 \times (0.0046 - 0.0009) = 69 \text{ lbs. of water/hr.} \).

Since 69 lbs. is less than the 91 lbs./hr. produced, some
moisture would be stored as increased relative humidity. Solv­
ing at 50F and 80% RH would continue to require an air volume
of 18,780 lbs./hr. to maintain temperature, but would increase
the water holding capacity to 0.0061 lbs./lb. and would remove
97 lbs. of water/hr. Therefore it is possible to maintain a
condition of approximately 52F and 64% RH. Assuming that the
heat production remained the same, this ventilation rate should
keep the inside temperature below 80F until the outside temp­
erature reached 35F. This can be shown on the psychrometric
A straight line is used to connect the inside and outside operating conditions. This is the heavy line on Figure 9. The length of the line is determined by the total heat to be removed and the amount of ventilation air while the slope of the line represents the ratio of latent heat to sensible heat produced. As long as the ventilation rate, heat production and sensible-latent heat ratio remains constant the line can be moved to any area on the psychrometric chart and will show the inside conditions for any outside condition selected. This is the procedure used to determine the temperature at which additional ventilation is needed. Actually this results in additional ventilation being required at lower temperatures than the precision calculations since the total animal heat production becomes less as the temperature rises. There is also a change in the sensible-latent heat ratio, however, within the comfort zone this change does not seriously affect the calculations. A new slope of the line should be plotted for each change in ventilation rate until some familiarity with the psychrometric chart is obtained.

A series of calculations is needed to determine the exact outside temperature at which a new ventilation rate would be needed. At an outside temperature above 80°F it is impossible to keep the inside temperature at or below 80°F using only
ventilation air and some type of cooling must be used. A solution to the heat balance equation now becomes \( \sum = 246,500 \text{ BTU} + Q_b + Q_v + Q_s \).

The heat gained through the walls and ceiling may be calculated using the sol-air temperatures for 40° North Latitude given in Table 1. These will not be absolutely correct since Des Moines is at 42° North Latitude, but will give an indication as to the size of cooling equipment that may be needed. A white enameled steel building may be considered to be light colored and \( \alpha/h_o = 0.15 \) will be used. With the design temperature of 89°F, the sol-air temperature corresponding to 90°F air temperature will be used. This may introduce some error into the calculations, however, this is expected to be no larger than those introduced by ignoring ventilation of the attic space, the difference in latitude, roof slope or shading of the south wall by roof overhang.

The sol-air temperatures for 90°F air temperature are slightly higher than would be obtained from the sol-air temperatures for the hottest part of the day if corrected for the difference in air temperature. This is not significant except for a building with a north-south orientation since the late afternoon sun would have a significant warming effect on the west wall. The difference in latitude is not significant.
since the difference in the sun angle would give a change in sold-air temperature as a fraction of a degree and cannot be measured to such accuracy with thermometers in use outside the scientific laboratory. Ignoring roof slope is inconsequential since the increased angle of incidence on the north slope offsets the decreased angle of incidence on the south slope.

Ignoring the ventilation of the attic space will lead to a oversizing of cooling equipment with the result that comfort zone conditions will be maintained for a greater portion of the year. If the amount of air circulating through the attic is known, the temperature in the attic can be determined as

\[
\left[ U_{ra} + U_{ce} - M_a (0.24) \right] \left[ T_a \right] = U_{rs} T_s + U_{ri} T_i - 0.24 M_a T_o.
\]

Where

- \( U \) = heat transfer coefficient (BTU/hr. sq. ft.),
- \( r \) = roof,
- \( ce \) = ceiling,
- \( T_a \) = temperature of attic air (°F),
- \( T_s \) = surface temperature of roof (°F),
- \( T_i \) = room temperature inside building (°F),
- \( T_o \) = outside air temperature (°F) and
- \( M_a \) = mass of air moved through the attic.
When all of the ventilation air is drawn through the attic, the mass of the ventilated air is the mass of air to be used in the equation. Whenever a positive attic ventilation system is used, the effect of the ventilated attic should be calculated. If there is only a vented attic space, the amount of air moving through the attic is highly variable and the assumption of an unvented attic is of little significance.

