CHAPTER 10
VENTILATION CONTROL AND QUANTIFICATION OF PERFORMANCE

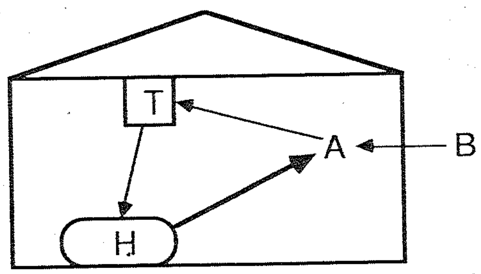
10-1. Introduction

Negative pressure ventilation systems with continuous slotted inlets have been a focus of this text to this point. This chapter will also emphasize slotted inlet ventilation, although the concepts of control and fan system duty factors apply regardless of the type of ventilation.

In feedback control, a signal (i.e., a temperature measurement) is obtained which is information for the control system to use to determine what control action should be taken. The information is compared to a predetermined reference input (i.e., the thermostat setpoint) and if the difference between the two is sufficient to cause some action, a control element (the heater or fans) is activated in a manner to reduce the difference between the reference input (setpoint) and the controlled variable (air temperature) .

A simple feedback control system is that which activates a heater or fan to control air temperature in a building.

A building has an air temperature, A, in response to outside air temperature, B, and heat sources within the building, including the heater, H. A thermostat, T, measures A and controls the operation of H. If A is above the thermostat setpoint, nothing happens. If A is below the thermostat setpoint, H is activated until A is brought back to the desired temperature.



In Figure 10-1, a generalized strategy to control air temperature in a building, either by venting when needed or heating, is sketched. Outside air temperature and other heating loads influence inside air temperature. Inside air temperature, in conjunction with thermostat setpoints, controls vents, fans or heaters. The influence of vents, fans, or heaters on inside air temperature is the negative feedback mechanism. The feedback is negative because the greater the action of the control element, the less need for more action. (For example, the more heat that was delivered during the recent past, the less will be needed during the immediate future.)

Control engineers have developed a classification system for different types of control. Common types are: on/off control, proportional control, integral control, derivative control, and combinations of one or more (for example, proportional plus integral control). These are all types of feedback control.

On/Off control. The first type of control, on/off, is the simplest and can be activated either manually (e.g., a light switch) or with an automatic system (e.g., a thermostat). On/off control is typically applied to fans in agricultural buildings. Single speed fans are used, and they are either on or off. On/off control is also frequently used for heating systems in both animal housing and greenhouses. The advantage is simplicity of control; the disadvantage is less accurate control because of an inability to modulate the response according to intensities of other conditions.

One way in which on/off control is less accurate is the manner in which it can lead to overshooting the controlled variable. For example, consider a steam heating system for a greenhouse during very cold weather. After the system has operated for some time, air temperature reaches the temperature setpoint and the call for heating ceases. The heating pipes still contain both leftover steam and a significant amount of heat within the thermal mass of the pipes. However, the rate of heat loss from the greenhouse during cold weather is great, so the extra heat does not create a significant degree of overshoot of greenhouse temperature. The same heating system, however, is also activated during mild weather, and in that situation, extra heat within the heating pipes can cause air temperature within the greenhouse to overshoot beyond the setpoint by several degrees. Fortunately, most biological systems are forgiving of this type of inadequate temperature control, although energy is wasted and control is not optimum.

Proportional control. In proportional control, the rate at which the controlled parameter (e.g., heat, ventilation) is delivered is proportional to the difference between the reference input and the controlled output (e.g., air temperature).

Proportional control is commonly used with variable speed fans and to control heating systems. In ventilation systems, fan speed and, thus, airflow rate are

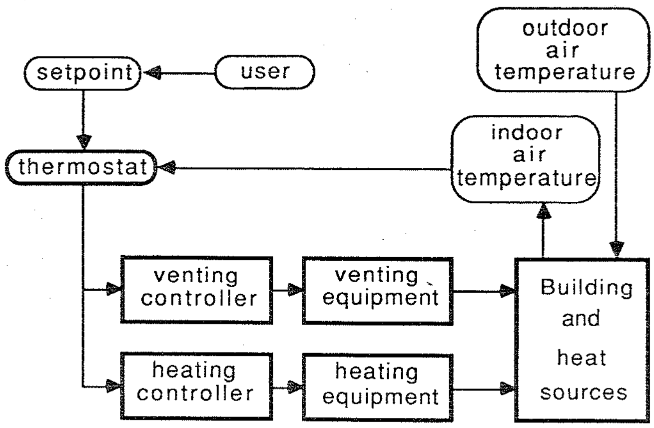


Figure 10-1. Generalized scheme for controlling building air temperature using venting and heating.

functions of the amount by which airspace temperature deviates from the desired temperature. As the temperature error increases (air temperature falls farther below the setpoint), fan speed of variable speed fans decreases in an attempt to limit heat losses and maintain the desired temperature.

Proportional control may be applied to heating systems. If it is used, for example, to control greenhouse heating in the system described above, control is more stable and results in less overshooting of temperature setpoints. As indoor air temperature approaches the setpoint, the rate at which steam is delivered slows and less heat is left within the heating pipes at the end of the cycle. Thus, there is less overshoot during mild weather.

However, proportional control requires a more sophisticated control system. Whenever possible for environmental control in agricultural buildings, on/off control has traditionally been used because of its simplicity. Proportional control has found more use in environmental control of commercial and industrial buildings. However, as computerized control systems are developed, sophisticated control strategies can be implemented through software development. One development can be applied in many buildings, significantly lowering the cost of application.

Integral control. Integral control limits small long-term offsets from setpoints and is not frequently used for environmental control in agriculture. Integral control acts to integrate error over time and bring the average error to zero. Its response is slow, but its long-term accuracy in obtaining an average is good.

One possible application is for temperature control. If the air temperature remains slightly below the thermostat setpoint for an extended period of time, integral control acts to integrate the error (difference between actual and desired temperatures) and eventually forces a correction.

Derivative control. Derivative control is used when a rapid response is desired. The time derivative of error determines the magnitude of the response.

Derivative control is useful to prevent sudden large departures of the controlled variable from the setpoint, but also has inherent instability and can cause responses which overshoot the desired setpoint. Derivative control is not frequently used for environmental control (although it can be part of a larger control strategy, especially as computerized control systems become more common).

Other control methods such as pseudo-derivative-feedback control (Phelan, 1977) and adaptive control are currently being explored by researchers for applications to environmental control in agriculture.

As long as items of environmental control equipment (fans, heaters, etc.) are in good repair, their capacity to deliver the desired flux of fresh air, heat, etc., can be considered constant over the long term. Although the static pressure differences from inside a building to outside influences the airflow rate through a fan, operation is usually within a sufficiently narrow range that fan capacity can also be considered constant in the short term.

The initial focus of this chapter will be on aspects of ventilation control and how control influences the duty factor of a ventilating system and thereby the cost of ventilation.

10-2. Duty Factors, Staged Fan Systems Using Single Speed Fans

A fan system duty factor is defined as the average ventilation rate for a year (m^3/s) divided by the maximum ventilating capacity of the fan system. The average ventilating rate is composed of operation during all times of the year – times when operation is at the lowest stage, the highest stage, and all stages between. If the duty factor and efficiencies of the fans are known, the yearly cost of ventilation can be calculated. The yearly cost is sensitive to ventilation stages and thermostat setpoints, as well as the inherent efficiencies of fans, and thermal parameters of the building being ventilated.

Ventilation staging was discussed in Chapter 6. Methods to choose temperature setpoints were described and a procedure was outlined to choose ventilation stages between the maximum and minimum ventilation rates.

Staged control can be viewed in several ways. Consider the hypothetical situation of sensible heat production within a barn which remains constant while the outdoor air temperature varies. The ventilation rate as a function of indoor air temperature would be as shown in Figure 10-2. The sketch demonstrates a system with four fan stages, $\dot{V}_1$, $\dot{V}_2$, $\dot{V}_3$, and $\dot{V}_4$. Three thermostat setpoints are required for control, one setpoint between each adjoining pair of stages. A fourth setpoint, for freeze prevention, might be recommended but would not be a factor in determining the fan system duty factor unless environmental control was poorly designed and ventilation was deactivated to prevent freezing for extended periods of a typical year.

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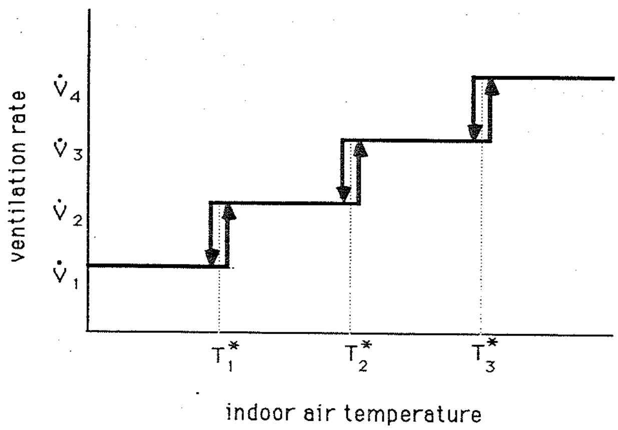

Figure 10-2. An example of ventilation control using single-speed, staged fans, as a function of indoor air temperature. Starred values are thermostat setpoints.

