CHAPTER 6
VENTILATION RATES

6-1. Introduction

Frequently, the first step in designing environmental control for agricultural buildings is to determine required ventilation rates. Ventilation in both animal housing and greenhouses is critical. In animal housing, ventilation is frequently the only means of environmental modification. In greenhouses, both heating and ventilation are used for environmental modification, and ventilation is critical when dealing with the large heating effect of the sun.

A properly designed and installed ventilation system must provide sufficient ventilation for summer conditions, adequate minimum ventilation rates for winter, proper staging between the minimum and maximum ventilation rates, and safety alarms to alert the operator should the system fail to maintain the environment within prescribed limits.

Sensible energy and mass balances are the tools used to calculate required minimum and maximum ventilation rates. This chapter will focus on calculating ventilation rates, not on applying field rules which can differ from region to region. However, field recommendations and calculated values should not differ greatly, and if they do, the differences should be explainable (otherwise one must be in error).

Strategies to calculate minimum and maximum ventilation rates for greenhouses differ significantly from strategies for animal housing. Greenhouses usually are ventilated only for temperature control, but may be ventilated for a short time during the late afternoon in cold weather to "dry out the greenhouse". However, for most of a cold day, the only ventilation will be infiltration. During warm weather, greenhouses are ventilated at rates sufficient to limit the temperature rise of the ventilation air as it passes through the greenhouse, or may be evaporatively cooled, with the temperature rise of the evaporatively cooled ventilation air limited to a prescribed amount between inlet and exhaust. A typical limit of temperature rise of ventilation air is 4 K during warm weather.

In contrast, animal housing is usually ventilated during all weather, with the ventilation rate modulated depending on inside air temperature. Fresh air is needed during even the coldest weather to remove moisture, odors, and aerosols from the aerial environment within the building. Oxygen depletion is not normally a concern. The major danger of inadequate ventilation arises from stress and disease potential created by high levels of humidity, noxious gases, carbon dioxide and aerosols; and excessively high temperatures during warm weather.

6-2. Design Weather Data

Weather data has been collected in the United States and worldwide for many years. Air temperature data is always collected; precipitation, wind and solar data may be available. For animal housing design air temperatures usually suffice, but proper greenhouse design requires solar data in addition to air temperatures.

Mean coincident wet-bulb temperature is also important for agricultural building design, especially with regard to evaporative cooling system design. Mean coincident wet-bulb temperatures are the expected wet-bulb temperatures at the design dry-bulb temperature values.

Weather data may be presented in one of two ways. The traditional method has been to compile "design temperatures". Design temperatures are based (usually) on averaged data collected for at least 15 years, and frequently for 40 years or more. For example, a 97.5% summer design temperature for a location is the temperature not exceeded for more than 2.5% of the time during summer (June, July, and August). Outdoor air temperatures will be above the 5% winter design temperature for all but 5% of December through February.

Extensive weather design data is available, for example, in the *ASHRAE Handbook of Fundamentals*. If this information is not available in reference tables, it can usually be obtained from local utility companies, heating system contractors, and Cooperative Extension offices. Sample design temperatures are available in Appendix 6-1 and Figure 6-1, which contains maps of the 48

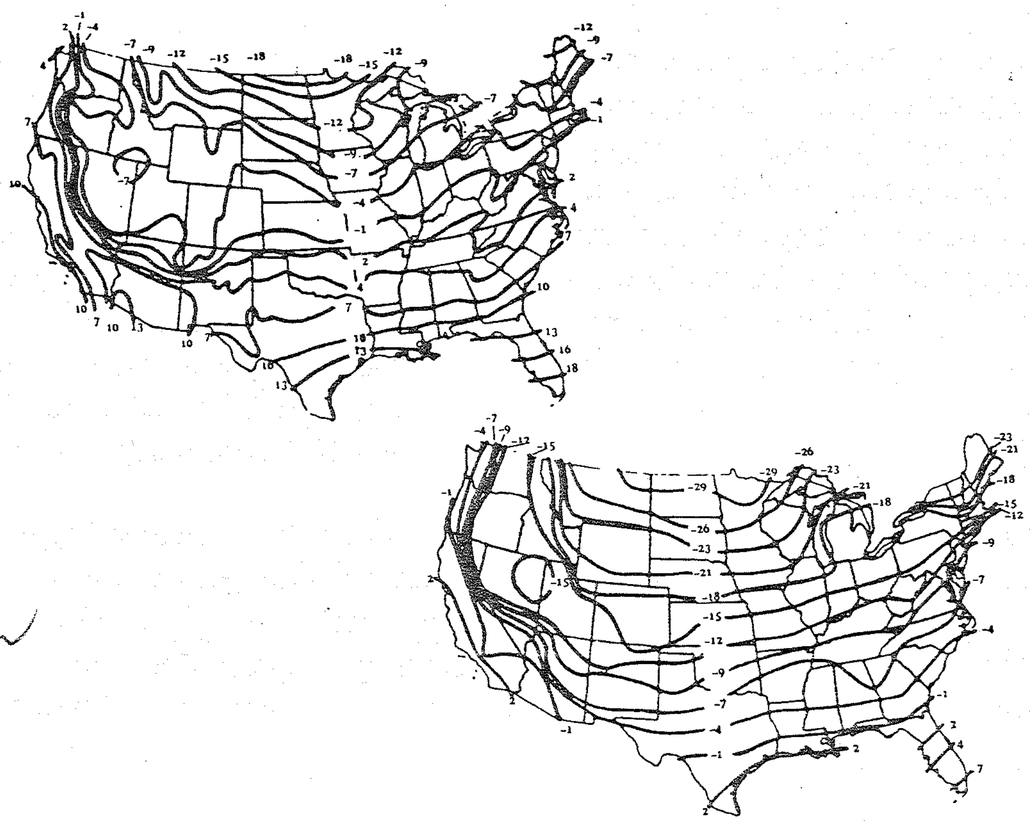

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Figure 6-1. Average daily air temperature, C, during January and 97.5% winter design temperature in the contiguous United States.

contiguous states of the United States with averaged and 97.5% winter design temperatures.

More recently, with the advent of computerized data collection methods, air temperatures have been presented as "bin data". In this technique, the number of hours during an average year that outdoor air temperature is within a certain range is determined, for temperature ranges spanning the local climate. For example, the number of hours the temperature is between - 20 and - 15 C, between - 15 and - 10 C, etc., up to the range where maximum summer temperatures are found. Bin data have been found useful for energy use calculations and are available from several sources: ASHRAE, The Government Printing Office (AFM 88-29), The National Climatic Data Center of NOAA, and engineering handbooks as examples. Much of this data, for example the ASHRAE and NOAA databases, are available on magnetic media format. Sample bin data can be found in Appendix 6-2.

6-3. Animal Housing Ventilation

6-3.1. General. Mature animals normally produce sufficient body heat to permit ventilation with no need for supplemental heat during typical cold weather, provided stocking density is at or near full capacity. For example, when weather is cold, a large dairy cow is approximately equivalent to a 1 kW heater in terms of her sensible heat production. If a building design does not result in adequate temperature at the minimum ventilation rate without supplemental heat, a design change should be considered with insulation added to make ventilation possible during cold weather. Supplemental heat should be considered as a final resort when housing mature animals because of its cost. Of course, young animals often require higher temperatures for comfort and health than do mature animals and do not produce sufficient body heat to maintain that temperature during cold weather. Supplemental heat is frequently necessary in those cases.

During warm weather, animal housing ventilation is normally designed to limit indoor air temperature rise above outdoor air temperature and evaporative cooling may be used where the outside air temperature is high and midday relative humidity is low. The choice of a maximum temperature rise must be tempered with a realization of the cost to keep the rise very small. As a close approximation, the temperature rise is halved when the ventilation rate doubles. Thus, the ventilation rate grows geometrically while the added benefit diminishes at each step. If the ventilation rate is X at an 8 K rise, it is 2 X for a 4 K rise, 4X for a 2 K rise, 8 X for a 1 K rise, etc. To increase ventilation from 4X to 8X to attain a benefit of only one degree may be very costly. Field experience has shown the practical minimum air temperature rise is approximately 1.5 to 2 K during the warmest weather and likely should be no more than 4 K unless the ventilation air is cooled such as by evaporative cooling.

6-3.2. Maximum Ventilation Rate. The situation of no evaporative cooling will be examined first. The sensible energy balance, Equation 5-10, applies,

$$q_s + q_m + q_{so} + q_h = \Sigma UA(t_i - t_o) + FP(t_i - t_o) + 1006\rho\dot{V}(t_i - t_o) + q_e.$$

Maximum ventilation rates are based on warm weather conditions, as a consequence, q_h will be zero. Also, for simplicity and to agree with animal heat production data in Appendix 5-1, the terms q_s and q_e will be combined into a single term, q_s. This assumes animal heat production data will be based on confinement conditions. If not, a correction must be made. As will be demonstrated in a later section, temperature control dictates the maximum ventilation rate during warm weather. If ventilation is adequate to control temperature during warm weather, it is adequate to control moisture and other aerial contaminants.

The sensible energy balance can be rearranged to calculate the ventilation rate, $\dot{V}$, m³/s,

$$\dot{V} = \frac{q_s + q_m + q_{so} - (\Sigma UA + FP)(t_i - t_o)}{1006\rho\,(t_i - t_o)}. \qquad (6\text{-}1)$$

Equation 6-1 has the form shown in Figure 6-2 (the line for temperature control) where the ventilation rate is a function of outside air temperature and other parameters are unchanged.

