CHAPTER 5
STEADY-STATE ENERGY AND MASS BALANCES

5-1. Introduction

Three important concepts underlie environmental analysis of buildings: (a) control volumes, (b) conservation of energy, and (c) conservation of mass. The concept of energy conservation is applied to sensible heat, and mass conservation is applied to latent heat (humidity) and gaseous (or other) contaminants. Both conservation concepts rely on use of control volumes.

This chapter will begin with descriptions of control volumes and energy and mass balances. It will then focus on the components of each balance in turn, and, finally, meld the components together into solutions to answer questions such as: What will be the inside air temperature? How much heat will be needed? How many fans will be needed for ventilation? What size should the cooling system be?

Agricultural buildings are typically characterized by having single airspaces. This simplifies the analysis computationally, but in concept everything to be covered also applies to multiple airspace buildings.

Control volumes. Processes in thermodynamics, fluid mechanics, and heat transfer can be viewed in two ways. One way is to examine a process with a focus on its internal features. The other is to enclose the process with an imaginary boundary and examine only what passes across the boundary. The second is the control volume approach and is commonly used in environmental analyses and system design.

Although, at first, the control volume approach may appear to simplify processes excessively (a black box approach), in practice it is a very powerful tool. Many processes can be analyzed without knowing the details of what actually happens within the control volume. In some ways, this is like what a depositor knows of a bank account.

A person with a bank account has little concern for the internal matters of postings, proofings, and other recordkeeping functions within a bank. Knowing if the same dollar which was deposited is later the one withdrawn is immaterial. Everything the customer needs to know about the account can be determined by knowing what goes into the account and what comes out. The balance is obtained knowing the data. The current balance figure which the bank provides is redundant information (and, of course, a check on the accuracy of the account owner's and the bank's records). Internal workings of the banking process are sufficiently complicated to prevent ordinary customers from determining their bank balances if the only information available is complete detail of the internal processes.

The same is true of certain thermal processes in buildings. Some processes can be analyzed knowing only what enters the airspace and what leaves it. Obviously a bank account is not a perfect analogy to building environmental control, but the comparison illustrates how processes can be described sufficiently by what crosses an imaginary boundary surrounding the process.

A simple example of applying control volume concepts is the following. Consider a forced hot air heating system in an older, single family house. Some of the heating ducts pass through the attic; the attic is otherwise unheated. Should the heating ducts be insulated? If they are, the attic will be colder and the lower attic temperature will increase heat loss through the attic floor from the heated rooms below. Will the increase of heat loss by conduction from the rooms below be greater than the savings from insulating the ducts? Will insulation be a wise investment or a mistake?

It would be a significant exercise to answer this question in detail by analyzing the heat transfer processes involved, that is, by concentrating on heat transfers from place to place within the house. Instead, mentally place a control volume around the entire house. When it is cold outdoors, heat escapes from all parts of the house and the rate of heat loss depends on the temperature difference from inside the house to out. If the attic is colder when the ducts are insulated, there will be less heat loss from the attic and, thus, less from the house. This can be stated with confidence although nothing is known about the rate of heat loss from the ducts before or after insulating or the insulation value of the floor between the attic and the rooms beneath.

Sensible energy balance. A simple set of sensible energy transfers for a single airspace building is shown in the sketch of Figure 5-1.

The energy flow terms are:

q_s: sensible heat gain from animals (or people) within the airspace.

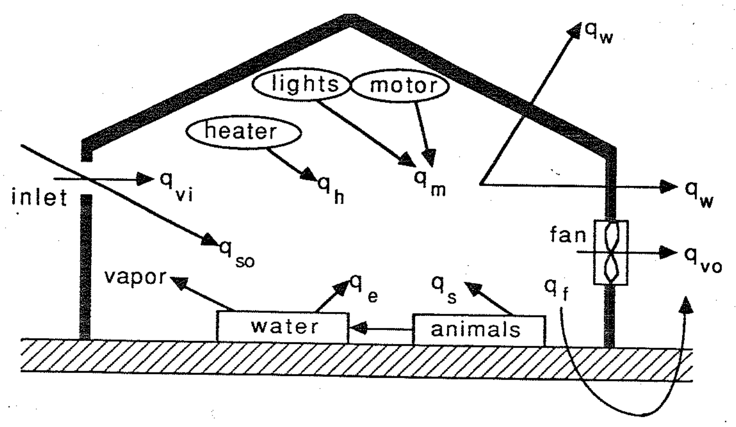

Figure 5-1. Contributors to a sensible energy balance in an animal housing building.

q_m: sensible heat gain from "mechanical" sources such as motors and lights. Such sources are usually electrical devices, and the heat gain is from conversion of electrical energy to sensible heat.

q_{so}: sensible heat gain from the sun. This can be gain through windows in a barn, and be relatively small, or it can be solar gain into a greenhouse and dominate all other heat gains.

q_h: sensible heat gain from a heating system.

q_{vi}: the sensible heat contained in the ventilation air entering the space referenced to some temperature datum. The datum is immaterial as, eventually, only the difference of sensible heat contents of the entering and exiting airstreams will be considered.

q_w: the transfer of sensible heat through the structural cover of the building; i.e., walls, ceiling, windows, doors, etc.

q_f: sensible heat transfer to the floor of the building primarily at the perimeter. It will be assumed that heat exchange with the floor in the interior of the building is relatively insignificant.

q_e: the rate of conversion of sensible heat to latent heat within the airspace. For example, evaporation of water from the floor of a barn or transpiration and evaporation of water from plants in a greenhouse are conversions of sensible to latent heat.

q_{vo}: the sensible heat contained in the ventilation air leaving the space referenced to the same temperature datum as q_{vi}.

The control volume for the energy balance is the air within the space bounded by the walls, floor, ceiling, and imaginary planes at the ventilation inlets and outlets. We need not specify the type of ventilation, or if total ventilation has a component of air infiltration through cracks. The arrows in Figure 5-1 indicate the assumed directions of heat transfers. The assumed directions need not be fixed, as long as the resulting energy balance is written to agree with the directions.

The general form of an energy balance for a control volume is

Gains - Losses = Change of Storage,

and if conditions are steady-state, there is no change of storage.