Suppose an animal housing facility is being ventilated but outdoor air temperature is very low. Ventilation will be at stage 1, the lowest stage, with indoor air temperature below the lowest thermostat setpoint. Then the outdoor air temperature rises. The first effect is to permit indoor air temperature to rise to the lowest thermostat setpoint. Thermostats have hysteresis, which is a lag in response caused by, among other factors, thermal mass. Indoor air temperature rises slightly above the first thermostat setpoint before the second stage of ventilation is activated.

After the second stage of ventilation is activated, the increased ventilation rate causes barn temperature to begin to fall – outdoor air temperature is not sufficiently high to permit the barn to maintain air temperature at the first thermostat setpoint when the ventilation rate is as high as stage 2. After passing the hysteresis region, the thermostat deactivates stage 2 ventilation. However, conditions which caused activation of stage 2 still exist and the barn temperature rises again through the hysteresis region.

As long as outside air temperature remains within some narrow range, fan control will alternate between stages 1 and 2 and indoor air temperature will remain within the narrow region defined by the hysteresis of the thermostat. When outdoor air temperature is in the lower parts of the narrow (and still undefined) temperature range, ventilation is primarily in stage 1, with limited excursions to stage 2. As outdoor air temperature rises through this narrow range, ventilation gradually changes from mostly stage 1 to mostly stage 2.

An outdoor air temperature is eventually reached where indoor air temperature can be sustained at or above the first thermostat setpoint with constant stage 2 ventilation. As outdoor air temperature continues to rise, stage 2 ventilation is maintained and indoor air temperature rises toward the second thermostat setpoint. When the second setpoint is reached, the same oscillation between stages 2 and 3 occurs as has been described between stages 1 and 2. If the outdoor air temperature rise continues, the same scenario holds to eventual continuous operation at stage 4.

2 A different way to describe ventilation staging is in Figure 10-3. The average ventilation rate is a smooth function of outdoor air temperature even though regions of oscillation between stages exist. The average ventilation rate can be calculated as a function of outdoor and indoor temperatures and thermal parameters of the building.

3 A third means to view ventilation staging is shown in Table 10-1. The table contains values of outdoor air temperature which correspond to steady-state ventilation at combinations of ventilation stages and thermostat setpoints. For example, at an outdoor air temperature of $(t_o)_{1,2}$, and a ventilation rate of $\dot{V}_2$, indoor air temperature will remain steady and equal to the first setpoint, SP_1. The indoor air temperature will also be steady and equal to SP_1 when the ventilation rate is $\dot{V}_1$ but the outdoor air temperature is $(t_o)_{1,1}$. The ventilation

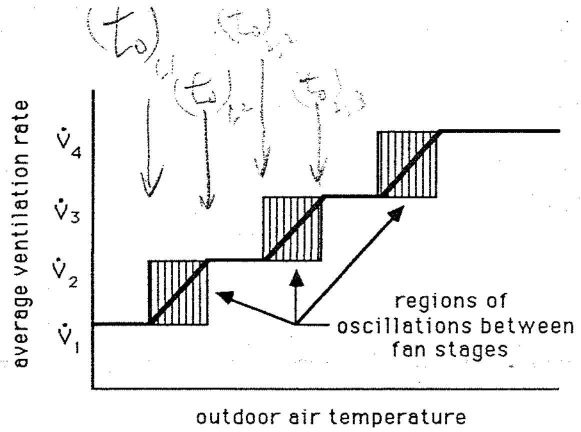

Figure 10-3. An example of ventilation control using single-speed, staged fans, as a function of outdoor air temperature.

rate $\dot{V}_1$ is less than $\dot{V}_2$, thus, $(t_o)_{1,1}$ must be less than $(t_o)_{1,2}$. Similarly, SP_2 is higher than SP_1, thus, $(t_o)_{2,2}$ is higher than $(t_o)_{1,2}$.

Values of outdoor air temperature in Table 10-1 are calculable using a steady-state thermal energy balance,

see p182 (6-3)

$$t_o = [\,-q_{prod} + (\Sigma UA + FP + mc_p)t_i] \,/\, (\Sigma UA + FP + mc_p), \quad (10\text{-}1)$$

where q_{prod} is the sensible heat production within the airspace, $\Sigma UA + FP$ is the structural heat loss factor(HLF), and m and c_p are the mass flow rate of ventilation air and specific heat of air, respectively.

Equation 10-1 is more convenient to use when the sensible heat production of animals is treated as a function of indoor air temperature (expressed for the entire herd or barn),

$$q_{prod} = a + bt_i + ct_i^2. \quad (10\text{-}2)$$

Sensible heat production can be expressed as a second order polynomial using program POLYNOM, for example. When sensible heat production is expressed in this form, indoor air temperature may be calculated by

(10-4)

$$t_i = \frac{-(b - HLF - mc_p) - \sqrt{(b - HLF - mc_p)^2 - 4c(a + HLFt_o + mc_pt_o)}}{2c} \quad (10\text{-}3)$$

Table 10-1. Outdoor air temperatures corresponding to n thermostat setpoints and n + 1 ventilation stages

Thermostat Setpoint	Ventilation Rate $\dot{V}_1$	$\dot{V}_2$	$\dot{V}_3$	...	$\dot{V}_n$	$\dot{V}_{n+1}$
SP_1	$(t_o)_{1,1}$	$(t_o)_{1,2}$				
SP_2		$(t_o)_{2,2}$	$(t_o)_{2,3}$			
SP_3			$(t_o)_{3,3}$			
⋮						
SP_n					$(t_o)_{n,n}$	$(t_o)_{n,n+1}$

where HLF is the building's Heat Loss Factor ($\Sigma UA + FP$) and outdoor air temperature may be calculated using

$$t_o = \frac{-ct_i^2 - (b - HLF - mc_p)t_i - a}{HLF + mc_p}. \qquad (10\text{-}4)$$

Note: In Equations 10-3 and 10-4, the constant term "a" may include sensible heat produced by sources such as lights and motors. Heat from such sources is not produced as a function of indoor air temperature.

The outdoor air temperature ranges in Table 10-1 have known probabilities of occurrence for any location where weather records are available. For example, weather bin data such as in Appendix 6-2 can be used to estimate the number of hours during a year when outdoor air temperature is within a certain range, and the probability is that number of hours divided by the total number of hours in a year (8760).

Each temperature range also has a known average ventilation rate associated with it. For example, the range $(t_o)_{1,2}$ to $(t_o)_{2,2}$ defines when the average ventilation rate is $\dot{V}_2$. Temperature ranges when ventilation oscillates between stages require an assumption. It will be assumed when outdoor air temperature is between $(t_o)_{1,1}$ and $(t_o)_{1,2}$, for example, the average ventilation rate is $(\dot{V}_1+\dot{V}_2) / 2$. That is, the probability of t_o within the range $(t_o)_{1,1}$ to $(t_o)_{1,2}$ is uniformly spread over the temperature range and the range is small enough that sensible heat production is effectively constant.

The probability of a temperature range is denoted by P. For example, $P[(t_o)_{1,1} < t_o < (t_o)_{1,2}]$ is the probability outdoor air temperature will be within the indicated temperature range during a year. It equals the number of hours during a year the outdoor air temperature is within the range $(t_o)_{1,1}$ to $(t_o)_{1,2}$ divided by the number of hours in a year. The sum, $\Sigma P[\text{all ranges}]$, equals unity.

The average ventilation rate for a year is the weighted average of ventilation rates within each temperature range possible for the year. The average is weighted by the probabilities of occurrence of the ranges,

$$\dot{V}_{ave} = \Sigma(\dot{V}_{range} P[\text{range}]) \qquad (10\text{-}5)$$

For data in Table 10-1,

$$\begin{aligned}
\dot{V}_{ave} = {} & \dot{V}_1 P[-\infty < t_o < (t_o)_{1,1}] \\
& + 1/2(\dot{V}_1 + \dot{V}_2) P[(t_o)_{1,1} < t_o < (t_o)_{1,2}] \\
& + \dot{V}_2 P[(t_o)_{1,2} < t_o < (t_o)_{2,2}] \\
& + 1/2(\dot{V}_2 + \dot{V}_3) P[(t_o)_{2,2} < t_o < (t_o)_{2,3}] \qquad (10\text{-}6) \\
& + \dot{V}_3 P[(t_o)_{2,3} < t_o < (t_o)_{3,3}] +
\end{aligned}$$

The duty factor is,

$$\text{duty factor} = \dot{V}_{ave} / \dot{V}_{n+1}. \tag{10-7}$$

10-3. Ventilating Efficiency Ratios and the Cost of Ventilation

Fan efficiency can be expressed in several ways, but the way most informative to agricultural producers relates the useful output (m^3/s of air delivery) to input which must be paid for (kWh of electricity). The ratio of these two parameters will be termed the Ventilating Efficiency Ratio (VER),

$$\text{VER} = (\text{m}^3/\text{s of air delivery}) / (\text{kW of electricity}) \tag{10-8}$$

A recent trend has been to rate agricultural ventilation fans in terms of their VER (termed the cfm/watt ratio in IP units). Data, when available, usually is presented for several static pressure differences. Data for the difference which represents expected operating conditions should be chosen.

A further caution is that only data which represents operation of the fans as installed should be used. That is, safety guards, louvers, and other attachments should have been in place during tests. If not, significant differences from operation with all attachments can result.