During early afternoon, when outside air temperature is highest, the sun is high in the sky and solar entry through glazing is often limited and usually is neglected. Heat from mechanical sources is often small during the middle of the day. Lights, for example, would not be on. Thus, it is common practice to eliminate q_m and q_{so} from the sensible energy balance. However, a designer should always be aware of the assumptions contained in that approach, and be prepared to challenge the validity of the assumptions for each specific design situation. Solar heat, heat from lights, etc., if significant, must be calculated and added to the energy balance.

The air temperature at which to determine animal sensible heat production is not immediately obvious. As indoor air temperature rises, sensible heat

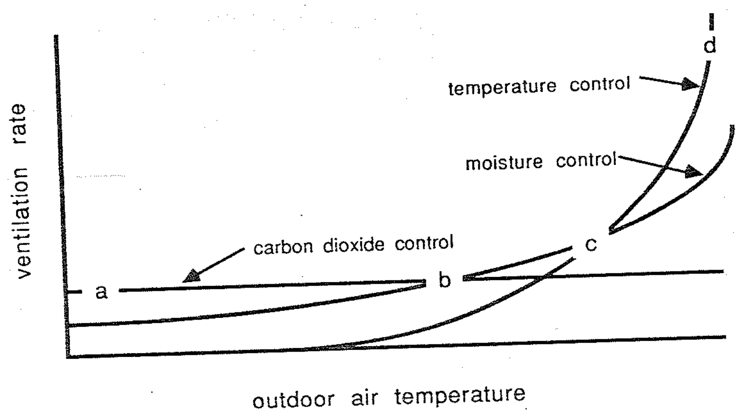

Figure 6-2. Example ventilation graph for temperature (c-d), moisture (b-c), and carbon dioxide (a-b) control in animal housing.

production from animals drops as metabolic heat production is reduced in response to heat stress, and a greater fraction is dissipated as latent heat. This is a negative feedback mechanism which assists in limiting the temperature rise of ventilating air during the hottest weather. An implication is that it may be more logical to calculate maximum ventilation rates based on warm summer conditions, rather than the hottest weather (not summer high temperature design conditions). t_o

A criterion to choose a design temperature is to use the temperature at which production from the animals begins to decrease significantly. However, there is a rational basis for arguing that the design temperature should equal the set-point temperature which activates the highest ventilation stage. In this way, the temperature rise chosen to calculate the maximum ventilation rate will not be exceeded at any time during which the highest ventilation stage is activated. (As the air temperature rises, the sensible heat production decreases.) There is a symmetry in calculating the maximum ventilation rate based on the highest thermostat setpoint, and having that occur before the onset of heat stress. t_i

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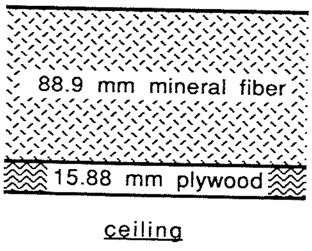
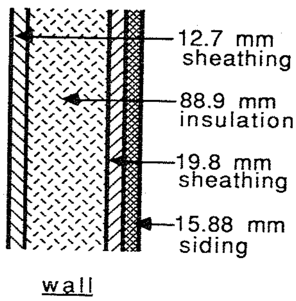
Example 6-1

Problem: Determine the maximum ventilation capacity to be installed in the following mechanically ventilated dairy barn. The barn is to house 100 milking cows (averaging 570 kg weight) in a zero pasture operation. The barn is 13 m wide and 80 m long with 3 m high sidewalls. It is to be located at an elevation of 1000 m.

The ceiling is sheathed on the lower side with 15.88 mm plywood and is insulated with 88.9 mm of mineral fiber blanket. The walls are sheathed on the inside with 12.7 mm plywood, insulated with 88.9 mm of mineral fiber blanket, sheathed on the outside with 19.8 mm thick vegetable fiberboard, and sided with 15.88 mm plywood.





There are two overhead panel doors (44 mm thick with 11 mm thick panels) and each door is 2.5 m high and 3.5 m wide. Double glazed windows are used (6 mm thick airspace, no storm sash) and total window area is 20 m^2.

Solution: The first design decision is to determine an indoor air temperature on which to base the design. Appendix 5-1 contains animal heat production data and for dairy cows the temperature range for which data are available ends at 27 C. Milk production drops rapidly at temperatures above 27 C (ASAE Data D270.4) although summer weather design conditions in much of the United States can be significantly above 27 C. For convenience and to follow the concept discussed above of designing summer ventilation rates for moderate, not extreme, temperature, the barn will be assumed at 21 C. A moderate relative humidity of 50% will be assumed (and can be adjusted by knowledge of the climate for which the barn is being designed). Program PLUS can be used to determine air density, which is 1.04 kg/m^3 (assuming an exhaust ventilation system and air density measured at indoor conditions).

Sensible heat production from dairy cows at 21 C is 1.1 W/kg, thus, total animal heat production will be

$$q_s = (100 \text{ cows})(500 \text{ kg/cow})(1.1 \text{ W/kg})(570 \text{ kg} / 500 \text{ kg})^{0.734}$$
$$= 60.6 \text{ kW}.$$

The total thermal conductance, ΣUA, of the barn will be obtained using program RVALUE for convenience. Adjustment factors for wood framed windows (assumed to be used) range from 0.90 to 1.0 for double glass; 0.95 will be assumed. This changes the R-value for the windows to (0.29 m^2 K/W)/0.95 (see Appendix 4-2 for window data), or 0.31 m^2 K/W. An "Other" type of window will be specified in RVALUE with this R-value. Door area is 17.5 m^2. The wall framing factor is not specified but the wall appears to be framed with 38 x 90 mm (2 x 4) lumber, thus, the factor will be approximately 20%, the number which should be used in RVALUE.

When these data are used in the program, a building ΣUA value of 784. 8 W/K is obtained. (It would be a useful exercise to use RVALUE to obtain ΣUA and even more useful to try the calculation by hand.)

No perimeter insulation was specified and construction of dairy barn floors is generally slab on grade. From the data in Section 4-6, a perimeter factor of 1.5 W/mK will be chosen (moderate winter severity). The perimeter is 186 m, thus, FP is 279.0 W/K. The temperature difference, t_i - t_o, is chosen to be 3 K, thus, the ventilation rate (Equation 6-1) is

$$\dot{V} = \frac{60{,}600 \text{ W} - (784.8 \text{ W/K} + 279.0 \text{ W/K})(3 \text{ K})}{(1006 \text{ J/kgK})(1.04 \text{ kg/m}^3)(3 \text{ K})}$$

1063.8 W/K

t_i t_o

21-18

$$= 18.3 \text{ m}^3\text{/s}.$$

Note an important implication of this calculation. Nothing has been stated about the actual summer weather. Whether the region where the barn is to be built has cool, moderate, or hot summers is not known. However, by the reasoning stated above, this should not matter unless the outdoor air temperature remains always above 18 C(21 C-3 K).

If the summer design temperature is greater than 18 C, the temperature difference from inside the barn to outside will be less than 3 K, but ventilation will not increase no matter how hot the weather becomes. Should it? Not for temperature control, for if the ventilation rate were even to double the gain would be less than 1.5 K at outside air temperatures above 18 C.

This approach does not agree totally with field recommendations. The calculated ventilation rate of 18.3 m^3/s agrees with field recommendations (on a per-cow basis) for climate zones having moderately hot summer weather, but the lack of ventilation rate dependence on climate does not.

The ventilation rate calculation also demonstrates the importance of ventilation for temperature control. Animal heat production is 60.6kW, and ($\Sigma UA + FP$) is 1063.8 W/K. When the temperature difference is 3 K, heat loss through the building's structural cover is (1063.8 W/K)(3 K) = 3.2 kW, which is less than 5.3% of the animal heat production. The remainder of the heat is removed by ventilation. As a final comment, there is precedent for ignoring the perimeter heat loss for warm weather calculations, for the soil outside buildings is normally quite warm due to solar heating.

When evaporative cooling is used for environmental modification, the sensible energy balance as already presented applies. However, ventilation air no longer enters at the outside air temperature. It enters at the cooled temperature, t_{oe}, the evaporatively modified outdoor air temperature. The calculation of ventilation rate changes to

$$\dot{V} = \frac{q_s + q_m + q_{so} - (\Sigma UA + FP)(t_i - t_o)}{1006\rho\,(t_i - t_{oe})} \qquad (6\text{-}2)$$

Evaporatively cooled air is significantly cooler than outside air, thus, the maximum ventilation rate will be lower.

The choice of a design criterion is not limited to maintaining the rise of temperature of the ventilating air to some prescribed limit. That may be the criterion, or the indoor air temperature itself may be prescribed and analysis completed to determine whether the limit can be reasonably attained. Obviously, the design indoor air temperature must be above the temperature to which the ventilating air can be cooled.

In many cases, using a design criterion of building air temperature rather than air temperature rise would be preferred. Using the criterion of a temperature rise is an admission that conditions best for the animals cannot be maintained and the best that can be done is to maintain air temperature near outdoor conditions. When air can be cooled, the possibility is greater that air temperatures which promote health and productivity may be possible.

Example 6-2

Problem: The dairy barn described in Example 6-1 is located in a region of hot summer temperatures and a dry climate. Evaporative cooling has been determined to be desirable to maintain high milk production during the summer. (Note: It is not necessarily true that a zero-pasture housing system would be best for such a climate; the example is used for illustration only. In warm climates, dairy cattle are often kept outdoors in pens or corrals, with sun shades provided to limit heat stress.)