The steady-state sensible energy balance for Figure 5-1 rearranged in the form

Gains = Losses is

$$q_s + q_m + q_{so} + q_h + q_{vi} = q_w + q_f + q_e + q_{vo} \tag{5-1}$$

Mass balance. A simple mass balance for the same airspace is shown in the sketch of Figure 5-2. The same control volume is used as for the sensible heat balance.

The mass flow terms are:

m_p: the rate the material of interest (water vapor, carbon dioxide, etc.) is produced within the space.

m_{vi}: the rate at which the material of interest is carried into the airspace by ventilation air.

m_{vo}: the rate at which the material of interest is carried out of the airspace by ventilation air.

Several assumptions are contained within the sketch in Figure 5-2. Mass transfer by diffusion through the structural cover and floor is assumed to be sufficiently slow as to be negligible. The sources of the material of interest may be many but are shown as a collective whole. The material of interest is assumed not to undergo a transformation within the space to another material [such as the conversion of latent heat (in humidity) to sensible heat by condensation to form fog]. The only removal mechanism is ventilation; for example, dust in the air does not settle out onto the floor or adhere to the walls. Air in the space is well mixed.

The steady-state mass balance for Figure 5-2 is

$$m_p + m_{vi} = m_{vo}. \tag{5-2}$$

Individual terms of the energy and mass balances will now be examined in detail.

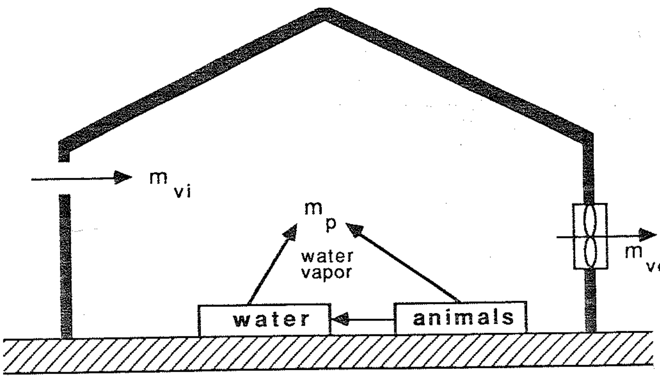

Figure 5-2. Mass balance for a single airspace.

5-2. Components of the Sensible Energy Balance

The energy balance can be used for many purposes. Heating or cooling required to maintain a predetermined air temperature can be calculated. The ventilation rate required to maintain design conditions can be found. The indoor air temperature in response to imposed conditions can be determined. The amount of insulation needed to limit heating may be of interest. However, before those calculations are possible, the components of the energy balance must be quantified. Some of the components have been discussed previously – some have not.

5-2.1. Sensible Heat Produced by Animals, q_s. Mammals are homeothermic creatures. That is, they attempt to maintain constant body temperatures (a state of homeothermy). Body temperatures are generally above those of the ambient air.

While poultry body temperatures are approximately 41 C, mammals of commercial importance in agriculture maintain body temperatures in the vicinity of 38 C. Internal physiological processes which maintain constant conditions are extremely complex and together are a condition termed "homeostasis". More complete descriptions of animal physiology and homeothermy can be found, for example, in Hellickson and Walker (1983), Esmay and Dixon (1986), and especially Clark (1981) and Curtis (1983).

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The first priority of homeostasis is to maintain body temperature. If there is not sufficient feed to support body temperature, growth, and production, then production and growth slow. During cold weather when more heat is needed to maintain body temperature, more feed is eaten if available. In extreme cold, growth may become negative when fat is metabolized to maintain body temperature.

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Sensible heat production is a by-product of maintenance, growth, and production; if conditions are extremely hot, production (e.g., milk or eggs) and feed intake are reduced to limit the quantity of sensible heat which must be expelled to the environment.

However, good agricultural practices mean ambient conditions should be regulated and sufficient food provided to the animals so all three functions can be supported with something approaching an optimal level. The temperature zone where this is possible is called the "comfort zone"; conditions outside the zone characterize "thermal stress".

Ambient thermal environment significantly affects animals, and the animals housed within a building significantly affect the environment within the building. A focus on the second effect is required to solve the energy balance in Equation 5-1.

Animals lose heat to their surroundings by radiative and convective transfer and, in certain circumstances, by conductive heat transfer (e.g., from a pig lying on the floor to the floor). Evaporative heat loss is a major component of total heat exchange, for expired air is nearly saturated with water vapor at body temperature. In cold weather, this means each breath expels a great deal of latent heat compared to the latent heat content of the ambient air.

Sensible heat is lost primarily from the outer surfaces of animals; latent heat is lost primarily from respiratory tracts. In general, farm animals do not have a sufficient number of sweat glands to make evaporation from the skin a highly significant factor.

Calorimetric studies have provided extensive data which can be used to estimate sensible heat production from commercially important farm animals. Such data is used widely, but limitations of the data should always be observed.

缺点3种 Calorimetric data does not contain the effect of evaporation of animal waste products from floors, etc., which is a conversion of sensible to latent heat. Thus, calorimetric data applied to a barn does not correctly estimate the amounts of sensible and latent heat actually added to the airspace.

Genetic differences of new breeds may affect heat and moisture production from animals. The best example of this is chickens, where the breeding process is ongoing and birds of a (human) generation ago were very different from today's birds.

The production level of the animals used to obtain the calorimetric data may have been very different from the production levels of the animals to be housed in the building being designed (for example, milk production from dairy cows). Different feeding and production levels lead to different sensible heat generation levels.

Finally, the ambient conditions used when the calorimetric data were obtained may have been significantly different from conditions expected in the animal housing being designed. For example, it is common to operate animal growth chambers and calorimeters at approximately 50% relative humidity. In winter, animal barns are frequently more humid than 50% and can be during summer drier. Total heat production from an animal may not be significantly affected by relative humidity (within reasonable ranges) but partitioning between sensible and latent heats will be.

However, much engineering design is accomplished using "best available" rather than "perfect" information. Such designs are adequate – adequate if sufficient safety margins are incorporated to guard against reasonable variations of material properties, operating conditions, and biological responses.