Three factors are needed to calculate the cost to operate a ventilating system for a year: the duty factor (or average ventilating rate for the year), the cost of electricity, and the VER of the fan system. The cost is calculated using

$$\text{cost} = \frac{(8760 \text{ hrs / yr})(\text{max capacity})(\text{duty factor})(\text{unit elect. cost})}{\text{Ventilating Efficiency Ratio}} \tag{10-9}$$

Example 10-1

Problem: Determine the cost to operate a fan (5.5 m^3/s capacity) for a year if the VER is 8 m^3/s-kW, the cost of electricity is expected to be \$0.11/kWh, and the expected duty factor is 0.6.

Solution: Equation 10-9 provides the solution directly,

$$\text{cost} = \frac{(8760 \text{ hrs / yr})(5.5 \text{ m}^3 / \text{s})(0.6)(\$0.11 / \text{kWh})}{8 \text{ m}^3 / \text{s-kW}}$$
$$= \$397.49 / \text{yr}$$

Note: Although it is not frequently realized, the cost to operate a ventilating fan during its lifetime is usually far greater than its purchase cost. It is usually recommended to invest slightly more at the time of installation to purchase energy efficient fans as the VER of seemingly identical fans may differ by 50% or more.

10-4. The Cost of Ventilating Animal Housing, an Example

Procedures have been presented at various places in this text to:

(a) determine heat loss characteristics of a building,

(b) determine ventilation requirements and methods to stage ventilation between the maximum and minimum ventilation rates,

(c) determine appropriate thermostat setpoints,

(d) estimate ventilating system duty factors, and

(e) calculate how much ventilation systems cost to operate.

The following example is presented to demonstrate an integration of the procedures.

Example 10-2

Problem: A mechanically-ventilated, tie-stall dairy barn is being designed for construction near Denver, Colorado. The barn is to house 150 milking cows, averaging 550 kg body weight. A design has been chosen which results in a structural heat loss factor (walls, ceiling, perimeter, etc.) of 900 W/K. General lighting, designed at approximately 10 W/m^2, is expected to operate around the clock. The floor area of the barn will be 1500 m^2.

Minimum and maximum ventilation rates will be 2.9 and 21 m^3/s, respectively. Fans are being considered which have a VER of 9 m^3/s-kW and electricity is expected to cost $0.10/kWh.

Choose fan stages and thermostat setpoints and estimate the yearly cost to ventilate the barn.

Solution: First, fan stages and thermostat setpoints will be determined. A five-stage fan system is assumed for now, with thermostat setpoints at 9, 13, 17, and 21 C. (These are example setpoints and not proposed as optimum.)

The lowest fan stage is 2.9/21 or 13.8% of the highest. To follow the concept of fan staging presented in Table 6-1, the following stages are chosen:

Stage	Percentage of Maximum	m^3/s
1	13.8	2.9
2	19	4.0
3	25	5.3
4	45	9.5
5	100	21.0

A next step is to determine sensible heat production within the barn. Lighting will produce a constant $(10\ W/m^2)(1500\ m^2) = 15{,}000$ W. Appendix 5-1 contains data to estimate sensible heat production from animals. For 500 kg dairy cows,

Air Temp,C	Sensible Heat Prod., W/kg
-1	1.9
10	1.5
15	1.2
21	1.1
27	0.6

These data were examined in Example 6-4, and polynomials fitted to the sensible heat data using program POLYNOM. The second order polynomial yielded:

$$q'_{prod} = 1.8603 - 0.03074t - 5.268E-04t^2,$$

expressed in W/kg for a 500 kg dairy cow.

Transforming q''_{prod} to the entire herd,

$$\begin{aligned} q_{prod} &= (150 \text{ cows})\,(500 \text{ kg})\,(550/500)^{0.734}(q'_{prod}) \\ &= 149{,}600 - 2473t - 42.4t^2, \end{aligned}$$

+ 15000

where t is indoor air temperature, C.

Sensible heat production within the airspace includes animal heat and heat from lights. Heat from lights is not a function of indoor air temperature, thus total heat production within the barn is

a = 164600

b = -2473

$$q_{barn} = 164{,}600 - 2473\,t - 42.4t^2.$$

c = -42.4

The elevation of the barn is not specified in the example, but being near Denver it is likely to be approximately 1500 m. Atmospheric pressure is estimated to approximate 85 kPa. The density of air is required to determine outdoor air temperatures as in Table 10-1. A changing air temperature but constant relative humidity within the barn of 60% will be assumed. Air densities are:

Indoor Air Temperature, C	Air Density, kg/m^3
9	1.05
13	1.03
17	1.02
21	1.00

The change of air density over the range of thermostat setpoints is small, an average value of 1.02 kg/m^3 is sufficiently accurate to be used as a constant, but the data were checked to be sure.

The point has been reached where the table of outdoor air temperatures as a function of thermostat setpoints and ventilation rates can be calculated. Equation 10-4 applies.

As an example calculation, consider 13 C setpoint and the third stage of ventilation, 5.3 m^3/s. The mass flow rate is

$$m = (5.3\ \text{m}^3/\text{s})(1.02\ \text{kg/m}^3) = 5.41\ \text{kg/s}.$$

The outside air temperature corresponding to this combination of indoor air temperature and ventilation rate is

$$t_o = \frac{-(-42.4)(13)^2 - [-2473 - 900 - (5.41)(1006)](13) - 164{,}600}{900 + (5.41)(1006)} = -6.75$$

Similar calculations yield the following data for outdoor air temperature, C:

Thermostat Setpoint, C	Ventilation Rate, kg/s 2.90	4.08	5.41	9.69	21.42
9	- 27.4	- 18.8 C			
13		- 12.0	- 6.75 C		
17			- 0.39	6.63 C	
21				12.2	16.8 C

There are now nine temperature ranges for which the probability of occurrence during a typical year must be determined. Using the data for Denver, the probability of the first temperature range is

$$\begin{aligned} P[-\infty < t_o < -27.4\ \text{C}] &= P[-\infty < t_o < -34.4] + P[-34.4 < t_o < -28.9] \\ &\quad + P[-28.9 < t_o < -27.4] \\ &= [0\ \text{hrs} + 1\ \text{hr} + (8\ \text{hrs})(-28.9 + 27.4) \\ &\quad / (28.9 + 23.3)] / 8760 \\ &= [0\ \text{hrs} + 1\ \text{hr} + 2\ \text{hrs}] / 8760\ \text{hrs} \\ &= 0.0003 \end{aligned}$$

Probabilities of the other temperature ranges are obtained in the same manner, leading to the following data:

Temperature Range	Hours Per Year in the Range (a)	Probability of the Range
to -27.4	3	0.0003
-27.4 to -18.8	35	0.0040
-18.8 to -12.0	157	0.0179
-12.0 to -6.75	366	0.0418
-6.75 to -0.39	1132	0.1292
-0.39 to 6.63	1833	0.2092
6.63 to 12.2	1485	0.1696
12.2 to 16.8	1227	0.1401
16.8 up	2522	0.2879

(a) rounded to the nearest hour

The average ventilation rate for the year can now be calculated using Equation 10-6. Note that volumetric airflow rates can be used again, mass flow rates were necessary only for the sensible energy balance.

$$\begin{aligned}\dot{V}_{ave} &= (2.9\ m^3/s)(0.0003) + 1/2(2.9\ m^3/s + 4.0 m^3/s)(0.0040)\\ &\quad + (4.0\ m^3/s)(0.0179) + 1/2(4.0\ m^3/s + 5.3\ m^3/s)(0.0418)\\ &\quad + (5.3\ m^3/s)(0.1292) + 1/2(5.3\ m^3/s + 9.5\ m^3/s)(0.2092)\\ &\quad + (9.5\ m^3/s)(0.1696) + 1/2(9.5\ m^3/s + 21\ m^3/s)(0.1401)\\ &\quad + (21\ m^3/s)(0.2879),\\ &= 0.00087 + 0.01380 + 0.07160 + 0.19437 + 0.68476 + 1.54808\\ &\quad + 1.61120 + 2.13653 + 6.04590\\ &= 12.30711, \text{ or } 12.3\ m^3/s.\end{aligned}$$

The duty factor is thus $(12.3\ m^3/s)/(21\ m^3/s)$, or 0.59. Typical duty factors for fan systems in ventilated animal housing are in the range 0.5 to 0.7 in moderate climates.

The yearly cost of ventilation can now be determined, using Equation 10-9.

$$\begin{aligned}\text{cost} &= \frac{(8760\ \text{hrs / yr})(21\ m^3 / s)(0.59)(\$0.10)}{9 m^3 / \text{s-kW}}\\ &= \$1206, \text{ or approximately } \$8 \text{ per cow-yr.}\end{aligned}$$

It should be noted that an overall VER of 9 m^3/s-kW is relatively high if numerous small fans (e.g., less than one meter diameter) are used in the system. A lower VER value raises the cost of ventilation.

10-5. Program DUTYFACT

The work required to complete Example 10-2 is substantial, and the effort involved is a disincentive to repeat the calculations for different fan staging schedules, thermostat setpoints, and building thermal parameters. Also, Example 10-2 was greatly simplified from a computational standpoint by assuming the VER is uniform among all the fans. In fact, each fan model has its own VER, ranging from approximately 3 for small fans (and large fans which are not efficient) to 10 and larger for efficient, large diameter ventilating fans (e.g., 1 m diameter and larger).

A computer program which includes the more general case of differing VER values, and permits the user to vary parameters easily, would be a design tool useful for an engineer to demonstrate the effect of ventilation control on the yearly cost of ventilation. Program DUTYFACT is provided as an example of a program designed for such simulations. It is intended for demonstration purposes, not as an everyday design tool.