The summer design weather condition (97.5% probability) is 36 C dry bulb temperature, with a mean coincident wet bulb temperature of 18 C. The 97.5% probability wet bulb temperature is 20 C. The evaporative cooling system is expected to be 75% efficient.

Determine the maximum ventilation capacity so indoor air temperature will not go over 27 C (the beginning of heat stress) for more than a few hours per year.

Solution: The decision of which weather condition to use for design is complicated by the need to choose two conditions, dry bulb and wet bulb temperature. The stipulation that the design temperature not be exceeded for more than a few hours per year will be assumed to mean design for the 97.5% probability level. It appears the 97.5% condition for dry bulb temperature has a coincident wet bulb temperature which is close to the 97.5% probability wet bulb temperature. When the wet bulb temperature is 20 C, the dry bulb temperature is below 36 C, thus, less cooling can be expected but less will be needed. Lacking other weather information, the conditions of 36 C dry bulb and 18 C wet bulb temperatures will be chosen, which is an 18 K wet bulb depression.

If the evaporative cooling system is 75% efficient, actual cooling will be 0.75(18) = 13.5 K. At design conditions, ventilating air will enter the building at 22.5 C, while the outdoor air temperature is 36 C. The relative humidity inside the building at this condition is not known but will be assumed for now to be approximately 60%. Thus, the density of indoor air will be 1.03 kg/m^3 (at 27 C, 60% relative humidity, 1000 m elevation).

It will be assumed that solar entry through windows is small enough to be ignored, and during the day there will be little heat generated from lights, etc. The ventilation rate to achieve design conditions with these assumptions is

$$\dot{V} = \frac{q_s - (\Sigma UA + FP)(t_i - t_o)}{1006\rho\ (t_i - t_{oe})}$$

The herd's sensible heat production at 27 C will be 33 kW, thus,

$$\dot{V} = \frac{33{,}000 \text{ W} - (1063.8 \text{ W/K})(27 \text{ C} - 36 \text{ C})}{(1006 \text{ J/kgK})(1.03 \text{ kg/m}^3)(27 \text{ C} - 22.5 \text{ C})}$$
$$= 9.1 \text{ m}^3\text{/s}.$$

Evaporative cooling has permitted nearly a 40% decrease of the maximum ventilation rate.

Excessively high humidity may be a concern when using evaporative cooling. The relative humidity expected in the barn can be determined. Ventilation air enters the space with a dry bulb temperature of 20 C, a wet bulb temperature of 18 C, a relative humidity of 66% and a humidity ratio of 0.012924 kg/kg. At 27 C, a 500 kg cow produces 0.50 mg/kg-s (see Appendix 5-1) of water vapor. The herd will produce

$$m_p = (100 \text{ cows})(0.50 \text{ mg/kg-s})(500 \text{ kg/cow})(570 / 500)^{0.734}$$
$$= 0.0275 \text{ kg/s}.$$

The ventilation rate is 9.1 m^3/s, air density is 1.03 kg/m^3, thus, the change of humidity ratio is

$$\Delta W = m_p / m_{air}$$
$$= 0.0275 \text{ kg/s} / [(9.1 \text{ m}^3\text{/s})(1.03 \text{ kg/m}^3)]$$
$$= 0.002936 \text{ kg/kg}.$$

Air at a mixed condition within the barn will have a humidity ratio of 0.012924 + 0.002936 = 0.015860 kg/kg. At 27 C, the relative humidity will be 62%. This should not stress the animals and is close to the assumption of 60% relative humidity which was made to determine air density. Although the animals add moisture to the air, their addition of sensible heat lowers the relative humidity of the cooled air. Note that, of course, the results and conclusions require the assumption that data for animal heat and moisture production apply at the conditions existing in the barn and the level of feeding and milk production expected.

6-3-3. Minimum Ventilation Rate. To determine minimum ventilation rates for animal housing, factors in addition to air temperature must be considered and criteria arising from the various factors may conflict. Humidity and other air contaminants may increase when the ventilation rate is low to the point where they dictate the minimum ventilation rate. If that happens, it may no longer be possible to maintain air temperature at the desired level without supplemental heat.

The alternative is to permit air temperature to drop. If air temperature is not far below the optimum level for many hours per year, or for long periods of time, production likely will not be affected. Of course, indoor air temperatures below freezing generally are not permitted, a practice which prevents damage to water pipes, etc. A design minimum indoor air temperature several degrees above freezing is typically chosen for safety. Even at 5 C at the thermostat, thermal stratification and other temperature nonuniformities may produce regions in the barn close to freezing.

The technique to determine which design criterion dictates the minimum ventilation rate is to develop a Ventilation Graph. The graph describes the required ventilation rate as a function of outdoor temperature according to several criteria, for example, temperature control, humidity control, and carbon dioxide control. Sensible energy and mass balances are used to determine the relationship between ventilation and outside air temperature based on these criteria.

To calculate the minimum ventilation rate based on temperature control, contributions of solar heat and heat from lights, etc., are usually discounted – design is for the worst condition when only animal heat is available to warm the air. The energy balance is as before

$$\dot{V}_{temp} = \frac{q_s - (\Sigma UA + FP)(t_i - t_o)}{1006\rho\,(t_i - t_o)} \tag{6-3}$$

The moisture balance can be used to determine ventilation for moisture control,

$$\dot{V}_{H_2O} = m_p/\rho_{air}(W_i - W_o), \tag{6-4}$$

and a mass balance can be used to determine ventilation for carbon dioxide control,

$$\dot{V}_{CO_2} = (CO_2)_p/((CO_2)_i - (CO_2)_o), \tag{6-5}$$

where $(CO_2)_p$ is the volumetric rate of carbon dioxide production, and $(CO_2)_i$ and $(CO_2)_o$ are the volumetric concentrations of carbon dioxide in the indoor and outdoor air, respectively.

The ventilation rate for carbon dioxide control does not vary as a function of outdoor air temperature. Ventilation for temperature control obviously will, and ventilation for moisture control will also vary indirectly as a function of outdoor temperature. A frequently chosen ambient condition for design is a constant relative humidity. In this event, W_o will vary as the outdoor air temperature changes, thus, $\dot{V}_{H_2O}$ must also vary.

A ventilation graph is obtained by calculating the required ventilation rate at a series of outdoor temperatures ranging from the coldest expected for the climate, up to nearly the design indoor temperature, and graphing the results as indicated in Figure 6-2. The graph is drawn for a fixed indoor air temperature, the winter indoor design temperature.

Because the carbon dioxide level in ambient air is not a function of temperature, the line for carbon dioxide control is horizontal on the graph. Although very low at low outdoor temperatures, ventilation for moisture control is always positive because cold air is very dry. The shape of the moisture control curve reflects the shape of the psychrometric chart.

Ventilation for temperature control may be calculated to be negative when it is very cold outdoors. This should be interpreted to mean that when it is very cold, heat loss through the structural cover is so great that the design indoor air temperature cannot be maintained even if there is no ventilation or air infiltration. Supplemental heat is needed.

As outdoor air temperature approaches indoor air temperature, the required ventilation rate becomes very large due to the small value of $(t_i - t_o)$ in the energy balance. This does not indicate the ventilation rate must become increasingly great as the temperature difference becomes small. Instead, the indoor air temperature should be permitted to rise above the winter design temperature (which is generally near or at the lowest desirable temperature).

At a given outdoor air temperature, the criterion which dictates the minimum ventilation rate is that which produces the maximum ventilation rate at that outdoor air temperature. In Figure 6-2, carbon dioxide control dictates when it is very cold outdoors (line ab), moisture control dictates for moderately cold temperatures (line bc), and finally temperature control dictates (line cd). Ideal control of the ventilation rate would follow line abcd on the graph. However, typical fan control is not so finely modulated as to permit this. Instead, common practice is to choose the ventilation rate at point c as the minimum ventilation rate.

When outdoor air temperature is below that at point c, moisture and carbon dioxide criteria are satisfied – they are based on maximum levels. The temperature criterion is not satisfied – it is based on a minimum temperature level. However, most mature farm animals are not affected adversely by air temperatures below those considered optimum. The animals may eat more to maintain production and body temperature, but health, vitality, and productivity are not likely to be affected. There are obviously limits where health and production will be affected, but with typical animal stocking densities there will be sufficient body heat to prevent severely cold indoor air temperatures. If there is not sufficient body heat, supplemental heat may be required, the building may be redesigned for better insulation qualities, or management practices may be changed to increase stocking density.

The temperature at which point c occurs is a function of many design parameters, the insulation value of the structural cover of the building being one. At the minimum ventilation rate, ventilation is still a dominant mechanism of heat loss from the airspace but it is the same order of magnitude as loss through the structural cover. Thus, insulation in the walls, ceiling, and perimeter, and the choice of doors and glazings can affect the temperature criterion line and the location of point c. This can be an important design parameter if the outdoor air temperature at point c is sufficiently high that indoor air temperature cannot be maintained for a significant fraction of the winter.

Example 6-3

Problem: Reconsider the dairy barn described in Examples 6-1 and 6-2. Determine the minimum ventilation rate if the indoor air temperature desired for cold weather conditions is at least 10 C, with a relative humidity no more than 70% and a carbon dioxide concentration no more than 5000 ppm. Outdoor air temperature can be as low as - 30 C.

Solution: A ventilation graph will be used to solve this problem. Program PLUS will be used to determine psychrometric properties of air.