The shading of the south wall would be expected to reduce the surface temperature of the wall surface approximately 5°F. This is not expected to be significant when the magnitude of other heat sources is considered and will result in a slight oversizing of the equipment.

The sol-air temperatures are as follows:

<table>
<thead>
<tr>
<th>N</th>
<th>E</th>
<th>S</th>
<th>W</th>
<th>Roof</th>
</tr>
</thead>
<tbody>
<tr>
<td>96F</td>
<td>97F</td>
<td>112F</td>
<td>97F</td>
<td>127F</td>
</tr>
</tbody>
</table>

The heat gain through the building walls and ceiling then becomes

\[
Q = 0.080[(8)125(96 - 80) + 8(125)(112 - 80) + 8(36)(97 - 80) + 0.068(4500)(127 - 80)].
\]

\[
Q = 60,400 \text{ BTU/hr.}
\]

The total heat to be removed then is 306,900 BTU/hr. + \(Q_v\).

Again there is one equation with two unknowns. There are two approaches that might be used to determine the amount of ventilation heat that may be added to the environment. One method would be to determine the amount of ventilation
required to remove all of the latent heat, toxic gases or pro-
vide adequate oxygen levels and the other would be to deter-
mine the volume of air required if evaporative coolers were
used to cool the incoming air to the wet bulb temperature.

The removal of latent heat appears to require an exces-
sive amount of air movement due to a difference of only 0.0024
lbs. of water/lb. of air increase in holding capacity. There
is insufficient data to estimate ammonia or hydrogen sulfide
production. Odors may be a problem with inside temperatures
above 70°F due to bacteriological decomposition of the manure.
If winter ventilation rates are used, an increase in odors may
be tolerated during daytime operation if the ventilation rate
is increased at night to remove them. Oxygen consumption by
the pigs could be calculated by knowing the metabolism rate of
the pigs. The rate of oxygen consumption is known to be below
the minimum ventilation rate for winter conditions as is evi-
denced by the research information showing no deaths at these
ventilation rates. Therefore this rate will be used to calcu-
late heat added by the ventilation air. The sensible heat to
be removed from the air is \( Q_v = M_a (0.24)\Delta T = 13,700(0.24) \)
(9) = 40,400 BTU/hr. The total heat to be removed by the
cooling equipment is 40,400 + 60,400 + 246,500 = 347,300 BTU/
hr. If the condition of the air discharged from the cooling
equipment is known, the air flow through the cooling equipment can be calculated, however, the cooling equipment must be designed to handle the volume of air to be passed through it and to remove the desired amount of heat in the proper ratio of sensible to latent heat.

The second alternative would be to use evaporative cooling. Assuming 100% efficiency the air discharged from the cooler would be at 75°F and 100% RH. To pick up 306,900 BTU's of heat would require 102,300 lbs. of air/hr. The inside relative humidity would be approximately 90% and is higher than that determined as being part of the comfort zone. Actually an efficiency of 85% to 90% is more realistic with the type of evaporative coolers presently on the market which forces air through a wetted pad. This decrease in efficiency makes the use of evaporative coolers even less practical.

(e 2) Determine the minimum ventilation rate required for odor, toxic gases and dust control. There is no information available to indicate the rate of production of odor, toxic gases or dust. When available, this can be calculated using the same equation that has been used for moisture control.

\[ P = M(\%_{\text{out}} - \%_{\text{in}}). \]

Where \( P \) = rate of production of obnoxious material,

\[ M = \text{mass of air flow}, \]

\[ \%_{\text{out}} = \text{concentration of obnoxious material in the} \]
discharge air in % and

\[
\%\text{in} = \text{concentration of obnoxious material in incoming air in } \%.
\]

Until more information is available, assume that ventilation for temperature control will be adequate at inside temperature between 50\(^\circ\)F and 70\(^\circ\)F. At temperatures between 70\(^\circ\)F and 80\(^\circ\)F additional ventilation may be needed or else higher odor levels tolerated.