Fortunately, animals and plants have elastic responses to their environments. A column overloaded by a small amount for a short time may collapse, but a plant or animal kept a few degrees too warm or cold for a day will survive and respond to the improper conditions with only a temporary change of growth or production. Of course, if conditions become too extreme, the plant or animal will also "collapse" and may die or suffer long-lasting effects. Environmental control and alarms must be installed to prevent such extreme conditions and provide warnings of potential disasters.

Appendix 5-1 contains animal heat and moisture production data, abstracted from data standard D270.4 of The American Society of Agricultural Engineers (ASAE). Data are generally for a standard size of animal – an animal unit. For example, one animal unit for dairy cattle is a 500 kg cow. Many dairy cows are larger than 500 kg and some are smaller, so a means must be found to extrapolate heat production data to other animals of various sizes. P.403

It has been determined that mammalian sensible heat production is linearly correlated more closely to body surface area than body weight, and surface area and weight are not linearly correlated. The relationship between heat production and weight is such that heat production is proportional to weight raised to the power of 0.734. Thus, for example, a cow which weighs 20% more than another will produce $(1.2)^{0.734} = 1.14$ times as much heat.

此外指顯熱損失，非單位體重之熱損失

Example 5-l

Problem: Determine the sensible heat production from 100 (590 kg) dairy cows housed in a barn at 12 C.

Solution: Data in Appendix 5-1 for dairy cattle provide the needed information after interpolation and extrapolation. At 10 C a 500 kg dairy cow loses 1.5 W/kg of sensible heat and at 15 C she loses 1.2 W/kg. Interpolating linearly between these two values for heat production from one 500 kg cow at 12 C,

$$(q_s)_{500\ kg} = 1.2 + (2 / 5)(1.5 - 1.2) = 1.32 \text{ W/kg}.$$

A cow with a mass of 590 kg will lose heat at a rate greater than will a 500 kg cow, by the ratio $(590 / 500)^{0.734}$, the heat loss is

$$(q_s)_{590\ kg} = 1.32(500)(590 / 500)^{0.734} = 745 \text{ W/cow}.$$ → 1.26 W/kg for 590 kg cow

With 100 cows in the barn, total sensible heat addition to the air will be

$$q_s = (745 \text{ W/cow}) (100 \text{ cows}) = 74{,}500 \text{ W} = 74.5 \text{ kW}.$$ 2.43 W/kg for 50 kg cow

Note: The heat production data do not state the relative humidity at which the data was obtained. The original reference (Yeck and Stewart, 1959) states that humidities between 50% and 70% were maintained during their tests. We can be confident of our calculations only if expected relative humidities are within this range. In practice, barns are between 50% and 70% relative humidity much of the time.

For computerized analysis of building thermal environment, it is easier to use equations than tabled data. One means to express data is in polynomial form; the animal heat production data in Appendix 5-1 can be expressed in such a form. For example, sensible heat loss from 500 kg dairy cows is given at five air temperatures. The data can be expressed as a polynomial function of air temperature, and the function can have an order as high as four. However, it is best not to use high order expressions because of the possible lack of smoothness between data points. Generally, a second order polynomial suffices.

Example 5-2

Problem: Determine a second order polynomial which can be used to estimate sensible heat production from 500 kg dairy cows as a function of ambient air temperature.

Solution: The enclosed program POLYNOM was used for five data points and second order. A second order polynomial fit of the five heat production data points for dairy cows yielded the following expression:

$$(q_s)_{500\ kg} = 1.86 - 3.074E - 2t_{air} - 5.268E - 4(t_{air})^2$$

where t_{air} is air temperature in C and q_s is in W/kg.

To assess adequacy of the fit, the observed and predicted sensible heat production data can be compared.

Observed	Predicted
1.9 W/kg	1.9 W/kg
1.5	1.5
1.2	1.3
1.1	1.0
0.6	0.6

Agreement is not perfect but is likely to be adequate within our knowledge of other design parameters in a typical engineering design problem.

5-2.2. Mechanically Produced Heat, q_m. Lighting is the largest source of electrically generated heat in most agricultural buildings. Unless motors operate for a significant fraction of the time, their contributions are negligible.

To estimate heat production from lights, total electrical input must be determined. That is, incandescent lights produce heat approximately equal to the wattage of the installed lights. Fluorescent, metal halide, mercury, and sodium vapor lights add more heat than the installed wattage because of the power required to operate the ballasts (unless the ballasts are remote, as might be the case in a growth chamber, for example).

Motors are generally rated based on their outputs not their inputs. Small, single-phase motors have efficiencies typically in the range from 55% to 75%. Motors may not be used to their full rated capacity; thus, to estimate the heat addition from a motor, its expected load and efficiency must be known. Although some of the power delivered by a motor will ultimately be in the form of mechanical energy (for example, potential energy in feed raised from a storage to the birds in a poultry house), most of the electrical power input appears as heat from the motor and frictional heat along the mechanical linkages driven by the motor. Of course, heat from motors on exhaust fans is generally carried immediately outdoors and does not contribute to the building energy balance.

In many designs of environmental control systems for buildings, the addition of heat from lights and motors is assumed negligible as a worst case for winter designs and, at a maximum, as a worst case for summer designs (if not ignored entirely). In growth chambers and growth rooms in greenhouse operations, lights are the major source of heat within the airspace and must be carefully accounted for. In greenhouses where artificial lights are used, the lights may be sufficient to provide even a majority of the heat needed during cold weather, especially if movable insulation is used.

Example 5-3

Problem: A poultry building is lighted by fluorescent lights, designed for approximately 40 W of installed capacity per square meter of floor area. The barn is 12 m wide and 40 m long. Lights are the only significant source of electrically generated heat within the barn. Estimate the sensible heat production rate from this source.

Solution: With a floor area of 480 m^2 and lighting installed at 40 W/m^2, a total lighting capacity of 19,200 W is present. However, fluorescent lights have ballasts which typically draw approximately an additional 20% power. Thus, the heat released into the space will be 1.2(19,200 W) = 23,000 W = 23 kW.

5-2.3. Solar Heat Gain, q_{so}. When sunlight irradiates a glazing, three things may happen. The solar energy may be transmitted, it may be reflected, or it may be absorbed. If I is the intensity of solar insolation at a transparent surface, the three components are:

absorbed = αI, where α is absorptance,
reflected = ρI, where ρ is reflectance, and
transmitted = τI, where τ is transmittance.