Data in Appendix 5-1 for dairy cows at 10 C are: total heat production is 2.2 W/kg, sensible heat production is 1.5 W/kg, and moisture production is 0.28 mg/kg-s, each based on a 500 kg animal unit. Scaled to 100 cows averaging 570 kg, heat and moisture production rates for the entire barn are:

$$\begin{aligned} q_{total} &= (100 \text{ cows})(2.2 \text{ W/kg})(500 \text{ kg/cow})(570 \text{ kg} / 500 \text{ kg})^{0.734} \\ &= 121 \text{ kW}, \end{aligned}$$

$$\begin{aligned} q_{sensible} &= (100 \text{ cows})(1.5 \text{ W/kg})(500 \text{ kg/cow})(570 \text{ kg} / 500 \text{ kg})^{0.734} \\ &= 83 \text{ kW, and} \end{aligned}$$

$$\begin{aligned} m_{H_2O} &= (100 \text{ cows})(0.28 \text{ mg/kg-s})(500 \text{ kg/cow})(570 / 500)^{0.734} \\ &= 0.0154 \text{ kg/s}. \end{aligned}$$

The simplest line to determine on the ventilation graph is for carbon dioxide control. Ambient carbon dioxide will be assumed at 345 ppm. The conversion from total heat production to carbon dioxide production (Section 5-6.1) yields the carbon dioxide production rate,

$$\begin{aligned} (CO_2)_p \quad \dot{V}_{CO_2} &= (121{,}000 \text{ J/s})/(24{,}600 \text{ J/L}) = 4.92 \text{ L/s} \\ &= 0.00492 \text{ m}^3\text{/s}. \end{aligned}$$

The change permitted in the carbon dioxide concentration is

$$\begin{aligned} (C_{CO_2})_{in} - (C_{CO_2})_{out} &= 0.005 \text{ m}^3/\text{m}^3 - 0.000345 \text{ m}^3/\text{m}^3 \\ &= 0.004655 \text{ m}^3/\text{m}^3. \end{aligned}$$

The ventilation rate for carbon dioxide control is

$$\dot{V}_{CO_2} = (0.00492\ m^3/s)/(0.004655\ m^3/m^3)$$

$$= 1.06\ m^3/s.$$

The ventilation rate for moisture control will depend on the outdoor relative humidity, especially as the outdoor air temperature approaches that of the indoor air. For design, a condition of high (but reasonable) outdoor relative humidity is more conservative. The climate where the barn is located is stated to be dry, but during cold weather relative humidities are likely to be high at times in most climate zones. For this design, an outdoor relative humidity of 80% will be assumed.

Indoor conditions of 10 C and 70% relative humidity result in a humidity ratio of 0.006076 kg/kg (recall that this is at 1000 m elevation). Moisture production is 0.0154 kg/s, thus, ventilation for moisture control is calculated from

$$\dot{V}_{H_2O} = (0.0154\ kg/s)/\left[\rho_{air}(0.006076 - W_o)\right],$$

where W_o is the humidity ratio of the outdoor air. At 10 C and 70% relative humidity, air density (ρ_{air}) is 1.10 kg/m³. The following table contains humidity ratio and ventilation rate data for the outdoor temperature range from - 25 C to 5 C.

Outdoor Temperature	Humidity[a] Ratio, kg/kg	Ventilation Rate, m³/s
-25 C	0.000353	2.45
-20	0.000577	2.55
-15	0.000925	2.71
-10	0.001457	3.03
-5	0.002256	3.66
0	0.003441	5.31
5	0.004924	12.15

[a] at 80% relative humidity, 1000 m elevation

The ventilation rate for temperature control is determined by the sensible energy balance for the barn, as used in Example 6-2, changed to reflect conditions for this example.

$$\dot{V}_{temp} = \frac{83{,}000\ W - (1063.8\ W/K)(10\ C - t_o)}{(1006\ J/kgK)(1.10\ kg/m^3)(10\ C - t_o)}$$

The following table contains data for the ventilation rate as a function of outdoor air temperature.

Outdoor Temperature	Ventilation Rate, m³/s
-25 C	1.18
-20	1.54
-15	2.04
-10	2.79
-5	4.04
0	6.54
5	14.04

The ventilation rates for carbon dioxide, moisture, and temperature control are shown on the ventilation graph in Figure 6-3. Carbon dioxide never governs the minimum ventilation rate in this example. The lines for moisture control and temperature control cross at a ventilation rate of approximately 3.3 m^3/s, at an outdoor air temperature of approximately - 8 C. Thus, the installed minimum ventilation rate would be 3.3 m^3/s. This is high for a minimum ventilation rate for a dairy barn, due to the design criterion of only 70% relative humidity. It would be a useful exercise to assume 80% relative humidity and determine the resulting minimum ventilation rate.

When outdoor air temperature is below - 8 C, the temperature control criterion will be violated, resulting in a barn temperature lower than the design value. It would be a useful exercise to determine how low the barn temperature would fall without supplemental heat and how much supplemental heat, if any, would be needed to prevent freezing in the barn.

The sensible energy balance can again be used, although the question of what the barn temperature will be must be determined. Animal heat production is a function of air temperature and air temperature is influenced by the heat production. As the temperature falls, heat production increases, which is a negative feedback mechanism acting to help prevent freezing in the barn. If animal heat production is expressed as a second order polynomial (see section 6-4.1), a quadratic equation results which can be solved to determine the indoor air temperature in closed form.

Before completing this example, a note regarding humidistats is in order. Humidity control in this example was achieved by indirect means – based on assumed conditions and assumed heat and moisture production data applied to the cows. Direct measure of humidity was not considered; humidity measurement in agricultural buildings is not a general practice. The reason can be traced to a lack of suitable humidity sensors. Agricultural buildings often are at high relative humidities and provide aerial environments with high dust levels at times and trace amounts of many corrosive gases. The environment is hostile to sensors and a humidity sensor which can retain calibration in these conditions has not yet been developed. A wet and dry bulb sensor is possible, but requires

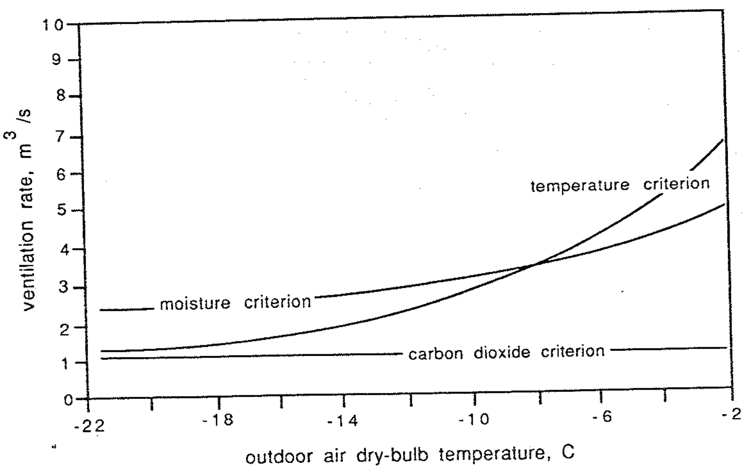


Figure 6-3. Ventilation graph for Example 6-3. Moisture and temperature criteria curves intersect at a ventilation rate of 3.3 m^3/s and an outdoor air temperature of -8 C.

frequent maintenance and careful use, needs often neglected in the press of daily activities.

6-4. Computerized Procedures to Determine Ventilation Rates

An automated procedure to draw ventilation graphs and determine minimum ventilation rates would be useful for design, for such a program would permit the designer to explore various design options and immediately see the results in terms of their effects on minimum ventilation rates.

The procedures used above to develop a ventilation graph could be implemented in a computer program to calculate and graph data such as in Figure 6-3. The one factor needed for the graphs is a means to calculate heat and moisture production from animals. Data such as in Appendix 5-1 can be used, with interpolation procedures implemented in a computer program to estimate production rates at temperatures intermediate to those in the appendix. Another approach is to approximate the data using a mathematical function which expresses heat or moisture production as a function of air temperature.

6-4.1. Polynomial Regression. Statistical regression procedures are used frequently in engineering and the biological sciences to describe how one parameter varies in response to changes in another. Data to describe the dependence may exist in table form, but for convenience an equation is preferable to express the relationship.

The convective heat transfer equations described in Chapter 3 were developed by the original researchers after they accumulated a large base of experimental data where heat transfer rates by convection were measured, along with other pertinent variables. Numerous regression models were attempted until one was found which provided the "right answer" for the ranges of values of the independent variables which had been considered. Regression analysis is a technique which can be used to express a dependent variable in terms of independent variables thought to influence the response of interest.

Regression models are not based on fundamental principles of the phenomenon under consideration. They are a means to approximate or describe the value of a dependent variable if values for all independent variables are known. Regression models can be expected to have no mathematical resemblance to the true dependence, but they provide adequate answers within a limited range of the independent variables. The limit depends on the width of the original range of data when the equation was developed.

Many forms of regression models are possible–linear, power curves, exponential functions, logarithmic functions, etc. Regression models based on polynomials can also provide suitable means to describe data points. While not best for all situations, polynomials provide an adequate approximation of animal heat and moisture production as a function of air temperature.

Program POLYNOM is provided to calculate the coefficients in a polynomial regression equation for Y as a function of X,

$$Y = a_0 + a_1X + a_2X^2 + a_3X^3 + ... \qquad (6\text{-}6)$$

The program is limited to no more than a sixth order polynomial, and no more than 100 data points. Two measures of fit are provided. The standard deviation of the data about the regression line is calculated and the data and regression equation can be graphed so the regression adequacy can be evaluated by eye.