(f 1) Design the air distribution system. For the design of the air handling system the following amounts of air will need to be handled:

<table>
<thead>
<tr>
<th>Outside Temp.</th>
<th>Air flow/pig (cfm)</th>
<th>Total air flow (cfm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>7F to 30F</td>
<td>9.1</td>
<td>4,550</td>
</tr>
<tr>
<td>30F to 50F</td>
<td>20.0</td>
<td>10,000</td>
</tr>
<tr>
<td>50F to 70F</td>
<td>30.0</td>
<td>15,000</td>
</tr>
<tr>
<td>70F to 80F</td>
<td>9.1</td>
<td>4,550</td>
</tr>
</tbody>
</table>

To provide for the refrigeration equipment, two air handling systems will be designed. The first for ventilation air and the second for refrigerated air.

According to Esmay inlet air velocity during cold weather should be limited to 500 fpm for good control. This would require 9.1, 20.0 and 30.0 sq. ft. of inlet for the three ventilation rates. A continuous slot inlet could be constructed to draw air directly from the attic. For 9.1 cfm/pig this inlet would need to be 125' long by 0.875" wide, and 1.95" and 2.92"
wide for 20 and 30 cfm/pig respectively. To simplify conversion to cooling the slot will be located on one side of the building with an adjustable baffle as shown in Figure 5. The air will be drawn into the attic and then through the slot inlet into the building. For hot weather operation the air will be drawn over the eave and through the slot. Exhaust fans will be located in the opposite wall (south in this case is preferred due to prevailing winds). Six fans will be equally spaced, each with 2,500 cfm capacity. These fans are readily available commercially, and the use of several fans increases the flexibility of the system over that of a single fan system. The fans at the 1/3 points will operate during the coldest weather, in addition, the end fans operate when the outside temperature reaches 30°F and all fans operate when the outside temperature is between 50°F and 70°F. At 70°F only the fans at the 1/3 points operate and mechanical refrigeration begins. Thermostatic controls are located outside of the building but protected from direct sunlight.

The location of the thermostats outside of the building is a departure from normal operation of heating and cooling equipment. If the building is only partially occupied, it is difficult to predict the inside temperature and relative humidity conditions and thermostats and humidistats have been
placed inside to respond to these partially filled conditions or to compensate for oversized equipment. Humidistats have consistently failed due to dust accumulating on the sensing element and the failure of the operator to feel a need for cleaning of the elements. This has left the thermostat as the only operable control and during most of the winter season the moisture levels inside were uncontrolled.

With the facility fully occupied and the heat and moisture levels predictable, the moisture levels can be controlled by sensing the condition of the incoming air. This requires that the temperature of the inside air be allowed to vary rather than attempting to control the inside temperature within ± 5°F.

On the psychrometric chart (Figure 9) the saturation line reflects the change in water vapor holding capacity (absolute humidity) of the air with changes in temperature. With constant inside temperature and a ventilation rate based on control of moisture, the moisture will be controlled for all outside temperatures below the design temperature. If the inside temperature decreases at a rate equal to or less than the decrease in outside temperature and the outside temperature is below 40°F, moisture will continue to be controlled. The slope of the saturation line on the psychrometric chart approaches
the slope of the produced sensible-latent heat ratio line at temperatures below 40°F.

Air conditioning units will be installed along the slot inlet wall (north in this problem). Each unit will be equipped with its own circulation fan and dust filter. The air will be drawn from the inside of the building, through the dust filter and blown across the cooling coil and back into the building. Due to the length of the building, a number of units would need to be installed or a single unit with duct work. To remove 307,000 BTU's of heat would require 6 - 50,000 BTU units. This would not result in complete control, however, the definition of the comfort zone and the fact that the system would be inadequate for at least 1/3 of July are sufficient reasons to round the design off to a size commercially available.