Absorption: A simple model for absorption within a transparent material uses the extinction coefficient, K, which characterizes the material. The extinction coefficient is a measure of absorption,

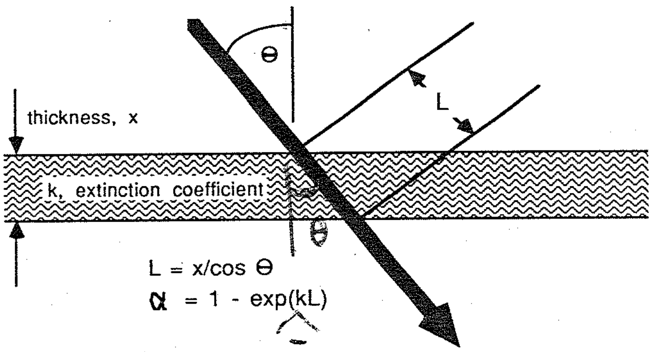
$$- dI_\lambda = I_\lambda K_\lambda dx, \tag{5-3}$$

where I_λ is the monochromatic intensity at a point along the insolation's path through the glazing, K_λ is the monochromatic extinction coefficient, dx is the differential path length, and dI_λ is the differential change of monochromatic intensity. Fortunately, most common glazings show little change of extinction coefficient over the solar spectrum, a wavelength range of 0.3 to 3 microns. The major error in this assumption occurs in the ultraviolet wavelength band which contributes only a little to the total energy within the insolation. (Note that many glazings have extinction coefficients for longwave thermal radiation which are very different for solar insolation. For example, there is the "greenhouse effect".)

If L is path length,

$$\alpha = 1 - \exp(- KL), \tag{5-4}$$

where K is not a function of wavelength within the solar spectrum, and represents an average over the spectrum. Extinction coefficients of typical materials are in Table 5-1.



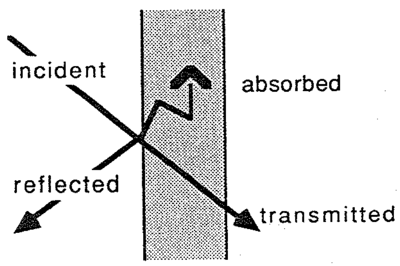


Table 5-1. Extinction coefficients for transparent materials.

Material	Extinction Coefficient, mm^{-1}
ordinary window glass	0.03 (approx.)
polyethylene	0.165
low-iron glass (<0.01%Fe_2O_3)	0.004 (approx.)
heat-absorbing glass	0.13 to 0.27
Tedlar[a] (polyvinyl fluoride)	0.14
Mylar[a] (polyethylene terephthalate)	0.205
Teflon[a] (fluorinated ethylene propylene)	0.06

[a]trademark of E. I. DuPont de Nemours, Wilmington, DE.

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If the angle of solar irradiation and thickness of the glazing are known and the extinction coefficient is estimated, absorptance can be calculated using Equation 5-4.

Example 5-4

Problem: Calculate the solar absorptance at normal incident angle for insolation passing through ordinary window glass 3 mm thick.

Solution: The extinction coefficient (from Table 5-1) is assumed to be approximately 0.03 mm^{-1}. Using Equation 5-4,

$$\alpha = 1 - \exp(-(0.03)(3\ mm)) = 0.0861.$$

Nearly 9% of the insolation is absorbed as it passes through the glass. This is in addition to insolation lost due to reflection.

Reflection. The index of refraction, n, determines the speed of light through a material, and also determines reflection from the surface. As shown in Figure 5-3, when light strikes a reflective surface of a transparent material at an angle of incidence, ϕ, specular reflection will also be at the angle ϕ. Light passing through the transparent material will be bent to an angle ϕ θ. The refractive index, n, is

$$n = \sin\phi/\sin\theta. \tag{5-5}$$

Snell's law relates the refractive index and surface reflectance, ρ. Reflectance has two components: one polarized parallel to the plane of incidence and one polarized perpendicularly. The two components are

$$\rho_{parallel} = \sin^2(\phi - \theta)/\sin^2(\phi + \theta), \text{ and} \tag{5-6a}$$

$$\rho_{perpendicular} = \tan^2(\phi - \theta)/\tan^2(\phi + \theta). \tag{5-6b}$$

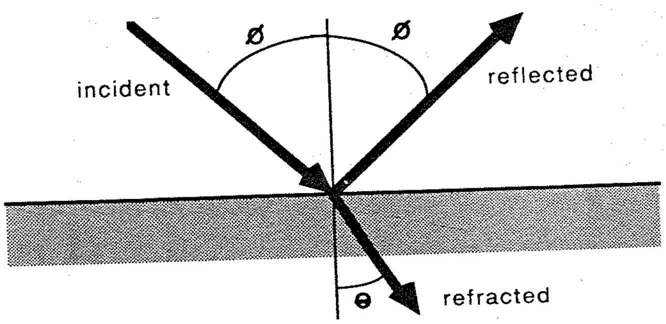

Figure 5-3. Angles of incident, reflected (specular), and refracted light on striking a transparent material.

When the incident radiation is perpendicular to the surface, the two components are equal and when the incident angle approaches 90 degrees each component of reflectance approaches 1.0. Table 5-2 contains refractive indices for common glazing materials. Example 5-5 illustrates a calculation of specular reflectance.

As an alternative means to estimate specular reflectance, an approximation of solar reflectance may be estimated using Figure 5-4. The graph applies to specular reflectance from a smooth glass surface.

Example 5-5

Problem: Determine the two components (parallel and perpendicular) of reflectance of light irradiating a glass surface at an angle of 50 degrees.

Solution: We will assume, based on Table 5-2, the refractive index is 1.50. The incident angle is 50°, thus, the refractive angle is

$$1.5 = \frac{\sin \phi}{\sin \theta} = \frac{\sin 50^\circ}{\sin \theta}$$

$$\phi = \arcsin(\sin 50° / 1.50) = 30.7°$$

Using Equations 5-6a and b,

$$\rho_{parallel} = \sin^2(50 - 30.7) / \sin^2(50 + 30.7) = 0.112, \text{ and}$$

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$$\rho_{perpendicular} = \tan^2(50 - 30.7) / \tan^2(50 + 30.7) = 0.003.$$

Sunlight is relatively unpolarized, thus,

$$\rho_{average} = (0.112 + 0.003) / 2 = 0.058.$$

Table 5-2. Refractive indices for light in the visible waveband.