Printed copies of the graph of data and the regression equation can be obtained using the PrintScreen key. Note: Before a copy can be obtained, the computer must be in Graphics mode.

Example 6-4

Problem: Develop a polynomial to express sensible heat production (SHP, W/kg) from dairy cows as a function of air temperature (C).

Solution: The basic data are in Appendix 5-1.

Air Temperature	SHP W/kg
-1 C	1.9
10	1.5
15	1.2
21	1.1
27	0.6

Program POLYNOM will be used. There are 5 data points. The maximum possible order of the regression is 4 (the constant term plus 4 coefficients). The coefficients for each order of regression are:

Order	a_0	a_1	a_2	a_3	a_4
0	1.260				
1	1.897	-0.04425			
2	1.860	-0.03074	-5.268E-4		
3	1.845	-0.05708	2.504E-3	-7.693E-5	
4	2.006	0.07884	-2.546E-2	1.540E-3	-2.874E-5

Graphs of the regressions for first through fourth order are in Figure 6-4. The standard deviations for the four orders of regression are: 0:0.483; 1:0.105; 2:0.106; 3:0.124; 4:0.000.

Standard deviations provide a limited measure of goodness of fit. Order 1 (linear regression) is obviously much better than order 0 (only the average). The standard deviation for order 2 shows no improvement (in fact, a very slight increase of the standard deviation), and order 3 is an even poorer fit. Order 4 provides a perfect fit of the five data points, but that should not be taken to mean 4th order is the best means to represent the data. Graphs of the equations for orders 1 to 4 are in Figure 6-4. The fourth order curve, although it passes through every data point, oscillates between the first and second data point in a way inappropriate to describe how the data would be expected to behave. Thus, for this example, a first order equation would be appropriate.

6-4.2. Program VENTGRPH. It is necessary for an engineer designing environmental control systems for barns to know how to construct a ventilation graph. However, to do so repeatedly is laborious, and the effort involved quickly subdues a designer's enthusiasm to explore the many possible options in modifying the initial design of a barn. Program VENTGRPH is provided for such explorations.

All heat and moisture productions from animals are expressed as second order polynomials in VENTGRPH. Although the results of Example 6-4 indicated a first order regression was appropriate to express sensible heat production from dairy cows, a second order polynomial was more appropriate for other data in Appendix 5-1 and was the form adopted for general use in VENTGRPH.

The default data incorporated into VENTGRPH are based on data in Appendix 5-1, but an opportunity is presented to change the data if desired. Also, an "other" category exists to permit ventilation graphs to be constructed for

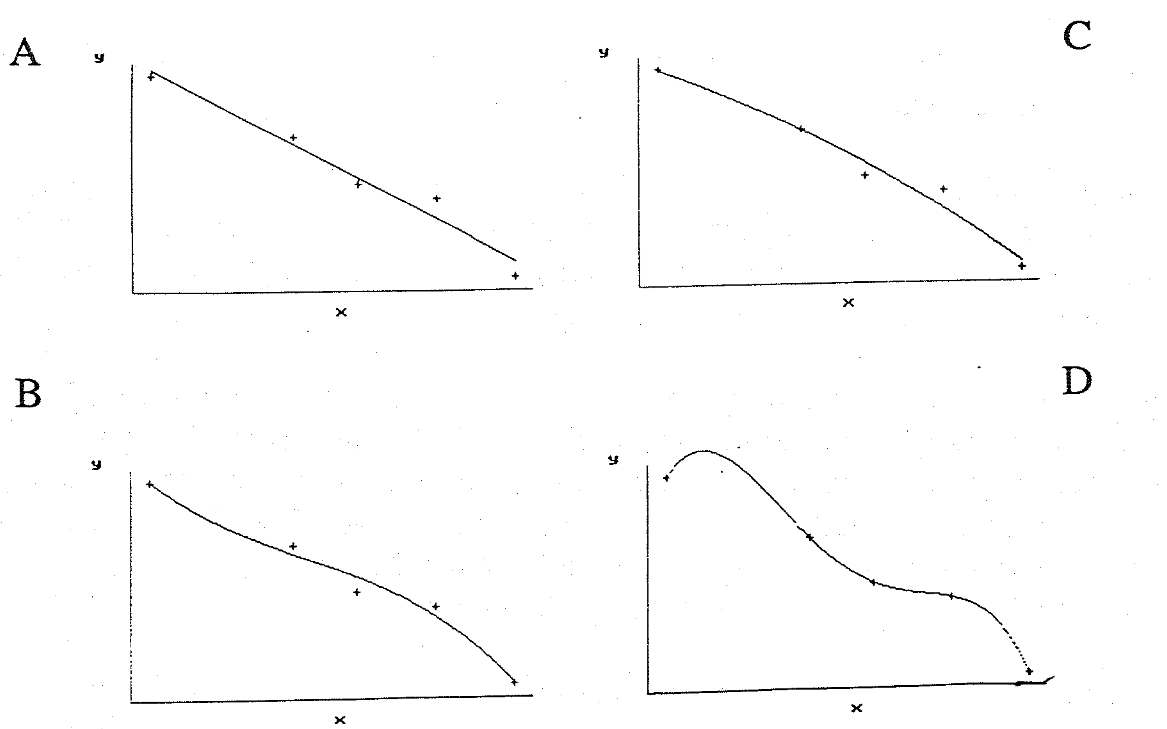


Figure 6-4. Data and regression curves for Example 6-4: (a), first order; (b), second order; (c) third order; and (d), fourth order.

animals other than dairy cows, swine, broilers, and laying hens. To use the "other" category, second order polynomials to approximate sensible and latent heat productions must first be determined.

Example 6-5

Problem: Use program VENTGRPH to develop a ventilation graph for the barn in Example 6-3 and explore the effect of the insulation value of the structural cover on the minimum ventilation rate which should be installed in the barn. Determine the minimum ventilation rate, m^3/s, as a function of the barn's average U value, $W/m^2 K$, over the range of U from 0.1 to 1.0.

Solution: Program VENTGRPH can be used repeatedly to determine the minimum ventilation rate at the various insulation values. Data from previous examples of this chapter will be used as input. The air pressure is 89.874 kPa. Indoor air temperature is 10 C and the relative humidity and carbon dioxide levels are 70% and 5000 ppm, respectively. There are 100 cows averaging 570 kg mass. The perimeter of the barn is 186 m and the perimeter heat loss factor is 1.5 W/mK.

The overall UA value is 784.8 W/K, and the area of the ceiling and walls is: $(13)(80) + (186)(3) = 1598\ m^2$. Thus, the average U value of the structural cover is: $784.8/1598 = 0.49\ W/m^2 K$. This is the parameter which will be varied from 0.1 to 1.0, and the result will be a variation in ΣUA from 159.8 to 1598 W/K. A change of this magnitude spans the range from no insulation and very light construction, to a level of insulation which exceeds that normally installed in animal housing.

When program VENTGRPH is used, the minimum ventilation rate (intersection of the moisture/temperature curves) is displayed on the screen. If carbon dioxide controls, its corresponding ventilation rate must be estimated from reading the screen.

Using VENTGRPH repeatedly, the data in the table below can be accumulated where the outdoor air temperature at crossing is the temperature at which the moisture criterion and temperature criterion curves intersect. The carbon dioxide criterion never governs in this example.

As an example of the output from VENTGRPH, in Figure 6-5 is the ventilation graph for the original example, with a ΣUA value of 784.8 W/K. The intersection of the moisture and temperature criteria lines is at approximately 3.5 m^3/s. This is slightly above the result in Figure 6-3. The difference arises from the approximation of animal heat production data by using a polynomial, and graphing the lines more smoothly in Figure 6-5.

The trend for the minimum ventilation rate as a function of the overall building

UA value may appear counter-intuitive at first. Without knowledge of the form of the ventilation graph, one might think that as the ΣUA value increases, more heat will be lost from the building by conduction through the structural cover and less loss can as a consequence be tolerated through ventilation, resulting in a lower minimum ventilation rate.

The data show the opposite is true. Why is this? The reason lies in the shape of the moisture criterion curve. As the ΣUA value increases, the temperature criterion can be met only at a higher outdoor air temperature. At the higher outdoor temperature, the air is less dry (recall the assumption of fixed outdoor relative humidity). When the outdoor air is less dry, more ventilation is needed for moisture control and the moisture control criterion must be satisfied. The result is a higher minimum ventilation rate. What happens to temperature control? The temperature criterion is still met, but at a higher outdoor air temperature.

The crossing of the moisture and temperature control criteria lines at a lower outdoor air temperature is an important, and subtle, result of insulating barns. A first approach to determining the proper insulation level might include only two criteria: prevent condensation on the inside surface of the wall, and conserve sufficient energy to prevent the barn from freezing.

Building U value W/m^2K	Building ΣUA value, W/K	Minimum Ventilation Rate, m^3/s	Outdoor Air Temperature at Crossing
0.1	159.8	2.9	-13 C
0.2	319.6	3.1	-11
0.3	479.4	3.2	-9
0.4	639.2	3.4	-8
0.5	799.0	3.6	-6
0.6	958.8	3.8	-5
0.7	1118.6	3.9	-4
0.8	1278.4	4.2	-3
0.9	1438.2	4.4	-2
1.0	1598.0	4.6	-2

The ventilation graph shows two additional advantages of insulation. A lesser minimum ventilation capacity may be installed and this should reduce the electricity cost to ventilate (how the savings can be estimated will be covered later in the text). In addition, the curves cross at a lower outdoor air temperature, thus, the temperature criterion will be violated for fewer hours per year. This could save a little feed and provide an environment within the thermoneutral zone of the animal population for a greater fraction of the year.