A second alternative would be to use a pressure system. A single duct could be used, however, its size would limit its usefulness if duct air velocities are to be kept near 1000 feet per minute. The 1000 fpm limit is somewhat arbitrary, although higher velocities will result in pressure drops in excess of 1/8" of water, the pressure at which most ventilation fans are presently rated. There is also some difficulty in getting satisfactory air distribution with a duct.
Since the exhaust ventilation system is the easier of the two to construct and presents fewer operating difficulties, the author suggests its use in this particular problem. If zone cooling were to be used or this were a remodeling of an older leaky building, a pressure ventilation system would be more desirable in the author's opinion.

(f 2) Adjust and calibrate the system. To insure satisfactory operation of the system each fan and refrigeration control thermostat must be adjusted to respond at the proper temperature. The inlet baffle adjustments must be made to insure good air distribution. This calibration needs to be done because not all of the design parameters are known at this time. Dust filters on the cooling units will be a problem unless something like a self-cleaning filter can be used. Ventilation rates may need to be increased to control odors.

Step 3. Evaluating the information

Up to this point a ventilation rate and refrigeration rate have been calculated for various outside conditions and a rate of heat and moisture production based on an assumption of pig size in the facility. Assumptions were made on the effect of solar radiation and evaporation from the floor, waterer and manure pits. This plus the fact that the heat and moisture production values are averages for a biological
material allow some freedom in deviation from the "exact" solutions and actual practice with little sacrifice. The refrigeration rate of 307,000 BTU/hr. was rounded off to 300,000 BTU/hr. and the minimum ventilation rate of 4,550 cfm was rounded off to 5000 cfm. This results in a slightly higher temperature during the summer conditions, but this was agreed to in the design and a small additional variation is not critical. Ventilation of the attic will help to modify the heating effect since the calculations were based on an unvented attic. At the lower temperatures there may be a slightly lower temperature than called for in the design but again this should not be critical. The fact that the air is being drawn from the attic will help to modify the cooling effect. Any animal heat escaping into the attic is recaptured and will act to prewarm the incoming air, in effect acting as increased insulation in the attic. In addition, any solar radiation acting on the roof will assist in warming the attic air even though its effect were neglected due to the fact that the coldest temperatures would be encountered when solar radiation would not be present.

Throughout the example each calculation was made using the maximum accuracy of the figures available. It is part of the engineer's responsibility to carry out calculations with
as much accuracy as is possible, however he needs to keep in mind the accuracy of the data he is working with. In some cases the data appear to be accurate to three significant digits when in fact it is an average of several tests which may have been accurate to only one or two significant figures. Whenever data are collected from biological materials, animals in this case, there is no assurance that a representative sample is selected and the accuracy of the measuring equipment may be limiting. When all the possible sources of error are considered, the Extension Agricultural Engineer can expect his calculations to be within 10% of the actual solution. A variation of 10% or less is not critical in livestock environment as long as critical conditions are avoided.

In the example no supplemental heat was required. If heat had been required, the engineer would have had to determine the location of the heater as well as the size of the heater. A procedure similar to the one used for refrigeration would give the size of the heater needed. The location and type of heater is less easy to decide. The heaters should be located in such a manner that the heated air mixes with the ventilation air as rapidly as possible. Locations near the slot inlet would accomplish this if the heated air stream is directed perpendicular to the air stream from the slot inlet.
A location along the wall with the exhaust fans would be expected to waste some heat through short circuiting directly from the heater to the exhaust fan.

An alternative would be a single heater with ducts to discharge heat at several locations. If the cold air return to the heater was near the exhaust fans and the discharge ducts near the slot inlet, excellent heat distribution would be expected.

Zone heating with heat lamps or brooders works well with small animals or poultry. This would involve the lowest operating cost since only the space actually occupied by the animals is warmed significantly above the temperature the incoming air would reach if no supplemental heat was used.