Material	Refractive Index
air	1.00
window glass	1.50 to 1.55
Tedlar[a] (polyvinyl fluoride)	1.45
Mylar[a] (polyethylene terephthalate)	1.64
Teflon[a] (fluorinated ethylene propylene)	1.34

[a]trademark of E.I. DuPont de Nemours, Willmington, DE.

Transmittance. Transmittance, reflectance, and absorptance sum to unity. When absorptance and reflectance are known, transmittance may be readily calculated. However, the methods outlined above to determine absorptance and reflectance through a glazing are relatively crude. Multiple interreflections within the glazing change absorptance and reflectance values; Stokes' equations may be used to estimate the effects of these reflections,

$$\rho_{actual} = \rho_s\left(1 + \frac{\tau_s^2 (1 - \rho_s)^2}{1 - \rho_s^2 \tau_s^2}\right), \text{ and} \tag{5-7a}$$

$\rho_s (1 + \rho_s \cdot \tau_{actual})$

$$\tau_{actual} = \tau_s\left(\frac{(1 - \rho_s)^2}{1 - \rho_s^2 \tau_s^2}\right). \tag{5-7b}$$

The subscript, s, indicates values obtained for single passes of radiation (values obtained using Equations 5-6a and b). Absorptance for this situation is calculated using the fact that absorptance, reflectance, and transmittance sum to unity.

$$\alpha_{actual} = 1 - \tau_{actual} - \rho_{actual} \tag{5-7c}$$

Example 5-6

$= \frac{(1-\tau_s)(1-\rho_s)}{(1-\rho_s \tau_s)}$

Problem: Calculate the absorptance, reflectance, and transmittance of direct beam solar radiation striking 3 mm thick window glass at an angle of incidence of 50°. Estimate the amount of radiation reflected, absorbed, and transmitted if the intensity of solar radiation is 500 W/m^2.

Solution: Assume the window glass has a refractive index of 1.50 and an extinction coefficient of 0.03 mm^{-1}. The angle of incidence is 50°. The angle after refraction is 30.7° as calculated in Example 5-5. Thus, the path length of travel through the glass is

$L = 3 \text{ mm}/\cos(30.7°) = 3.49 \text{ mm}.$

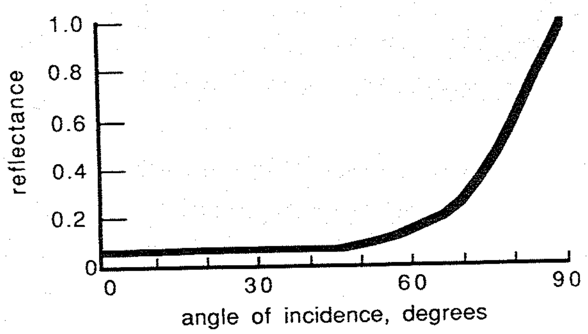

Figure 5-4. Reflectance (specular) for light incident on glass as a function of the angle of incidence.

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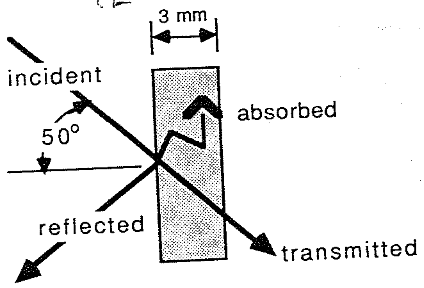

Absorptance for a single pass through the glass is

$$\alpha_s = 1 - \exp\left(-\left(0.03\ \text{mm}^{-1}\right)(3.49\ \text{mm})\right) = 0.099$$

(based on that which passes into the glass).

Reflectance for irradiation at this angle of incidence is 0.058 as calculated in Example 5-5 (assuming the insolation is unpolarized). By conservation of energy, the transmittance for a single pass once the radiation passes into the glass is

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$$\tau_s = 1.0 - 0.099 = 0.901.$$

Corrected for interreflections,

$$\rho_{actual} = 0.058\left(1 + \frac{0.901^2(1-0.058)^2}{1-0.058^2\,0.901^2}\right)$$
$$= 0.100, \text{ and}$$

$$\tau_{actual} = 0.901\left(\frac{(1-0.058)^2}{1-0.058^2\,0.901^2}\right)$$
$$= 0.802.$$

The overall solar absorptance is

$$\alpha_{actual} = 1.0 - 0.100 - 0.802 = 0.098 \text{ (based on irradiation).}$$

Solar insolation is 500 W/m^2, thus,

reflected radiation = (0.100) (500 W/m^2) = 50 W/m^2,
absorbed radiation = (0.098) (500 W/m^2) = 48 W/m^2, and
transmitted radiation = (0.802) (500 W/m^2) = 401 W/m^2.

The three sum to 499 W/m^2; the single W/m^2 which was lost in the process can be attributed to rounding effects. The transmitted radiation passes into the airspace and is the contribution to the energy balance of Equation 5-1, q_{so}.

5-2.4. Heating System, q_h. This energy term is often zero in many agricultural buildings, or it is the term in the energy balance to be determined and is unknown. If there is a predetermined heating contribution, its magnitude must be determined from the rating of the heating device, the expected level of average operation, and the heat addition is then used in the energy balance.

5-2.5. Ventilation, q_{vi} and q_{vo}. The two ventilation energy terms, q_{vi} and q_{vo}, will be considered together.

The mass flow rates of air at the inlet and exhaust are usually assumed to be equal. This is not strictly true because as air passes through a ventilated space moisture will likely be added and the mass flow rate of moist air exiting will be slightly greater than that entering. However, the difference is generally so small that a constant mass flow rate is usually assumed.