6-5. Staging Between Minimum and Maximum Ventilation Rates

To this point, procedures have been developed to calculate the required minimum and maximum ventilation rates for an animal barn. The next design step is to prescribe how the rate shall be varied through the range between.

One means to vary ventilation rates is to use variable speed fans, chosen to provide the maximum ventilation rate at full capacity and the minimum rate when some or all the fans operate at their minimum capacities. To reach the minimum rate, only a few fans may be needed. Variable speed fans have the advantage of continuous variation between their minimum and maximum ventilation rates. They have the disadvantage of losing their ability to resist wind effects when operated at less than full capacity. Fan characteristics will be explored more fully in a later chapter.

Single speed fan systems must be "staged" if they are to modulate between the minimum and maximum ventilation rates. This means, for example, one or more small fans are included to provide the minimum rate (and larger fans not operated). Middle sized and large fans are also installed, and activated as barn temperature rises and the need for ventilation increases. The barn's ventilation rate thereby changes by stages rather than continuously as would be the case with variable speed fans.

Frequently, the small fans continue to operate as the higher stages are needed; each higher stage is an addition of fans. The small fans frequently draw air from near the floor (the coldest air) to conserve a little heat during the coldest weather and the large fans draw air from near the ceiling to remove the warmest air during hot weather.

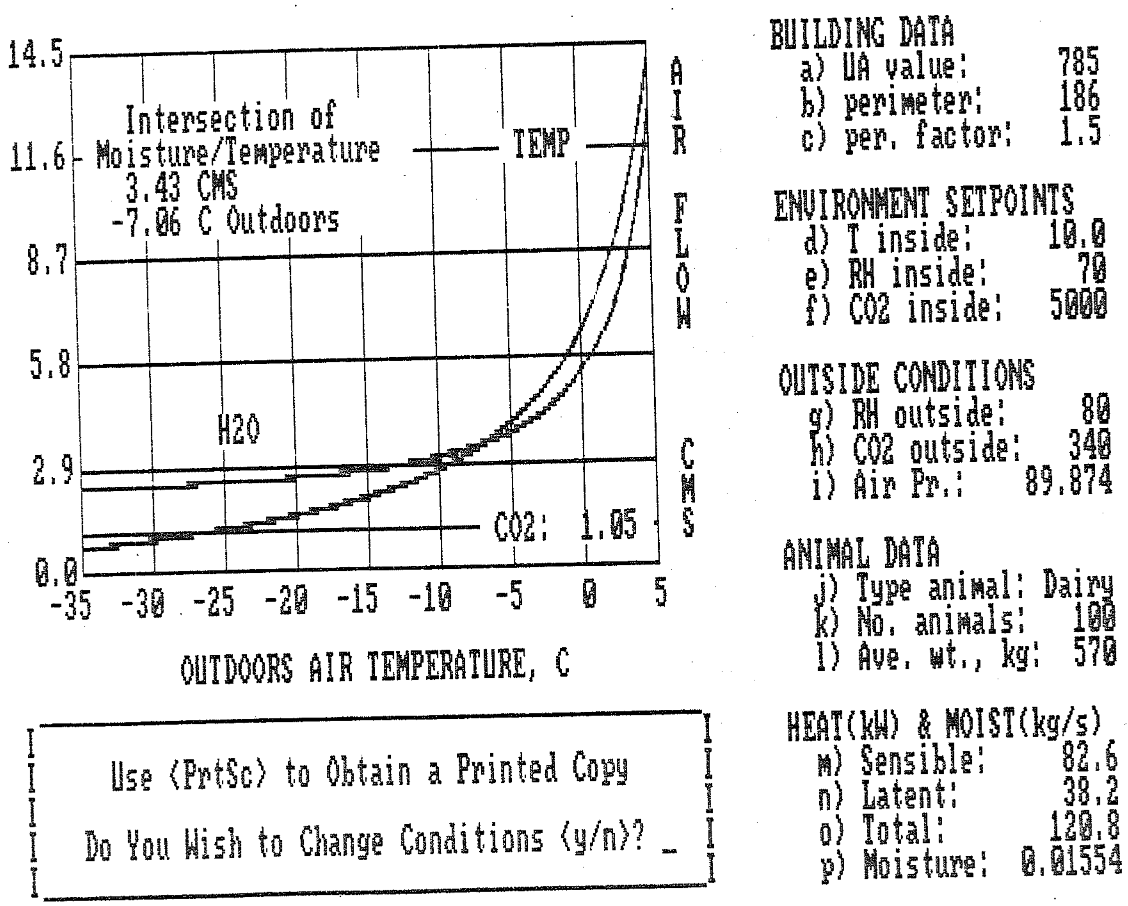


Figure 6-5. Results screen of program VENTGRPH with data and results of Example 6-5, and actual barn design (proposed).

Two factors must be considered when staging a fan system. One is to determine barn temperatures at which control moves from one ventilation stage to another – the control system setpoints. The other is to determine the magnitude of each stage between the minimum and maximum ventilation rates.

The choice of setpoints for ventilation stage changes should reflect the biological needs of the animals. What is best for dairy cows, for example, would be inappropriate for laying hens. A guide to determine what is best are data which reflect production and feed consumption as functions of temperature. For example, the thermoneutral zone for dairy cows centers around the range from 10 to 15 C, but production changes little over a wide temperature range centered about 10 to 15 C. Optimal production and feed consumption for laying hens occurs near 24 C, and is limited to a narrow temperature range (ASAE Data D270.4).

Data in Figure 6-6 can be used to estimate optimum temperature ranges for dairy cows, laying hens, and swine, and the effects of varying from those ranges. Control system setpoints are generally chosen to provide the minimum ventilation rate at the lower end of the optimum production range, and the maximum ventilation rate at the upper end. For example, one might choose 8 C as the lowest thermostat setpoint for Holstein dairy cows. When indoor air temperature is below 8 C, only the minimum ventilation rate for moisture (or perhaps carbon dioxide) control would be used. Above 8 C, the second stage of ventilation would be used. (Of course, a freeze protection would be used in the control sequence; all fans might be off at barn temperatures below 4 C, for example.) The highest stage of ventilation would be activated at the temperature where heat stress could affect production. For dairy cows, this would be approximately 24 C, although common practice is to activate the highest stage several degrees below 24 C.

The number of stages between minimum and maximum ventilation rates is left to the designer's discretion. There should be enough stages so ventilation rate steps from stage to stage do not cause extreme environment changes. For example, a system with only two stages would result in rapid oscillation

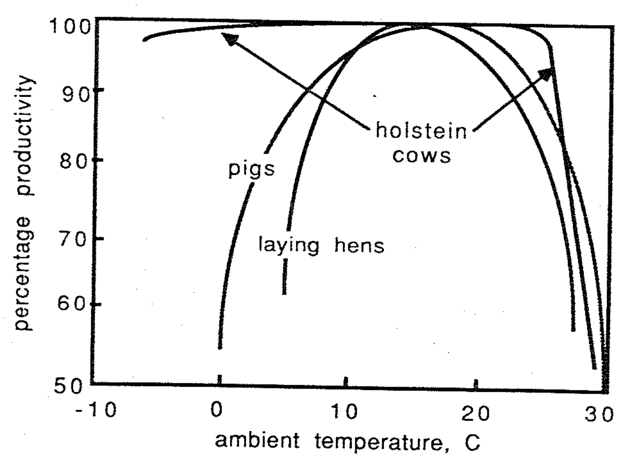


Figure 6-6. Productivity potentials of dairy cows, swine, and laying hens as a function of ambient air temperature (in still air with no solar effect).

between the stages during moderately cold weather and perhaps frequent, sudden chilling of the animals as the ventilation rate changes from the minimum to the maximum. Such a situation would be unsatisfactory. Experience has shown four stages to be a practical minimum. A practical maximum number of stages have been found to be six. More than six stages do not bring significantly better control.

The decision of whether four, five, or six stages are best depends on the total amount of ventilation needed, and availability of fan sizes. If the ventilation system is to be small, fewer stages are more practical to avoid using many small fans. Small fans are generally less energy efficient than large fans and installation costs for many small fans would be greater than for a few large fans.

The second important decision in developing a staging schedule is to specify the magnitude of each stage. The minimum and maximum stages are calculated as shown above. The number of stages can be decided by the size of the total system. Intermediate stages must then be specified.

A traditional method to size the stages is to divide the range between the minimum and maximum rates evenly. Another method is to specify the beginning stages as small steps, and the difference between the next to last and last stages as large.

In Figure 6-7a is a conceptual graph of the need for ventilation as a function of outdoor air temperature. If stages are determined by equal divisions of the ventilation rate range, there will be a large change of outdoor air temperature between stages 1 and 2 (Figure 6-7b). Also the change of ventilation between stages 1 and 2 will be large relative to the magnitude of ventilation at stage 1. When control causes ventilation to oscillate between stages 1 and 2 (outdoor air temperature is too warm to maintain stage 1 continuously and too cold to maintain stage 2), the large changes of ventilation rate could cause localized chilling and rapid changes of temperature within the barn. If stage 2 is only slightly greater than stage 1, these problems can be minimized.