Several assumptions are included in DUTYFACT. A simple model of atmospheric pressure as a function of elevation is included to calculate air density,

$$\text{air pressure} = 101.325 \exp(-0.00011943\, z) - 6.799E-06z - 6.976E-08z^2, \qquad (10\text{-}10)$$

where z is elevation in m and air pressure is expressed in kPa.

In addition, because the actual operation static pressure difference is not known, the VER of each fan (a function of static pressure difference) is assumed equal to the VER at a pressure difference of 12.5 Pa. This is an intermediate value and represents a desirable pressure difference in a well constructed and managed barn. Note: DUTYFACT assumes once a fan or group of fans is activated, it remains active for all higher stages. Fans can be turned off or changed, however, at higher stages by specifying them by letter and setting their number to zero (or some other number if appropriate).

Example 10-3

Problem: Reconsider the barn described in Example 10-2. Use program DUTYFACT to calculate the cost of ventilating the barn when the fan staging schedule described in Example 10-2 is used, and recalculate the cost when the range of ventilation is segregated into equal increments. Use the default fan data in DUTYFACT to select fans for staging and maintain the setpoints used in Example 10-2.

Solution: Program DUTYFACT is used and data is entered as follows:

1. elevation above sea level = 1500 m,

2. ΣUA value = 900 W /K,

3. FP value = 0 W/K (UA and FP are not separated in the statement of the

problem. For convenience, the Heat Loss Factor is consolidated into UA. In the sensible energy balance, the two are added so the consolidation has no effect),

4. (a), (b), and (c) are as calculated: 164,600 W, - 2473 W/C, and 42.2 W/C^2 (Note: for the entire barn),

5. the cost of electricity is \$0.10 per kWh (somewhat higher than current agricultural rates, if available, but not unusually so), and

6. the default fan data are not changed.

A. The cost of ventilation with fan staging as done in Example 10-2.

Fan Stage	Ventilation Rate, m^3/s
1	2.9
2	4.0
3	5.3
4	9.5
5	21.0

An example fan staging schedule to approximate these ventilation rates is:

Fan Stage	Fans to Operate	m^3/s at 12.5 Pa
1	5 - model b	2.85
2	7 - model b	3.99
3	7 - model b	
	2 - model c	5.35
4	7 - model b	
	2 - model c	
	2 - model e	9.35
5	7 - model b	
	2 - model c	
	2 - model e	
	2 - model d	
	2 - model j	21.21

Note: This number of fan models would likely not be used in an actual design. A more compatible selection, with real data, would be used to approximate the proper staging.

The setpoints entered in DUTYFACT are 9, 13, 17, and 21 C. The city to be chosen is Denver and the weather bin data need not be altered. When all the data are entered, the results of the program are as follows.

	Ventilation Rate, m^3/s				
Thermostat Setpoint, C	2.85	3.99	5.35	9.35	21.21
9	- 27.46	- 18.93 C			
13		- 12.19	- 6.69 C		
17			- 0.34	6.44 C	
21				12.00	16.83 C

The control table of outdoor air temperatures differs little from the one in Example 10-2, and would not be expected to differ. There are only slight changes of the ventilation rates and other data are identical. The probabilities of outdoor temperature ranges are provided by the program and need not be listed here; they differ little from the data calculated in Example 10-2.

The average ventilation rate at 12.5 Pa is calculated by the program as 12.38 m^3/s, with a duty factor of 0.58, again not very different than before. However, the cost of ventilation is greatly different. The average VER is 4.83 and the yearly cost to ventilate is $2244.84. In Example 10-2, the VER was stated to be 9, and such a value might represent very efficient, large diameter fans. However, for much of the year smaller and less efficient fans operate, thus, the cost is not solely a function of the efficiency of the largest fans. Also, the default fan data do not list fans with VER values as high as 9, but the default data are more realistic.

The probabilities of the outdoor air temperature ranges should be examined with some care, for they provide insight into strategies to lower the cost of mechanical ventilation. In this example,

Temperature Range	Probability of the Range	Fan Operation
to - 27.46	0.000349	stage 1 only
- 27.46 to - 18.93	0.003854	stages 1 and 2
- 18.93 to - 12.19	0.016545	stage 2 only
- 12.19 to - 6.69	0.043454	stages 2 and 3
- 6.69 to - 0.34	0.130651	stage 3 only
- 0.34 to 6.44	0.201918	stages 3 and 4
6.44 to 12.00	0.169281	stage 4 only
12.00 to 16.83	0.147024	stages 4 and 5
16.83 up	0.286924	stage 5 only

The VER values for the temperature ranges are

Fan Stage	VER, m^3/s-kW
1	3.660
2	3.660
3	3.631
4	4.075
5	5.642

As the stages increase, so too does the VER because larger and more efficient fans are activated.

The probability data show a third of the year involves the outdoor air temperature range from - 6.69 C to 6.44 C. This range involves either stage 3 by itself or alterations between stages 3 and 4. Yet, the VER of stage 3 is relatively low. A strategy to lower the cost of ventilation would thus be to obtain more efficient fans for stage 3 as a first priority. This, of course, involves also selecting efficient fans for stages 1 and 2 because of the assumption the stages are additive.

B. The next step is to divide the fan staging into equal increments.

Fan Stage	Ventilation Rate, m^3/s
1	2.90
2	7.43
3	11.95
4	16.48
5	21.00

With the default fan data in DUTYFACT, a candidate fan schedule is:

Fan Stage	Fans to Operate	m^3/s at 12.5 Pa
1	5 - model b	2.85
2	5 - model b	
	4 - model d	7.57
3	5 - model b	
	8 - model d	12.29
4	5 - model b	
	8 - model d	
	2 - model e	16.29
5	5 - model b	
	8 - model d	
	2 - model e	
	2 - model f	21.25

The same type of data as described in A above is presented by DUTYFACT, but need not be listed here. The example asks for the cost of ventilation, which in this second fan staging approach, is $2892.51 per year. This is significantly greater than the cost in part A. However, the cost difference cannot be generalized, for it depends on the probability distribution of weather and the fans selected for stagings and their VER values. However, it is not unusual to find that staging the fans with small initial steps and large final steps results in a lower overall cost of ventilation than the cost for equal increments of stages. This is an additional advantage of avoiding large steps between the lowest fan stages.

Example 10-4

Problem: Continue Examples 10-2 and 10-3, but in this example use the fan staging of Example 10-2 and examine the effects of changing various setpoints.

Solution: Program DUTYFACT can be used. For a variety of setpoint schedules, the resulting costs of ventilation are:

Setpoints	Yearly Ventilation Cost (Electricity)
9, 13, 17, 21	$2244.84
9, 13, 19, 21	$2180.09
9, 15, 19, 21	$2169.21
9, 13, 21, 25	$1916.93
9, 15, 21, 25	$1905.41 ←
9, 15, 21, 22	$2052.22
5, 13, 17, 21	$2245.37
9, 15, 17, 21	$2234.09

The setpoint schedules do not vary greatly from the original because the primary purpose of the schedule is to promote cow comfort, not lower the cost of ventilation. However, some effects can be seen. The magnitudes of the effects depend on the fans chosen for the fan schedule and the expected weather bin data, and so should not be seen as general rules for choosing setpoints.

The effect of changing the lowest setpoint is not great in this example. Lowering just the lowest setpoint from 9 to 5 C raised the yearly cost of ventilation by only approximately $10. This is an outgrowth of the weather data. With Denver weather, there are few hours of extreme cold where only stage 1 ventilation would be needed when the first setpoint is 9 C. Lowering the setpoint has little effect, but might be of some benefit to the cows by avoiding a few hours with conditions of minimum ventilation and high relative humidity. This observation may not be true for a region with a colder climate.

The effects of the middle setpoints are greater. Weather data for Denver show a peak in the bin data for outside temperatures between 10.0 and 15.6 C. Raising the middle setpoints so the control table of outdoor air temperatures shows more temperatures in the upper part of this range will yield more significant electricity savings. For example, raising the setpoint schedule to 9, 13, 21, and 25 C is projected to save more than $300, but a further change to 9, 15, 21, and 25 C leads to only $10 additional savings.

It must be emphasized that electricity cost is not the only criterion for choosing setpoints; it is not even the major criterion. Animal comfort and productivity as a function of ambient temperature must be the primary concern. However, within the range of thermoneutrality, the effects of various setpoint schedules can be seen using the duty factor approach, and some savings achieved. In the future, work to optimize setpoint schedules could include animal productivity models, ventilation cost models, heating cost models, etc., to achieve an optimum program of environmental control.

→ check matlab version vent_cost 程式

10-6. Quantifying Environment Control Effectiveness

Quantifying the interactions of ventilation control, building design, animal population, and weather can be useful for more than determining the cost of mechanical ventilation.

Environment control in confinement animal housing involves a relatively limited set of environment modification alternatives. Ventilation is the primary and frequently the only means available to modify environment in an animal barn. As a consequence, there will be times during a year when the aerial environment in a typical barn is less than desired, and may even be such as to stress the animals. A means to assist a building designer to estimate the fraction of a typical year that a building will provide conditions within desirable limits would be useful.

An Environment-Control Effectiveness Index (EEI) can be defined as the fraction or percentage of a year when conditions within a housing space are acceptable for the animals within the space. Values of the index will range from 0.0 (never suitable) to 1.0 (always suitable).