The change of sensible heat content of ventilation air is measured by its change of temperature. As we have seen before, the specific heat of air is approximately 1006 J/kg K and the heat added to the air is

$$q_{vo} - q_{vi} = 1006\rho\dot{V}(t_i - t_o), \tag{5-8}$$

where ρ, air density, is typically based on conditions at the inlet of the fan, $\dot{V}$ is the volumetric flow rate of air (also typically measured at the fan), and t_i and t_o are temperatures inside and outside the building, respectively. If conditions inside the building are known, specific heat corrected for humidity ratio (as described in Chapter 2, Psychrometrics) can be used in place of the value 1006 J/kg K.

5-2.6. Structural Heat Loss, q_w. Several of the concepts needed to calculate the heat transferred through walls, roofs, etc., have been seen in the chapter on heat transfer (Chapter 3) and the chapter on steady-state heat transmission through building boundaries (Chapter 4). We have seen the importance of R-values and the analogy to electrical resistance networks in calculating the overall conductive heat transfer through a thermal circuit.

The equation to calculate structural heat loss is

$$q_w = \sum_n (A / R)_n (t_i - t_o) \tag{5-9}$$

where there are n paths of transfer, each path is (most likely) a series thermal circuit.

Unit area thermal resistances are used in Equation 5-9. The factor $\sum(A / R)$ characterizes the overall conductance of the building shell and includes the effects of framing, windows, and doors. Procedures to calculate the overall conductance are shown in Examples 4-1, 4-2, 4-4, 4-5, and 4-6.

5-2.7. Heat Exchange with the Floor, q_f. The means to calculate heat exchange with a slab floor was developed in Chapter 4. The exchange can be calculated by

$$q_f = FP(t_i - t_o)$$

as presented in Equation 4-2.

Heat exchange with a basement wall and floor does not follow this model but such situations are rare in agricultural buildings and will be ignored for now. If such a situation exists, the heat exchange is treated as a constant value to be added to an energy balance. It is constant because soil temperature is viewed as unchanging.

5-2.8. Evaporation, q_e. Evaporation has two primary components in barns: evaporation from wash water and animal wastes, and evaporation from the animals (primarily from their respiratory systems). In greenhouses, evaporation is from the floors and benches when water is spilled, from the surface of the potting medium (or soil), and from transpiration by plants.

Data for heat produced from animals may or may not incorporate and obscure the conversion of heat from sensible to latent within the airspace. If the data are based on housing studies, they include conversion of sensible heat to latent (such as evaporation from the floor) within the space. If presented as calorimetric data, moisture production within the airspace must be estimated and sensible heat production decreased accordingly. The conversion of sensible to latent heat will depend on the housing system, waste handling system, and air temperature, and is not easy to estimate.

½ to air sensible heat

¼ to air latent heat

Latent heat production in greenhouses depends to a large extent on solar insolation, for transpiration is the major component of evaporation. As a rough rule, insolation which passes through the greenhouse cover can be partitioned - one half is converted immediately to sensible heat added to the air, one quarter is added to the air as latent heat, and one quarter is either reflected back to the outside (approximately 10%), used in photosynthesis (2% to 3%) or stored in the intrinsic thermal mass to be released later. (13% to 12%)

5-3. Uses of the Sensible Energy Balance

A sensible energy balance may be used for several design purposes. The ventilation rate to maintain specified inside conditions may be calculated. The inside air temperature which would result from ambient conditions and other factors may be determined. The rate of heat addition needed to maintain inside air temperatures during cold weather may be of interest. These can be found starting with the sensible energy balance.

Equation 5-1 can be rewritten in the following form:

$$q_s + q_m + q_{so} + q_h = \Sigma UA(t_i - t_o) + FP(t_i - t_o) + 1006\rho\dot{V}(t_i - t_o) + q_e. \tag{5-10}$$

The addition of solar heat, q_{so}, is kept in symbol form and would be calculated separately as the product of transmittance, glazing area, and solar intensity; less the fraction reflected back outside, used for photosynthesis if the building of interest is a greenhouse, and stored in the intrinsic thermal mass inside the building.

In common practice, Equation 5-10 is simplified for applications in barns and greenhouses.

Animal housing. When animal heat data are presented as net sensible heat production, the terms q_s and q_e are combined into one, which will be written as q_s and understood to be a net sensible heat addition. Transmitted solar input to barns is usually neglected. Heat from mechanical sources is frequently assumed to be negligible in barns. Finally, supplemental heat in barns (except when young animals are housed) is not frequently used. When these simplifying assumptions are accepted, the energy balance reduces to

$$q_s = (\Sigma UA + FP + 1006\rho\dot{V})(t_i - t_o). \tag{5-11}$$

The inside air temperature can be determined if other parameters are known,

$$t_i = t_o + q_s / (\Sigma UA + FP + 1006\rho\dot{V}). \tag{5-12}$$

and the energy balance can be rearranged to calculate the required ventilation rate to maintain desired conditions,

$$\dot{V} = \frac{q_s - (\Sigma UA + FP)(t_i - t_o)}{1006\rho\ (t_i - t_o)}. \tag{5-13}$$

Example 5-7:

Problem: Determine the ventilation rate (m^3/s) required to maintain a dairy barn at 15 C, given the following conditions:

elevation: 500 m
number of cows: 60
average weight: 580 kg
wall area: 274 m^2
wall R-value: 2.05 m^2 K/W
ceiling area: 520 m^2
ceiling R-value: 1.97 m^2 K/W
window area: 12 m^2

window R-value: 0.30 m^2 K/W
door area: 15 m^2
door R-value: 0.49 m^2 K/W
perimeter length: 110 m
perimeter heat loss factor: 1.5 W/m K
outside air temperature: - 5 C
inside air relative humidity: 70%
an exhaust ventilation system

Assume there is no significant heat addition from lights and motors and little solar heating of the barn can be expected. The attic is well ventilated. The animal heat data for dairy cows in Appendix 5-1 reflect net sensible heat after latent heat conversion has been deducted.

Solution: Equation 5-13 can be used, but several parameters are needed in addition to the data given.

A 500 kg dairy cow at 15 C can be expected to add 1.2 W/kg of sensible heat to the air in a barn (see Appendix 5-1). Cows in the example have a mass of 580 kg and there are 60 of them. Total sensible animal heat production is, thus,

$$q_s = (60)(1.2)(500)(580 / 500)^{0.734} = 40{,}000 \text{ W} = 40 \text{ kW}.$$

Air inside the barn is 15 C, with a relative humidity of 70%. The barn is at 500 m elevation; standard atmospheric pressure for that elevation is assumed (95.461 kPa). The program PLUS can be used to estimate air density, which is 1.14 kg/m^3.