One method to assure the early stages will be separated by small steps is to divide the ventilation rate range using equal divisions of outdoor air temperature, as shown in Figure 6-7c. Experience has shown that equal

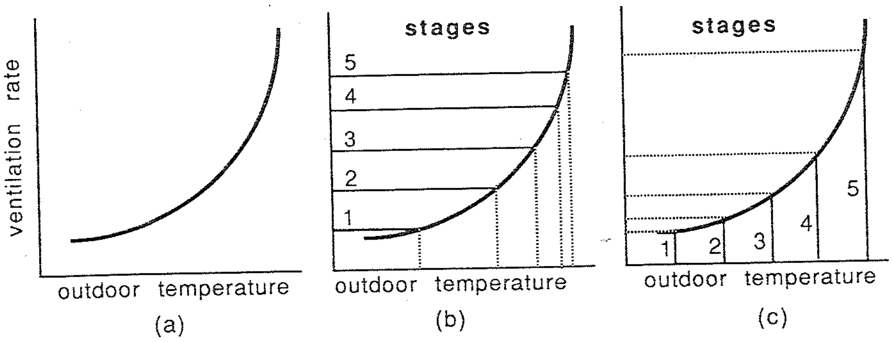


Figure 6-7. Ventilation rate as a function of outdoor temperature: (a) general curve to maintain fixed indoor air temperature, (b) with fan stagings based on equal increments of ventilation rate, and (c) with fan staging based on equal increments of outdoor temperature.

divisions of outdoor air temperature result in stages approximately as listed below in Table 6-1.

Each schedule in Table 6-1 begins with the minimum ventilation rate being 10% of the maximum. Procedures used above to determine minimum and maximum rates will not always result in the 1:10 difference. Depending on assumptions used for design, the ratio may be as small as 1:5. The actual schedule must therefore use the spirit of the data in Table 6-1 rather than always the actual data.

Table 6-1. Staged ventilation rates for 4, 5 and 6 ventilation stages in animal barns*

Number of Stages:	4	5	6
	10	10	10
	15	14	13
	28	19	17
	100	35	25
		100	44
			100

* Expressed as percentages of the maximum ventilation rate.

As an example, suppose a five stage system is desired and calculations have shown the minimum ventilation rate is 15% of the maximum rate. Data for the five stage system show stage 2 is 40% greater than stage 1, and stage 3 is approximately 40% more than stage 2. Stage 4 is nearly twice stage 3. With these as guides, a staging schedule of: 15%, 21%, 29%, 50%, and 100% could be used and fans chosen to provide approximately these stages.

The procedure to choose fans to provide proper staging will be left as a topic for a later chapter. The focus here is on determining required ventilation rates.

Example 6-6

Problem: Reconsider the dairy barn of Example 6-1, et seq. The minimum ventilation rate has been determined to be 3.3 m^3/s and the maximum rate 18.3 m^3/s (no evaporative cooling is to be used). Determine the ventilation rates for a 5 stage ventilation system and specify thermostat setpoints.

Solution: The minimum ventilation rate is 18% of the maximum rate thus the recommended schedule in Table 6-1 must be modified. A candidate is

stage 1: 18% = 3.3 m^3/s
stage 2: 25% = 4.6 m^3/s (stage 1 + approximately 40%)
stage 3: 35% = 6.4 m^3/s (stage 2 + approximately 40%)
stage 4: 60% = 11.0 m^3/s
stage 5: 100% = 18.3 m^3/s

There are many candidate ventilation schedules. The one listed may not be necessarily the optimum, but it reflects the concept contained in Table 6-1. If anything, stages 2, 3, and 4 might be slightly smaller.

There are 5 ventilation stages, there must be 4 thermostat setpoints plus a freeze protection setpoint. A candidate schedule of setpoints is

freeze protection setpoint:	4 C
stages 1 to 2 setpoint:	8 C
stages 2 to 3 setpoint:	12 C
stages 3 to 4 setpoint:	17 C
stages 4 to 5 setpoint:	21 C

The stage 1 to 2 setpoint was chosen to correspond to the previous assumption that the minimum desirable barn temperature will be 8 C. The highest setpoint was chosen to correspond to the temperature at which the maximum ventilation rate was calculated. Intermediate setpoints were rather arbitrarily chosen to divide the range into equal increments.

6-6. Supplemental Heat

Ventilation graphs provide a tool to determine minimum ventilation rates for animal housing. In cold climates, a significant number of hours will find outdoor air temperature below the temperature where moisture and temperature criteria (or carbon dioxide and temperature criteria) curves intersect to determine the minimum ventilation rate. It is possible that freezing temperatures may occur inside the barn under conditions of extreme cold. Freeze protection should be part of the control sequence, but it is useful to determine how frequently freezing conditions could be encountered.

The action of freeze protection is to turn all fans off. In any typical housing for mature animals at normal stocking densities, when there is no ventilation there is more than enough animal heat to maintain the barn at a reasonably high temperature. When fans are off, the barn temperature will rise past the freeze protection point and the fans will turn back on. Operation will oscillate between the lowest stage and off until the extreme cold weather has passed. The effect in the barn will be humidity higher than the design value.

If this occurs for only a few hours during a year the effects are negligible, but if this occurs for many hours, a redesign using more insulation in the structure should be considered, or the barn redesigned to increase stocking density, or both.

Sensible energy balances permit a designer to determine the outdoor air temperature at which the indoor air temperature is at the freeze protection point.

Weather bin data can be used to estimate the number of hours per year that operation might be in this extreme state. If periods with no ventilation are to be avoided completely, weather data and a sensible energy balance can be used to calculate how much supplemental heat will be needed to prevent freezing.

Example 6-7

Problem: Continue to design the dairy barn described in Example 6-1, et seq. The location is near Denver, Colorado. Determine:

(a) how many hours in a typical year the barn will be at the freeze protection temperature (4 C), and
(b) whether, and, if so, how much supplemental heat capacity would be required to prevent the barn from ever being below 10 C, and alternately, ever below the freeze protection temperature if it is desired that the fans never be off.

Assume the minimum ventilation rate is to be constant at 3.3 m^3/s.

Solution, part a: At the freeze protection point, indoor air temperature will be 4 C, the ventilation rate will be 3.3 m^3/s, and animal sensible heat production will be

$$\begin{aligned} q_{sensible} &= (100)(570/500)^{0.734}(500)(1.8603 - 0.03074(4) - 0.0005268(4)^2), \\ &= 95{,}200 \text{ W}, \end{aligned}$$

based on the polynomial for sensible heat production obtained in Example 6-4. The polynomial to express latent heat production is

$$(q_{latent})/kg = 0.521481 + 0.011559t + 0.000575092t^2,$$

which is the default data in VENTGRPH. Note that although in Example 6-4 the linear equation for sensible heat production was best, the second order polynomial is used as it is in VENTGRPH. From the herd,

$$\begin{aligned} q_{latent} &= (100)(570 / 500)^{0.734} (500) (0.521481 + 0.011559 (4) \\ &\quad + 0.000575092(4)^2) \\ &= 31{,}800 \text{ W}. \end{aligned}$$

The heat of vaporization of water is 2460 kJ/kg at body temperature, thus, moisture production is

$$m_p = 31{,}800/2.46E + 6 = 0.01293 \text{ kg/s}.$$

Compare this to the ventilation graph for 10 C in Figure 6-5. Sensible heat production increased from 83 to 95.2 kW, and moisture production decreased from 0.01554 to 0.01293 kg/s.

Air density at 10 C and 70% relative humidity was 1.10 kg/m³. Density will change at the freeze protection temperature, but for now assume it remains at 1.10 kg/m³. At this point indoor relative humidity is not known so the new density cannot be determined.

The sensible energy balance is a tool to determine outdoor air temperature when the barn is at 4 C and ventilation is 3.3 m³/s,

ref eq. 5-12 p.159

$$t_o = \overset{t_i}{4\,C} - \frac{\overset{q_s}{95{,}200\ W}}{(1006\ J/kgK)(\underset{\rho}{1.10\ kg/m^3})(\underset{V}{3.3\ m^3/s}) + \underset{\Sigma UA\ at\ FP}{1063.8\ W/K}} \quad p.178$$

$$= -16.2\ C.$$

At -16 C and 80% relative humidity, the humidity ratio is 0.000843 kg/kg. If the ventilation rate is 3.3 m³/s at an air density of 1.10 kg/m³, the mass flow rate is 3.63 kg/s and the humidity ratio change is (0.01293 kg/s)/(3.63 kg/s) = 0.003562 kg/kg. The indoor humidity ratio will, thus, be 0.000577 kg/kg + 0.003562 kg/kg = 0.004139 kg/kg.

At 4 C and a humidity ratio of 0.004139 kg/kg, the relative humidity is 72% and air density is 1.13 kg/m. The initial assumption of air density is in error by a few percent. To be accurate, results to this point will be recalculated.

$$t_o = 4\,C - \frac{95{,}200\ W}{(1006\ J/kgK)(1.13\ kg/m^3)(3.3\ m^3/s) + 1063.8\ W/K}$$

$$= -15.8\ C.$$

The change is so slight it is safe to assume another iteration is not needed; indoor air density will be 1.13 kg/m³, with a relative humidity of approximately 72%. You might check this assumption.