The ventilation system can be justified since the animals would die if the oxygen was not replenished and also disease problems could be expected if moisture was allowed to condense on wall and ceiling surfaces and then to drop back onto the animals during cold weather. During hot weather death would result if some cooling was not provided, however, the question remains as to whether air conditioning can be justified. Only rate of gain suffers until death becomes a factor. The use of a mechanical refrigeration system would be expected to give the best control of the environment, however, with the
expectation of little return to pay for the cost of operation, some alternative cooling method needs to be investigated. An alternative would be to use extra ventilation to remove air at a rate to allow a 2°F temperature rise over outside conditions and a sprinkler system to operate whenever inside temperatures exceed 85°F. In some operations zone cooling might be used instead of a sprinkler system, but successful operation is not expected under growing-finishing conditions. A zone cooler would create a situation similar to a fogging nozzle which has been described by Morrison as being undesirable.

The size of sprinkler to use has been a best guess usually using more water than necessary to get the job done. Sprinkler systems described in the literature did not describe the amount of water used for cooling of the animals. One method that may be used to size the sprinkler system is to provide sufficient water to remove the sensible heat produced by the pig by evaporation from the skin. As a starting point 100% efficiency will be assumed, even though 20% may be more realistic. With evaporation each pound of water would pick up approximately 1040 BTU of heat. A 175 lb. pig produces approximately 200 BTU of sensible heat per hour therefore a gallon of water per hour would cool 36 pigs. No information was given in the problem regarding pen size, therefore the author
assumes that not more than 20 pigs will be in any pen. At 20% efficiency a flow of 2.5 gallons per hour per pen would be sufficient. Three gallons per hour is as low a flow rate as can presently be obtained with commercially available spray nozzles without getting a fogging nozzle.

One item which has not been investigated in this example is whether condensation would occur within the wall cavity and cause the insulation to become wet and lose its insulating characteristics. To insure that condensation does not occur, the author suggests that a sheet of 6 mil polyethylene be placed between the exterior plywood and the insulation. The combination of the waterproof glue of the plywood and the polyethylene should limit the amount of water vapor entering the insulation to low enough values to prevent condensation if the construction permits the flow of air between the steel and the insulation. ASHRAE (3) outlines the procedure to be used to check for condensation and the reader is referred to it.

Step 4. Design the system

The author would use a continuous slot inlet along the north wall to draw air through the attic into the building in the winter and exhaust it with propeller type fans of 2500 cfm capacity each in the south wall. During coldest weather the system could be shut down so that only a single fan was
operating and allow moisture and odors to build up for short periods. For most of the winter two fans would operate and everything would be expected to be under control. At 35°F outside temperature 2 additional fans begin to operate. Above 50°F outside temperature the air is drawn over the eave and all fans begin to operate. If temperatures inside the building reach or exceed 85°F, a sprinkler system comes on to allow the pigs to wet their skin and lose heat by surface evaporation.

The control thermostats would be located outside of the building in an area protected from direct solar radiation and precipitation. In addition high and low temperature limit thermostats would be installed inside the building to turn on the sprinkler system or shut off all fans at 85°F and 32°F respectively. This would provide high temperature cooling as well as protect against freezing of plumbing fixtures. In addition to shutting off all the fans the low temperature thermostat would set off an alarm to let the operator know that the ventilation system was not operating and that some remedial action may be necessary. An alarm is also necessary to warn the operator of electrical failure with a consequent loss of ventilation.
Step 5. Summary of the problem

The equipment needed to maintain a satisfactory environment as described in the problem is as follows.

Ventilation fans: 15,000 cfm capacity to be reduced to 5000 cfm for cold weather operation. Use 6 fans of 2500 cfm at 1/8" static pressure rating.

Supplemental heat: none required.

Air conditioning capacity: 300,000 BTU/hr. or a sprinkler system delivering 3 gal./hr. to each pen of 20 pigs.

Slot inlet size: 125' long, adjustable from 0.875" to 3" in width.

Thermostat location: protected from solar radiation and precipitation outside of building, except for limit thermostats located near ceiling and alleyway inside building.
BIBLIOGRAPHY


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