The concept of "Acceptable Weather Space" (AWS) has been proposed (Cole, 1982) as a means to visualize weather conditions under which mechanical ventilation can be expected to provide suitable indoor air temperatures and relative humidities. The AWS concept can be understood by beginning with two more fundamental concepts, the Climate Space (CS) and the Production Space (PS).

An animal's CS is defined as the range of weather (environmental) conditions within which the animal can survive. Each type of animal has its own CS, and many animals (but obviously not all) tolerate a CS which is quite wide in terms of air temperature and relative humidity. However, in confinement animal agriculture, simple survival is not enough, animals must also be productive.

An animal's PS is defined as the range of conditions (typically air temperature and relative humidity) in which the animal will remain productive up to or nearly up to its genetic potential, assuming feeding is proper. That is, milk production will not be suppressed due to heat stress, feed conversion and growth will be near optimum, etc. In terms of temperature, the PS approximates the thermoneutral region for the animal in question. The PS is obviously more limited than the CS, but is a subset of it. For highest productivity, the ideal is to keep conditions always within the PS, although highest profitability may result if some deviation out of the PS is permitted during extreme weather.

It is unlikely a conditioned space can be held within a desired PS at all times unless sophisticated and expensive environment conditioning equipment is installed, to include: air conditioning, heating, ventilating, humidifying, and dehumidifying equipment. Environment design must be a compromise between the cost of total environment control and the cost of straying outside the PS occasionally. Practical experience has shown the usual best compromise in confinement animal housing is to have ventilation, evaporative cooling in warm and dry climates, and supplemental heat for small or sick animals, but seldom more. Thus, a barn with few options for environment modification will provide conditions within the PS for only part of the time, which will be quantified here through the Environment-Control Effectiveness Index (EEI).

Conditions inside a barn can be related to outdoor conditions through use of conventional energy and mass balances. For simplicity, at this point only ventilation will be considered, although the concept to be developed applies equally well when more extensive environment modification equipment is available, and to both confinement animal housing systems (barns) and controlled environment agriculture (greenhouses). The details, of course, become more complicated.

Simple, steady-state energy and mass balances for a ventilated airspace are:

$$q_s = (\Sigma UA + FP + \dot{m}c_p)(t_i - t_o) \qquad (10\text{-}11)$$

$$\dot{m}_p = \dot{m}(W_i - W_o) \qquad (10\text{-}12)$$

Equations 10-11 and 10-12 connect outdoor conditions to the resulting indoor conditions. The line in Figure 10-4(a) from point o (outdoors) to point i (indoors) can be termed a "conditioning line". For specific conditions at point i, fixed building thermal characteristics, unchanging animal population heat, and moisture production parameters (which is to say, a steady-state), only one outdoor combination of temperature and relative humidity will map to point i. The location of point o along the conditioning line is determined by the only parameter in Equations 10-11 and 10-12 which can change, the ventilation rate. If ventilation is high, point o is close to point i. If ventilation is limited, point o is not close to point i.

Each point i within a conditioned space can be uniquely mapped to an outdoor condition if the ventilation rate at point i is known. Of course, it is possible to find indoor conditions where mapping will take point o outside the realm of possible conditions. For example, moisture production in Figure 10-4(b) is zero and the sensible heat production is unchanged, thus point o must fall above the saturation line. The same is true if point i is at the dry bulb temperature shown, but near saturation (Figure 10-4(c)). Point i cannot be reached from any possible point o by using only ventilation (as long as the sensible heat production is not changed). Ventilation and humidification are required, with the amount of humidification determined by outdoor conditions.

Another assumption in this development is that the PS of the animal group being housed can be described by limits of relative humidity and dry bulb temperature. Assume (constant) lower and upper limits of both relative humidity and temperature. For example, a PS could be defined as between 15 and 25 C, and 30 and 80% relative humidity. Such a PS can be outlined on a psychrometric chart, as shown in Figure 10-5.

Staged ventilation will be assumed within the PS in Figure 10-5. Thermostat setpoints control the ventilation stages and can be superimposed onto the PS. A three-stage ventilation system with two thermostat setpoints, sp_1 and sp_2, is shown in Figure 10-6.

Individual points within the PS, as well as entire regions of the PS, map to uniquely defined outdoor conditions when the ventilation rate is fixed. For example, region 1 of the PS in Figure 10-6 maps as shown in Figure 10-7. One conditioning line, from c to c', is shown. The entire subregion could be mapped point by point or only the corner points could be mapped as a reasonable approximation. The width of the subregion of the PS is usually sufficiently narrow that only corner points are needed and relative humidity contours can be approximated as straight lines.

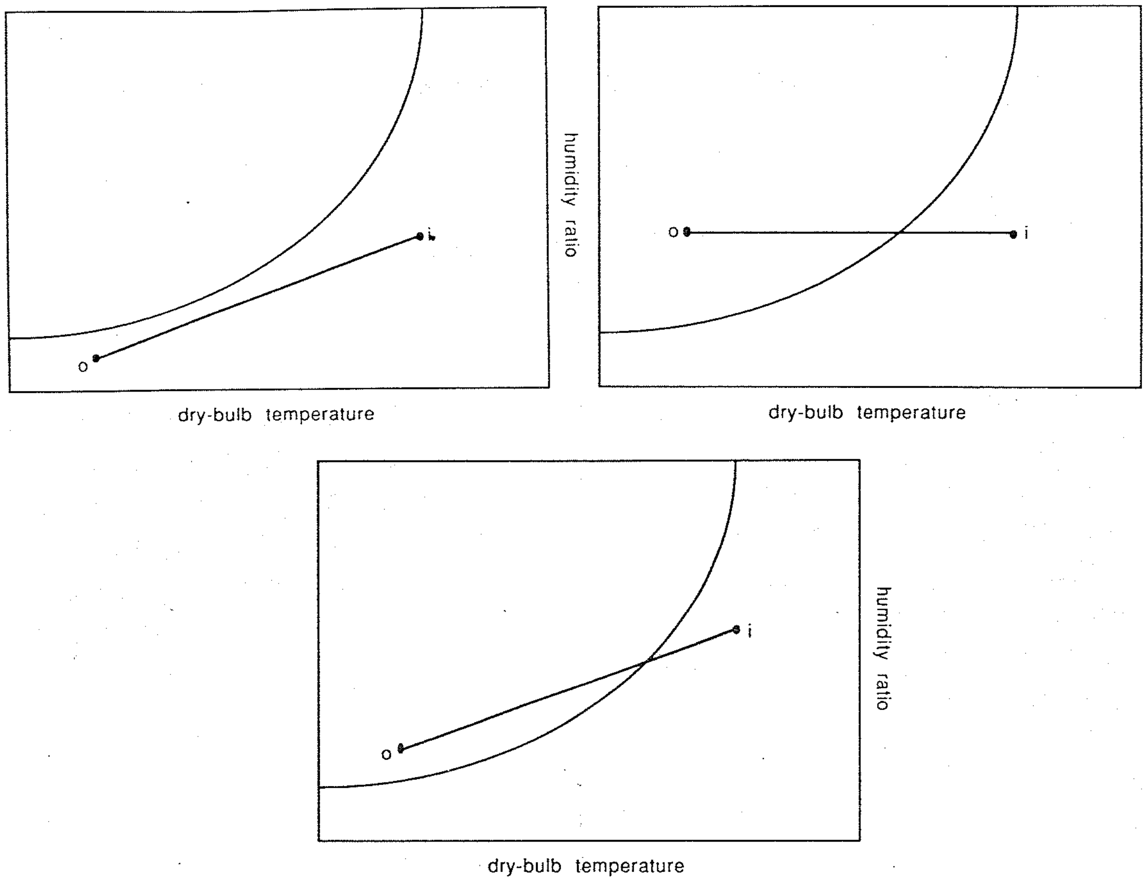

Figure 10-4. Sketch showing conditioning line between outdoor conditions of dry-bulb temperature and relative humidity (point o) and specified indoor conditions (point i). A possible situation is shown in (a). Impossible cases are in (b) and (c) where the outdoor conditions must be above the saturation line on the psychrometric chart to produce point i inside a building. In (b) there is no moisture production within the space, but there is sensible heat production. In (c) the desired indoor conditions are at too high a relative humidity, with insufficient moisture production, to achieve that state using only indoor air for ventilation. Thus, (b) and (c) demonstrate two causes of the same result, the PS can not be achieved by ventilation alone, for moisture production is too low.

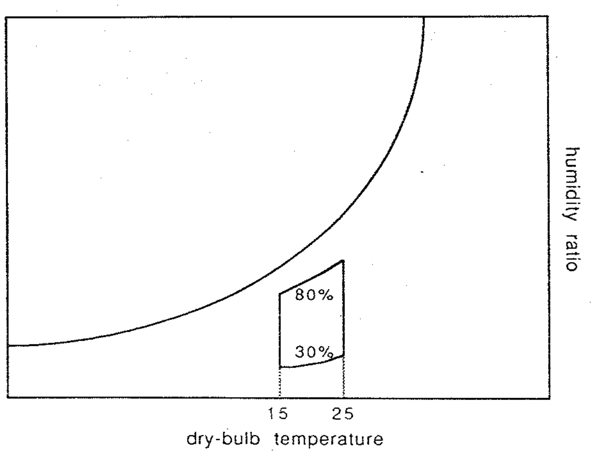

Figure 10-5. Sketch of Production Space defined by upper and lower limits of dry-bulb air temperature and relative humidity.