Data given in the statement of the problem can be used directly to calculate structural heat loss; perimeter heat loss conductance is

$$FP = (1.5 \text{ W/m K})(110 \text{ m}) = 165 \text{ W/K}.$$

Heat loss conductances from the walls, ceiling, windows, and doors are:

$$
\begin{array}{lll}
\text{walls:} & UA & = A / R = 274 \text{ m}^2 / 2.05 \text{ m}^2 \text{ K/W} = 134 \text{ W/K}, \\
\text{ceiling:} & UA & = A / R = 520 \text{ m}^2 / 1.97 \text{m}^2 \text{ K/W} = 265 \text{ W/K}, \\
\text{windows:} & UA & = A / R = 12 \text{ m}^2 / 0.30 \text{ m}^2 \text{ K/W} = 40 \text{ W/K}, \text{ and} \\
\text{doors:} & UA & = A / R = 15 \text{ m}^2 / 0.49 \text{ m}^2 \text{ K/W} = 31 \text{ W/K}.
\end{array}
$$

The sum, ΣUA, is 470 W/K.

Equation 5-13 can now be applied to calculate the required ventilation rate, where the indoor to outdoor air temperature difference is 20 K,

$$\dot{V} = \frac{40{,}000 \text{ W} - (470 \text{ W/K} + 165 \text{ W/K})(20 \text{ K})}{(1006 \text{ J/kg K})(1.14 \text{ kg/m}^3)(20 \text{ K})}$$

$$= 1.2 \text{ m}^3\text{/s}.$$

Ventilation is by an exhaust system and this ventilation rate was calculated at the indoor conditions, thus, the fans must ventilate at a rate of 1.2 m^3/s (or 1.14 kg/m^3 x 1.2 m^3/s = 1.37 kg/s). Fans are normally specified according to their volumetric capacity.

Example 5-8

Problem: Assume the same barn and conditions as in Example 5-7. A minimum barn ventilation rate of 0.9 m^3/s has been determined as necessary to maintain healthy conditions for the animals. How cold must the outdoor air temperature be before it will no longer be possible to maintain 15 C inside the barn?

Solution: Equation 5-12 can be rearranged to solve this problem. Air density will be 1.14 kg/m^3 and sensible heat from the animals will be 40 kW at 15 C. Outdoor air temperature to satisfy the energy balance is

$$t_o = t_i - q_s / (\Sigma AU + FP + 1006\rho\dot{V}),$$

$$= 15\,C - \frac{40{,}000\ W}{470 W/K + 165 W/K + (1006\ J/kg\ K)(1.14 kg/m^3)(0.9 m^3/s)}$$

$$= -9\,C.$$

If outdoor air temperature is below - 9 C and the ventilation rate of 0.9 m^3/s is maintained, the indoor air temperature will fall below 15 C. If the outdoor air temperature is extremely low and the minimum ventilation rate continues, the barn could freeze. Fortunately, sensible animal heat production increases at lower air temperatures. This effect acts as a negative feedback to help prevent freezing. Of course, freezing is always possible if weather is sufficiently cold, thus, alarms or other freeze protection must be incorporated into the environmental control system. This problem will be explored more fully later.

Example 5-9

Problem: For the same conditions as in Example 5-7 (except the indoor air temperature), but with a ventilation rate of 2 m^3/s, determine the expected indoor air temperature.

t_i 未知，则 ρ，不可決定

Solution: Equation 5-12 applies. However, two factors prevent direct application of the equation: the dependence of air density on indoor conditions and the effect of air temperature on sensible heat production from cows. One way to approach the problem is to assume values for air density and heat production, determine the resulting air temperature and density, and iterate until the temperature, relative humidity, heat production, and air density are internally consistent. For now, the effect of ventilation on relative humidity within the airspace will be neglected and a constant relative humidity of 70% will be assumed.

The ventilation rate is higher than the value calculated in Example 5-7, thus it is reasonable to expect the indoor air temperature to be lower, air density to be higher, and more sensible heat to be available from the cows. Assume air density is 1.15 kg/m^3 and sensible heat production is 45 kW. A candidate value for indoor air temperature, from Equation 5-12, is

$$t_i = -5\,C + \frac{45{,}000\ W}{470 W/K + 165 W/K + (1006\ J/kg\ K)(1.15\ kg / m^3)(2 m^3/s)}$$

$$= 10.3\ C$$

Now the assumptions must be checked. At 10.3 C, air density will be 1.16 kg/m^3 and animal heat production (per animal unit) will be

$$q'_s = 1.5 + (0.3/5.0)(1.2 - 1.5) = 1.48\ W/kg.$$

Total sensible heat production from the herd will be

$$q_s = (60\ cows)(1.48\ W/kg)(500\ kg/cow)(580\ kg / 500\ kg)^{0.734}$$
$$= 49{,}500\ W.$$

Air density and animal heat production were estimated slightly too low. Final air temperature will be above 10 C, thus, animal heat production will not be quite as high as 49.5 kW. For the next iteration, assume air density is 1.16 kg/m^3 and sensible heat production is 48 kW. The next candidate air temperature is, thus,

$$t_i = -5\,C + \frac{48{,}000\ W}{470 W/K + 165 W/K + (1006\ J/kg\ K)(1.16 kg / m^3)(2 m^3/s)}$$

$$= 11.2\ C$$

At 11.2 C, air density is 1.16 kg/m^3 and heat production per kg for a 500 kg cow is

$$q'_s = 1.5 + (1.2 / 5.0)(1.2 - 1.5) = 1.4\ W/kg.$$

Total animal sensible heat production is

$$q_s = (60)(1.4)(500)(580 / 500)^{0.734} = 47{,}000\ W.$$

This is sufficiently close to the final environment. Further iterations would converge at an indoor air temperature of approximately 11 C, an air density of 1.16 kg/m^3, and a sensible heat production of approximately 47.5 kW. However, data for building thermal parameters and animal heat are not known with sufficient accuracy to make further precision meaningful. We can conclude that indoor air temperature with the new ventilation rate will be 11 C.