Weather bin data for Denver (Appendix 6-2) show outdoor air temperature can be expected to be below - 16 C for p409

$$1\ hr + 8\ hrs + 35\ hrs + (137\ hrs)((-16\ C - (-17.8\ C)) / (-12.2\ C - (-17.8\ C)))$$

$$= 88\ hrs/yr.$$

Coincidentally, this number of hours per year is at approximately 4% of a winter, and -16 C is only slightly warmer than the 97.5% winter design temperature for Denver (which is -17 C). Differences among sets of weather data will arise frequently.

The 88 hours per year will accumulate during several short segments of time, probably only for several hours at a time during the coldest nights. For most dairy barns, this should create no problem. The fans would cycle on and off and the relative humidity would reach near 100%, but for only a short time.

Solution, part b: When the barn is held to be no colder than 10 C, supplemental heat will be required any time outdoor air temperature is below - 8 C (the intersection of the moisture and temperature criteria lines on Figure 6-5). However, the heating system must be sized to meet extreme heating needs when weather is at its coldest. From data in Appendix 6-2 for Denver, one hour per year is expected to fall within the range from - 34.4 to - 28.9 C.

The 99% winter design temperature for Denver in Appendix 6-1 is - 21 C but the 99% data can be exceeded for 1% of a winter, which is 21.6 hrs. (Note: the two sets of weather data do not agree exactly – they are from different sources and were obtained from weather data obtained for different periods of time. Such small differences are typical of variations among sets of weather data.)

Conservative design practice would lead to choosing a typical minimum outdoor air temperature for Denver to be - 34.4 C – the low end of the range which occurs 1 hr/yr. A sensible energy balance can be used to calculate supplemental heat required to maintain the barn at 10 C when it is this cold and the ventilation rate is 3.3 m³/s. from Eq 5-10, P. 159 $q_h = (\Sigma UA + FP + \rho \dot{V} c_p)\cdot\Delta t - q_s$

$$
\begin{aligned}
q_{suppl.} &= (1006\ \text{J/kg K})(1.10\ \text{kg/m}^3)(3.3\ \text{m}^3/\text{s})(10\ \text{C} + 34.4\ \text{C}) \\
&\quad + (1063.8\ \text{W/K})(10\ \text{C} + 34.4\ \text{C}) - 83{,}000\ \text{W} \\
&= 162{,}100\ \text{W} + 47{,}200\ \text{W} - 83{,}000\ \text{W} \equiv 126{,}000\ \text{W}.
\end{aligned}
$$

q_s at 10°C = 82583 W

The installed heating capacity must be 126 kW if the barn is never to go below 10 C, the minimum ventilation rate is to be continuous, and the lowest outdoor air temperature is - 34.4 C.

Note: Even at - 34.4 C the structural cover heat loss is only slightly more than half the sensible animal heat production. This would still permit fans to cycle to produce an average ventilation rate slightly less than a quarter of 3.3 m³/s. Supplemental heat would not absolutely be required, but if it were not used, relative humidity would be above the design value for many hours per year (the hours outdoor air temperature is between - 34.4 C and - 8 C).

If restrictions were relaxed to permit barn temperature to fall to 4 C, sensible animal heat production would rise to 95.2 kW and supplemental heat needs would be lowered to

$$
\begin{aligned}
q_{suppl.} &= (1006\ \text{J/kg K})\,(1.13\ \text{kg/m}^3)\,(3.3\ \text{m}^3/\text{s})\,(4\ \text{C} + 34.4\ \text{C}) \\
&\quad + (1063.8\ \text{W/K})(4\ \text{C} + 34.4\ \text{C}) - 95{,}200\ \text{W} \\
&= 144{,}500\ \text{W} + 40{,}800\ \text{W} - 95{,}200\ \text{W} \equiv 90{,}000\ \text{W}.
\end{aligned}
$$

Permitting barn temperature to be as low as 4 C has reduced the required heating capacity from 126 kW to 90 kW. But, of course, common practice would permit fans to cycle and humidity to rise during those hours of the winter when outdoor air temperature would be below - 16 C.

6-7. Greenhouse Ventilation

Greenhouse ventilation systems are sized for temperature control, when the major source of heat is the sun. General practice is to provide sufficient ventilation for 3/4 to 1 air change per minute as the maximum ventilation rate. This rate is sufficient to keep the temperature rise of ventilation air to approximately 5 K. The ventilation rate is sometimes increased for elevations over 600 m and regions with clear skies and high solar insolation. The increase is a field practice and is not based on calculations.

Fans usually are placed on sidewalls, and inlets on sidewalls opposite the fans. Thermal stratification causes the cool, fresh air to travel across the greenhouse and mix very little with hot air near the peak of the greenhouse. The result is an effective maximum ventilation rate larger than 3/4 to 1 air change per minute within the plant growing zone.

Sensible energy balances can be used to calculate the maximum required ventilation rate. As a rough approximation, heat loss by ventilation can be equated with solar gain. A more complete energy balance includes structural heat loss.

Example 6-8

Problem: A gutter-connected, ridge and furrow greenhouse is designed to be 200 m wide and 250 m long. The elevation where it is to be constructed is 500 m. The total UA value has been calculated to be 220,000 W/K.

Weather records show peak solar insolation to be approximately 700 W/m^2 (on a horizontal surface) and peak air temperatures to be 35 C. Calculate the ventilation required to keep the indoor air 5 K above outdoor air temperature.

Solution: Several assumptions are needed to begin this calculation. First, perimeter heat loss will be neglected compared to loss through the structural cover and ventilation. Second, the fraction of solar heat converted to sensible heat inside the greenhouse must be estimated. A rough rule is: a third of ambient solar insolation is reflected back to the outside or converted through photosynthesis, a third contributes through evapotranspiration to latent heat, and a third is sensible heat added to the indoor air. This rule will be used for this problem.

The total sensible heat contribution is estimated to be

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$$q_{sensible} = (1/3)\,(700\ \text{W/m}^2)\,(50{,}000\ \text{m}^2) = 11{,}700{,}000\ \text{W}$$

Ventilation is expressed on a volumetric basis, and air density is not known. Outdoor air temperatures of 35 C are expected, and the indoor temperature will be 5 K more, or 40 C. The relative humidity is not known, but will be assumed to be 50%. Program PLUS can be used to determine air density, 1.05 kg/m^3 at indoor conditions.

The required maximum ventilation rate can be calculated as

$$\dot{V} = \frac{q_{sensible} - (\Sigma UA + FP)(t_i - t_o)}{\rho_{air} c_p (t_i - t_o)}$$

The perimeter heat loss is assumed negligible compared with the structural cover heat loss, thus

$$\dot{V} = \frac{11{,}700{,}000\ \text{W} - (220{,}000\ \text{W/K})(5\text{K})}{(1.05\ \text{kg}/\text{m}^3)(1006\ \text{J/kg K})(5\text{K})}$$

$$= 2{,}000\ \text{m}^3/\text{s}.$$

$12000\ \text{m}^3/\text{h} \approx 24\ \text{m} \times 500\ \text{m}^2/\text{h}$

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SYMBOLS

for worse case design

$\Sigma UA + FP = 0$

$\dot{V} = \dfrac{q_{sen}}{\rho c_p \Delta t}$

Assuming $\Delta t = 5°$

$\dfrac{q_{sen}}{A} = \frac{1}{2} \cdot 700\ \text{W/m}^2$

$\dfrac{A.C}{\text{min}} = \dfrac{350}{1056 \cdot \Delta t} \cdot \dfrac{60}{H} = \dfrac{350}{88H}$

H | A.C
2.65m | 1.5
4m | 1

A	area, m2
CO_2	carbon dioxide volumetric concentration, ppm
CO_2	carbon dioxide production rate, m^3/s
F	perimeter heat loss factor, W/m K
m	mass flow rate, kg/s
m	mass production rate, kg/s
P	perimeter, m
q	heat flow, W
t	temperature, C
U	unit area thermal conductance, W/m^2 K
V	volume, m
$\dot{V}$	volumetric flow rate, m^3/s
W	humidity ratio, kg/kg
ρ	density, kg/m^3

EXERCISES

1. Continue Example 6-3 by determining the barn temperature when outdoor air temperature is - 30 C. If the barn temperature will fall below freezing, calculate the supplemental heat capacity (kW) to be installed to prevent freezing.

2. Repeat the design in Example 6-3, assuming the design criterion for relative humidity inside the barn is 80%.

3. Use VENTGRPH to obtain data for the minimum ventilation rate for the barn in Example 6-3 as a function of design indoor relative humidity. Explore the relative humidity range from 60% to 100%.

4. Determine the minimum required ventilation rate for a poultry laying house with 30,000 white leghorn laying hens averaging 1.8 kg when ΣUA for the building is 650 W/K, the perimeter is 150 m, and the perimeter heat loss factor is 1.2 W/m K. Assume the maximum indoor relative humidity is 80% and carbon dioxide cannot go above 4000 ppm. Identify any assumptions you make in your analysis.

5. Determine the maximum ventilation rate needed for the poultry house described in problem 4.

6. Develop a fan staging schedule for the poultry house described in problems 4 and 5, and the minimum and maximum ventilation rates calculated in the problems.

7. Use VENTGRPH to obtain data for the minimum ventilation rate for the barn in Example 6-3 as a function of design indoor air temperature. Explore the air temperature range from 5 to 25 C. Provide a physical explanation for the trend in your results.

8. Use VENTGRPH to design the ventilation system for the barn described in Example 6-1 for a range of elevations from sea level up to 2000 m. Graph the minimum ventilation rate (m^3/s and kg/s) as a function of elevation and interpret the trends of the two curves.

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