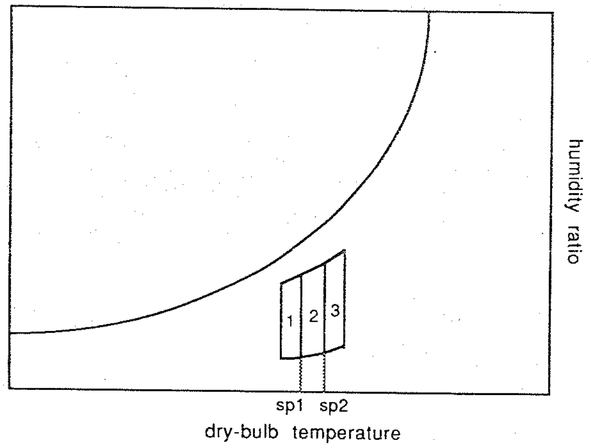

Figure 10-6. Sketch of Production Space demarcated by three ventilation stages and two thermostat setpoints.

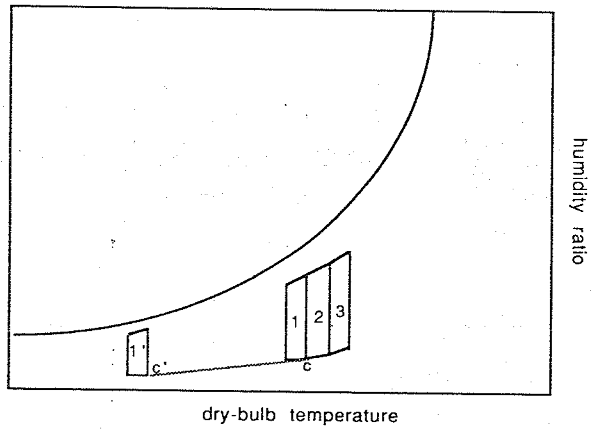

Figure 10-7. Sketch of how a single subregion of a Production Space maps to a region of outdoor conditions - a Weather Space.

The mapped region (1') is almost but not quite identical to region 1 of the PS. Animal heat and moisture production vary from point to point in subregion 1, thus conditioning lines which define the corner points of subregion 1' are not quite parallel and are not the same length.

The entire PS maps as shown in Figure 10-8 because each subregion of the PS is characterized by its own ventilation rate. In actual practice, it is likely the subregion of the PS having the highest ventilation rate will overlap the lower temperature region of the PS, and others may also. However, for clarity the mapped subregions in Figure 10-8 are separated from the PS.

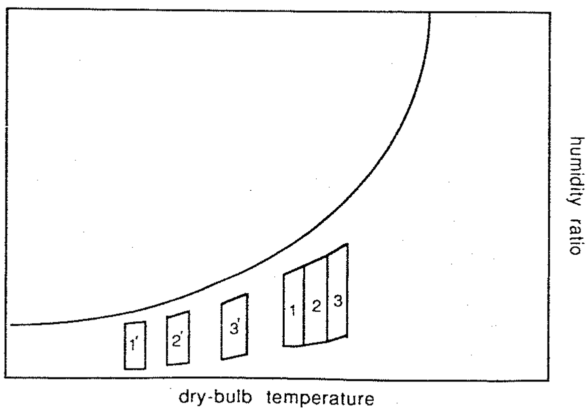

Figure 10-8. Illustration of mapping all subregions of a Production Space with staged ventilation.

A staged ventilation system has been assumed. At each thermostat setpoint a limit cycle is established and control is as shown in Figures 10-2 and 10-3. The result of cycling is to connect the mapped subregions shown in Figure 10-8 (1' to 3'). The right boundary of mapped subregion 1' is the same as the left boundary of mapped subregion 2', which is also true of the right boundary of 2' and the left boundary of 3'. Thus, the mapped subregions connect at their corner points by straight lines, as shown in Figure 10-9 (Cole, 1982).

In Figure 10-9, regions a and b are identified with times of limit cycling, while regions 1', 2', and 3' are identified with times of steady ventilation rates. What is important, however, is that regions 1', 2', 3', a and b form the outline of all possible outdoor conditions which map (permit) the indoor environment to be within the Production Space. This is an important point to realize. The implication is, whenever weather is outside the boundary of the mapped subregions, ventilation alone cannot provide conditions within the building which are inside the PS and production may be suppressed, with the degree of suppression being a function of how far actual conditions in the barn are from the PS. All points within the mapped subregions permit ventilation and operation within the PS. These subregions form what is termed the "Acceptable Weather Space" (AWS). Whenever weather falls within the AWS, indoor conditions are within the PS.

Next consider weather and its variation. Each combination of dry bulb temperature and relative humidity on the psychrometric chart can be identified with a probability of occurrence. Extreme conditions have a low probability of occurrence, while moderate conditions (e.g., 20 C and 50% relative humidity) have a relatively high probability in most temperate climates.

When a two-dimensional probability function, such as characterizes weather summarized as a probability density function on a psychrometric chart, is integrated over a region of its domain, the result is the probability of being within that region. If the probability function of weather is known, integrating over the mapped subregions in Figure 10-9 leads to what was defined previously as the Environment-Control Effectiveness Index (EEI).

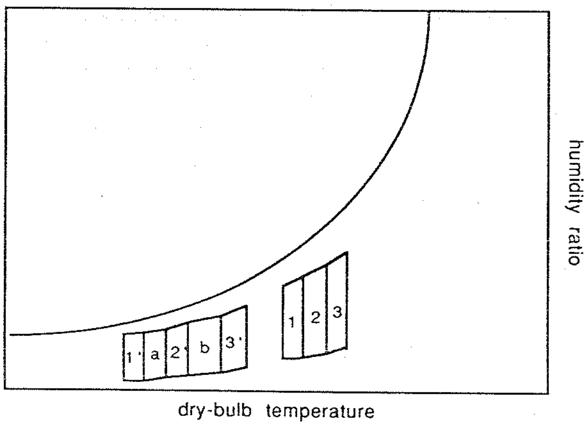

Figure 10-9. An illustration of how mapped subregions of a Production Space connect to form an Acceptable Weather Space.

A probability function to describe local climates cannot be readily found in analytical form, but can be obtained from weather records in discrete form – the probability of being at each of the combinations of dry bulb temperature and relative humidity which characterize the climate of the region of interest. As a practical matter, most climate zones fall within the ranges from - 30 to 40 C and 0 to 100% relative humidity. The probability function frequently will not be smooth and monotonically changing. Localized peaks and valleys may result from local weather peculiarities and other factors such as proximity to a large body of water. There frequently will be a peak near freezing at high humidity, a condition which characterizes times of snow and cold rain and nights during the spring and autumn in cool and moderate climates.

Hourly weather data today is sometimes available on a magnetic storage medium and, thus, can be summarized with some degree of ease into probability data. Few years of such data may be available, but as time passes the data will become more extensive. At least five years of data are suggested to approach a usable estimate of the long-term average, and 10 years is significantly better. It must be noted, however, that a designer wishing to obtain a probability function for a region of interest will likely find it necessary to generate one from local weather data. A probability function is not a format in which weather records are typically kept. Weather stations at land-grant universities and state experiment stations are possible sources of data.

10-6.1. Program WEATHER

Program WEATHER is provided in executable form to accomplish calculation of an EEI if probability data of dry bulb temperature and relative humidity is available. An example of such data is provided as data file WEATHER.DAT. The data characterizes a relatively humid climate having cold winters and moderate summers. The data are based on four years of hourly weather records from Ithaca, New York, and are only a rough estimate of the true data.

Program WEATHER is intended to illustrate EEI calculations; it is not a general purpose building design and evaluation program. The program calculates the EEI by following the sequence described in the previous section. The PS is defined by the user. Conditioning lines are calculated to define how subregions of the PS map to subregions of the AWS. Corners of the AWS subregions are connected using straight lines. The weather data file is read and the probability of being within the AWS is calculated.

The file WEATHER.DAT contains probability data for each combination of dry bulb temperature and relative humidity between - 30 and 35 C, and 0 to 100% relative humidity, as follows. If another data file is generated, it must conform to this format.

-30	0	0.0000000000
-30	1	0.0000000000
-30	2	0.0000000000
—	—	—
-30	100	0.0000000000
-29	0	0.0000000000
-29	1	0.0000000000
—	—	—
-20	58	0.0000293204
—	—	—
0	50	0.0000586407
0	51	0.0001466018
—	—	—
9	97	0.0011141735
—	—	—
35	100	0.0000000000

Each data line contains three items: dry bulb temperature (-30 to 35 C); relative humidity (0 to 100%), and the probability of that combination occurring (expressed as a decimal).

10-6.2. Example Use of WEATHER

This section presents an example to illustrate how WEATHER is to be used, and the results one can expect from it. It is best to follow the example on a computer, using WEATHER as we go along.

Example 10-5

Problem: Consider the barn described in Example 9-4, to be located near Ithaca, NY, and to house 62 milking cows averaging 550 kg body mass. A five-stage ventilation system is planned, with the ventilation stages described in the example (1.16, 1.74, 2.88, 4.24, and 8.96 m^3/s). The barn will be located at an elevation 300 m above sea level, and have ΣUA and FP values of 640 and 105 W/K, respectively. The Production Space is between 5 and 25 C, and 30 and 80% relative humidity. Thermostat setpoints of 8, 12, 16, and 20 C will be specified. Determine how well this design will provide the cows with conditions within the PS.

Solution: Program WEATHER and data files WEATHER.DAT and WEASPACE.DAT are needed to solve this problem. The first data file contains the weather probability data, while the second contains other input required by the program; for this example, data consists of default values in the file but can be changed if desired. The program permits this change part way through its sequence of menus.

Output from Program WEATHER can be obtained in several ways. One is as a printed summary, a copy of which is in Figure 10-10. Another is as a graph, a copy of which is in Figure 10-11. The graph is presented with axes of dry bulb temperature and relative humidity rather than the conventional axes of the psychrometric chart. To span the dry bulb temperature range from the lowest temperature of the AWS to the highest temperature of the PS, the relative humidity lines in the low temperature area of the graph would be compressed into such a small region as to be indecipherable.

A preliminary interpretation of the output might be that some error has occurred. The lowest subregion of the PS has been collapsed into a subregion of the AWS of zero height between temperatures of - 21.02 and - 16.40 C. This is, in fact, not an error. It instead indicates the environment control, as specified, is inappropriate at the lowest stage – the moisture removal capacity of a ventilation rate of 1.16 m^3/s is never sufficient in the lowest stage to keep the barn's relative humidity less than 80%. The other subregions of the AWS extend over a wide range of relative humidity and thus contribute to ventilation sufficiency. The net result is that the EEI is reduced somewhat by the ineffectiveness of the lowest ventilation stage. To summarize the control, the EEI (0.7363) shows the barn will provide conditions within the PS for slightly less than three-quarters of the hours in a typical year.

```
THE SUITABILITY INDEX IS:     0.7363   <<<<========

ANIMALS:                                    dairy
                                 Number:       62
                            Weight, kg:      550.0

BUILDING:                     elevation, m:   300.0
                            UA value, W/K:    640.0
                            FP value, W/K:    105.0

PRODUCTION SPACE:          Upper Temp, C:      25.0
                           Lower Temp, C:       5.0
                             Upper RH, %:      80.0
                             Lower RH, %:      30.0

VENTILATION:   5 Stages

    Stages, m3/s:               1.16       1.74       2.88       4.24       8.96
    Set Points, C:                   8.00      12.00      16.00      20.00

WEATHER DATA FILE:  WEATHER.DAT
OTHER DATA FILE:    WEASPACE.DAT
CHANGED DATA FILE:  WEASPACE.DAT

ACCEPTABLE WEATHER SPACE SUB REGION COORDINATES
-----------------------------------------------

                                      Temp.   Rel. Hum.

  Stage 1          upper left:       -21.02     0.00
                  upper right:       -16.49     0.00
                  lower right:       -16.40     0.00
                   lower left:       -20.95     0.00

  Stage 2          upper left:       -10.46    81.69
                  upper right:        -4.74    88.96
                  lower right:        -4.65     0.00
                   lower left:       -10.39     0.00

  Stage 3          upper left:         0.70   100.00
                  upper right:         6.01    98.85
                  lower right:         6.09     1.17
                   lower left:         0.77     0.00

  Stage 4          upper left:         8.80    96.76
                  upper right:        13.86    91.74
                  lower right:        13.93    17.54
                   lower left:         8.86    16.10

  Stage 5          upper left:        16.88    87.05
                  upper right:        22.65    83.55
                  lower right:        22.68    25.74
                   lower left:        16.91    26.06
```

Figure 10-10. Printed summary output of Program WEATHER for Example 10-5.

For comparison, the EEI was recomputed using slightly different ventilation stages: 1.8, 2.5, 3.5, 5.3, and 8.8 m^3/s, but with all other data the same. The resulting EEI is 0.7731 and the output graph is in Figure 10-12. With this change, even though the barn cannot hold temperature within the PS when it is colder than - 14 C outdoors, (compared to - 21 C in the base case) the EEI is approximately four percentage points better. This consists of a yearly equivalent of 322 hrs (= (0.7731 - 0.7363)8760 hrs/year) additional time in the PS, because the relative humidity criteria are satisfied at lower outdoor temperatures.

10-7. Ventilation Control in Greenhouses

In principle, the cost of ventilating a greenhouse could be determined using a sensible heat balance and determining times when fans would be needed. However, the procedure would be complicated by the interaction of solar gain and outside air temperature. Weather bin data which combine both solar insolation and air temperature are available from ASHRAE.

Greenhouses are not ventilated continuously during cold weather, thus in moderate climates the fan system duty factor should be less than the duty factor for animal housing in a similar climate. Also, temperature needs of greenhouse crops are sufficiently narrow that little opportunity exists to modify thermostat setpoints to save electricity. The primary opportunity to save electricity when

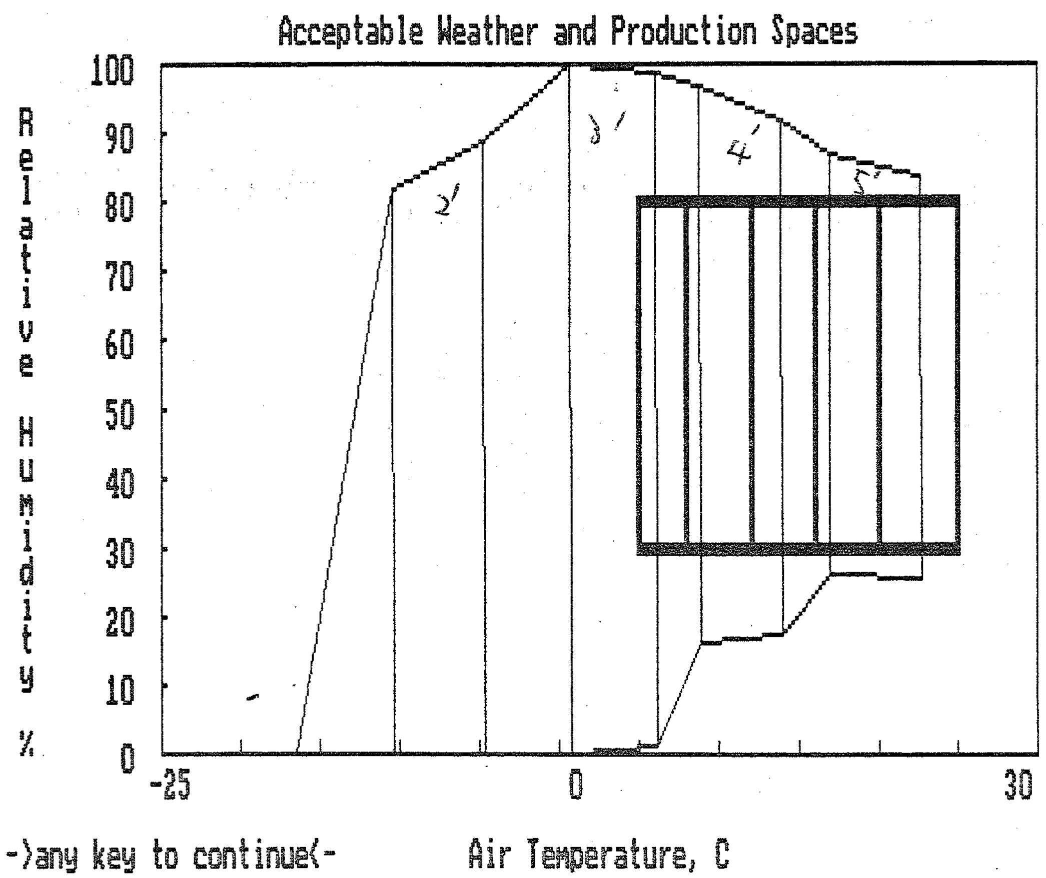

Figure 10-11. Copy of output graphics screen of Program WEATHER for Example 10-5.

ventilating greenhouses is to choose ventilating fans which are inherently energy efficient, and then maintain the fans to prevent a significant degradation of energy efficiency. That is, fan belts should be kept in adjustment, bearings should be oiled, louvers should be properly adjusted, etc.

A common recommendation for greenhouse fan systems is to have a three-stage system. The lowest stage is for winter ventilation and is typically only about 15% of the maximum installed ventilation capacity. Winter ventilation needs are small and too much air movement may cause severe localized chilling of plants. The second stage is typically approximately half the maximum ventilation rate.

Care is required to prevent simultaneous operation of both heating and ventilation in greenhouses. A setpoint for venting at least 3 K above the heating setpoint is recommended. Fan operation then begins at least 3 K above the heating setpoint. (In greenhouses, natural ventilation or "venting" may provide sufficient air movement during times of moderate need, such as during cool weather or warm and cloudy days with wind.) When computerized control systems are installed, where air temperature is always measured from the same sensor or sensors, the problem of simultaneous heating and venting may be avoided.

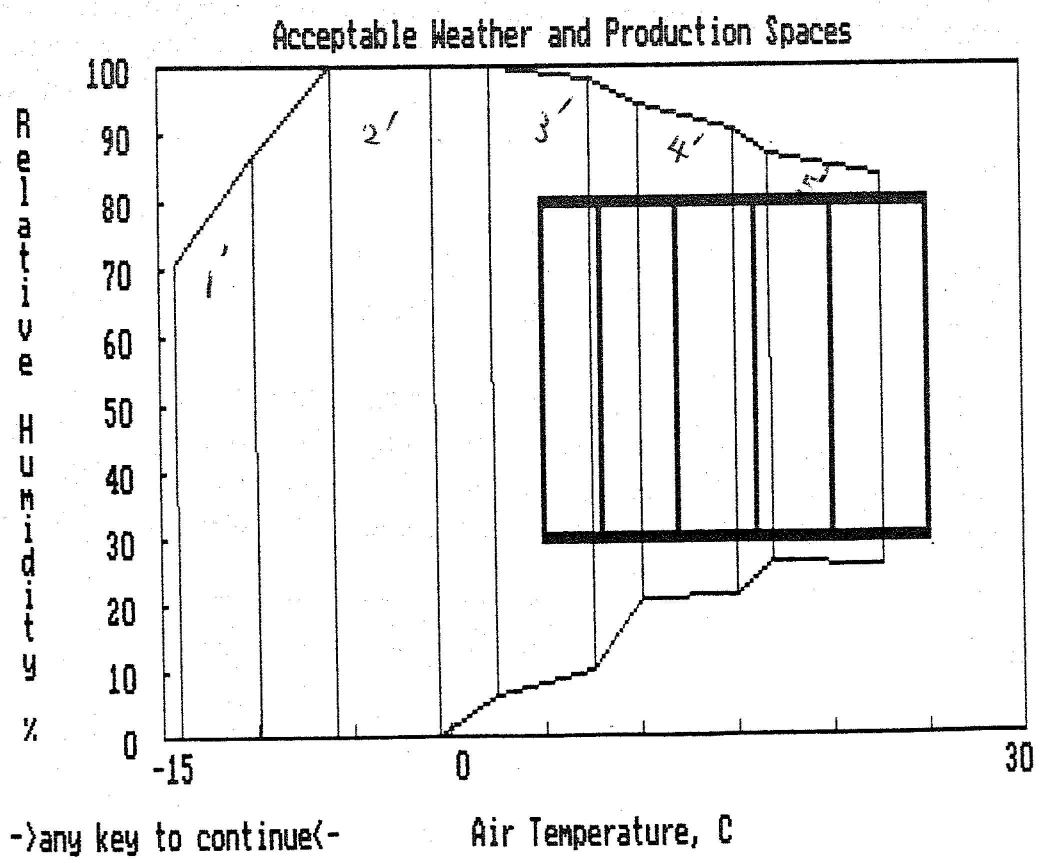

Figure 10-12. Copy of output graphics screen of Program WEATHER for Example 10-5, with ventilation stages changed.

SYMBOLS

a,b,c	coefficients for metabolic (and other) heat generation
A	area, m^2
c_p	specific heat, J/kg K
F	perimeter heat loss factor, W/m K
HLF	building heat loss factor, $\Sigma UA + FP$, W/K
m	mass flow rate, kg/s
$\dot{m}$	water vapor production, kg/s
P	perimeter, m
P[]	probability
q	heat generation or flow, W
t	temperature, C
U	unit area thermal conductance, W/m^2 K
$\dot{V}$	volumetric flow rate, m^3/s
W	humidity ratio, kg/kg
z	elevation above sea level

EXERCISES

1. A plant growth chamber is be ventilated with a two-stage ventilation system. When the temperature in the chamber is below 20 C, a small fan (0.5 m^3/s) is used. When the chamber's temperature is above 20 C, the smaller fan is inactivated and a larger fan (5 m^3/s) is used. The Ventilating Efficiency Ratio (VER) of the small fan is 2 m^3/s-kW, and the VER of the large fan is 5 m^3/s-kW. The heat source in the chamber is 2 kW of fluorescent lights (2 kW, including ballasts), and the UA value for the chamber is 60 W/K. Seventy-five percent of the heat load from the lights appears as sensible heat. The chamber is located outdoors near Fort Knox, Kentucky.

   From this data, estimate the yearly cost of electricity to ventilate this chamber if the unit cost of electricity is \$0.11/kWh.

3

2. A simple ventilating system for an animal housing facility has a single thermostat setpoint and two ventilation stages, 2 and 20 m^2/s. Each stage has a Ventilating Efficiency Ratio of 6 m^3/s-kW and electricity costs \$0.18 per kWh. Your calculations have shown the following duty factors

Setpoint, C	Duty Factor	Setpoint, C	Duty Factor
5	0.68	13	0.51
6	0.67	14	0.49
7	0.66	15	0.47
8	0.65	16	0.45
9	0.63	17	0.43
10	0.61	18	0.41
11	0.58	19	0.39
12	0.54	20	0.38

for the ventilation system as a function of the thermostat setpoint (to switch between the two stages):

The production of the animals is also a function of indoor air temperature (the setpoint). Data have shown net income (excluding the cost of ventilation) as a function of air temperature is as follows:

Setpoint, C	Net Income, $	Setpoint, C	Net Income, $
5	43,000	13	49,950
6	47,500	14	49,800
7	48,500	15	49,650
8	49,000	16	49,400
9	49,600	17	49,100
10	49,700	18	48,700
11	49,900	19	48,200
12	50,000	20	47,600

Based on these data, what is the thermostat setpoint for highest profitability?

3. A greenhouse operator in Eugene, Oregon has a separate small building for germinating seeds. Flats of soil are seeded, placed in this building under lights for 10 days, and then taken from the building where the seedlings are transplanted into pots and moved to the greenhouse.

The germination room is well insulated ($\Sigma UA + FP = 65$ W/K) and is ventilated using fans in four stages: 0.5, 1.0, 2.0, and 5.0 m^3/s (total for each stage). The heat load in the room is from lights; total wattage is 50 kW, and it is estimated 70% of the total heat is converted into sensible heat. The rest appears as latent heat and is removed from the room as humidity in the air. There are no other heat sources.

What will be the average ventilation rate (m^3/s) for the room if thermostat setpoints are 22, 24, and 27 C? What is the highest temperature the grower can expect to occur in the germination room during an average year? What fraction (or percentage) of the year can she not expect to maintain the 22 to 27 C temperature range?

4. This problem involves use of the program DUTYFACT. Consider an animal housing facility having a UA value of 850 W/K and an FP factor of 180 W/K. There are 1300 animal units in the barn, each producing sensible heat as a function of indoor air temperature as follows:

$$q(\text{watts}) = 58 - 1.33t - 0.0052t^2$$

where t is air temperature, C. The barn is located near South Bend, Indiana, at an elevation of approximately 200 m.

The calculated minimum ventilation rate is to be 0.01 m^3/s per animal unit, and the maximum will be 0.085 m^3/s. The fans available for use have the following performance data. Note: The airflow data are in cfm and the VER data are in cfm/watt. See the note at the end of the problem statement for conversion factors.

Model	cfm@0"	VER@0"	cfm@.05"	VER@.05"	cfm@.1"	VER@.1"
a	4192	16.4	3870	14.9	3461	13.3
b	4637	13.4	4372	12.5	4030	11.4
c	5769	21.0	5181	18.5	4305	15.1
d	6436	18.1	5926	16.5	5317	14.3
e	8496	22.7	7617	19.8	6481	16.4
f	9572	21.0	8796	18.7	8004	16.6

cfm/w

This problem is to estimate the yearly cost of ventilation for several possible control strategies. The base case is for six ventilation stages, with the stages incremented according to equal intervals of outdoor air temperature. The estimated unit cost of electricity will be $0.09/kWh.

Another possibility is to stage the fans according to even increments of ventilation rate using a six-stage system. Determine the yearly cost of ventilation for this possibility.

For both designs, use setpoints of 13, 16, 19, 22, and 25 C.

Think about why the difference in duty factors and costs should arise.

For selecting fans, assume there will be six fan banks. This should not dictate fan selection, but the expected number of fan banks should be kept in mind when selecting fans so the end results are approximately symmetrical among the fan banks.

Note: This problem and DUTYFACT are based on the SI system. Fan data is available today in the US in IP units. One way to approach the disparity is to convert all fan data to SI units. Conversion factors are:

$$2119 \text{ cfm} = 1.0 \text{ m}^3/\text{s}; \; 2.119 \text{ cfm / W} = 1.0 \text{ m}^3/\text{s-kW}$$

REFERENCES

Albright, L.D. and G.W. Cole. 1980. A procedure to estimate fan system duty factors. Transactions of the ASAE 23 (3): 661-670, 675. American Society of Agricultural Engineers, St. Joseph, MI.

Albright, L.D., A.N. Rousseau and B.L. Brockett. 1988. A method to quantify the suitability of mechanical ventilation to provide desired indoor conditions. Proceedings, Livestock Environment III, pp. 195-202. American Society of Agricultural Engineers, St. Joseph, MI.

ASHRAE. 1985. Bin and degree hour weather data for simplified energy calculations (5 computer disks). American Society of Heating, Refrigerating, and Air Conditioning Engineers, Atlanta, GA.

AMCA. Standard 210: Laboratory methods of testing fans for rating purposes. Air Movement and Control Association, Arlington Heights, IL.

Cole, G.W. 1982. A method to predict the performance of mechanically ventilated systems. Transactions of the ASAE 25(2):419-424. American Society of Agricultural Engineers, St. Joseph, MI.

Phelan, R.M. 1977. Automatic Control Systems. Cornell University Press, Ithaca, NY. 280 pp.

$15.1 \ cfm/W$

$= 7.126 \ \frac{m^3/s}{kW}$

$= 7.126 \times 2.119 \ cfm/kW$

改用台灣的氣象檔

重 run Example 10-5

		EEI
台北	692	54.9
花蓮	699	60.7
宜蘭	708	40.49
台南	741	65
高雄	744	75.9
嘉義	748	44.2
台中	749	56.2
阿里山	753	2.17
恆春	X	
台東	766	76
梧棲	777	61.03