5-4. Components of the Mass Balance, Humidity

Although the mass balance of Equation 5-2 applies in principle to any component of air within a control volume, in practice it can be used only when the production rate of the component can be quantified. Further, in the mass balance the only environmental control parameter which can be determined is the ventilation rate, for ventilation is assumed to be the only removal mechanism for the component. These limitations simplify applications of the mass balance.

In animal housing, only data for humidity and carbon dioxide production are known with some certainty. Each has been found proportional to the number and type of animals and their sizes. Rates of production of other components such as dust, ammonia, or other gases are functions more of the waste handling system than of the animals and their sizes. Such rates have not been quantified in a way which applies in general and is useful for design.

Fortunately, it has been found that design to remove moisture and carbon dioxide usually provides sufficient ventilation to maintain the other components below levels of concern. In certain situations where another component rises to a level which is too high, the reason can often be traced to improper handling of the animal waste rather than improper design of the ventilation system and often can be cured by changing the way waste is managed.

Greenhouses usually are ventilated only for temperature control. High humidity during cold weather is tolerated; relative humidity often will be 70% to 80%. The cold surfaces of the glazing are dehumidifiers which prevent humidity from reaching saturation.

Moisture production within a greenhouse is a function of the plant population, light intensity, and water management practices of the grower and has not been quantified in a way useful for environmental design. For these reasons, although mass balance relations obviously apply to greenhouses, they are seldom used to design greenhouse environmental control systems. Only temperature is controlled.

5-4.1. Moisture Production From Animals. Data are presented in Appendix 5-1 for moisture production of various animals. Moisture production rates for dairy cows must be extrapolated if other than the 500 kg standard animal unit is anticipated (see Example 5-1). Data for broilers and swine are presented for various animal sizes and extrapolation should not be needed. Data for laying hens apply to 1.8 kg white leghorns, but that is the standard size of a laying hen. If different sizes are expected, the data may be scaled.

Example 5-10

Problem: Determine the rate at which moisture is generated and must be removed from a swine barn housing 100, 40 kg; 100, 60 kg; and 100, 80 kg pigs. Air temperature is expected to be 20 C.

Solution: Data from Appendix 5-1 may be used. At 20 C, moisture production is

Size, kg	Moisture Production mg/kg-s	mg/pig-s
40	0.61	24
60	0.47	28
80	0.39	31

which is, for 100 of each size pig,

$$\text{moisture} = 100(24 + 28 + 31)(1.0\text{E} - 6\ \text{kg/mg}) = 0.0083\ \text{kg/s}.$$

Example 5-11

Problem: Determine the rate at which moisture is generated and must be removed from a dairy barn housing 100 cows averaging 570 kg, when the indoor air temperature is 10 C.

Solution: By the data in Appendix 5-1, at 10 C moisture production of a 500 kg dairy cow is 0.28 mg/kg-s. Moisture production rate from a 570 kg dairy cow can be estimated as

$$\text{moisture} = 0.28(570 / 500)^{0.734} = 0.31\ \text{mg/kg-s}.$$

Note: This formulation for moisture production is assumed to mirror the form which has been found to represent sensible heat production from mammals.

Moisture production from the herd will be

$$m_p = 0.31\ \text{mg/kg-s}(570\ \text{kg})(100\ \text{cows})(1.0\text{E} - 6\ \text{kg/mg})$$
$$= 0.018\ \text{kg/s}.$$

0.01577

5-4.2. Ventilation, m_{vi} and m_{vo}. As was done for energy transport by ventilation in the sensible energy balance, m_{vi} and m_{vo} will be considered together. The moisture content of air equals the product of the humidity ratio and the mass of air. Entering air, carrying m_{vi}, has a moisture content of

$$m_{vi} = \rho_o \dot{V}_o W_o, \tag{5-14}$$

and expelled air, carrying m_{vo}, has a moisture content of

$$m_{vo} = \rho_i \dot{V}_i W_i, \tag{5-15}$$

where ρ_o and ρ_i are air densities at the inlets and outlets, $\dot{V}_o$ and $\dot{V}_i$ are the volumetric rates of airflow at the inlets and outlets, and W_o and W_i are humidity ratios at the inlet and outlet, respectively. This assumes well-mixed air.

ρ ✓

The products of air densities and volumetric flow rates are mass flow rates at the inlets and outlets, $(m_{air})_i$ and $(m_{air})_o$, and can be assumed to be equal. This is not strictly true, for the moisture content changes as air moves from the inlet to outlet, but the approximation is close. Ideally, entering air flows through planned inlets or fans and exhausted air through planned outlets or fans. Air leaks and wind and thermal buoyancy effects distort this idealization so Equations 5-14 and 5-15 are based on total entering and total leaving air, not just that entering and leaving through planned inlets and outlets.

5-5. Uses of the Mass Balance, Moisture

In most design situations, outdoor conditions are chosen based on design weather data and indoor conditions are specified based on the needs of the animals or plants within the space. If the moisture production rate, m_{water}, is known, the ventilation rate required to maintain indoor humidity at its design value may be calculated using a rearrangement of the mass balance, Equation 5-2.

$m_p + m_{vi} = m_{vo} \Rightarrow m_{water} = m_{air}(W_i - W_o)$

In practical terms, this calculation provides an estimate of the minimum ventilation rate. If a higher ventilation rate is necessary to maintain conditions at the design temperature, a humidity lower than design conditions will result.

$$m_{air} = m_{water} / (W_i - W_0). \tag{5-16}$$

This presents no problem, for design humidity conditions are usually chosen to be the maximum desired. If an exact humidity level is desired, irrespective of the ventilation needed for temperature control, moisture must be added to or removed from the air by mechanical means (a humidifier or a dehumidifier).

Example 5-12

[illegible] P403

Problem: A poultry house is being designed for 30,000 leghorn laying hens (1.8 kg, average). The poultry house is located in a region with an elevation of 1000 m. Determine the ventilation rate required to maintain the indoor air at 23 C and 70% relative humidity when it is - 20 C and 55% relative humidity outdoors.

Solution: Equation 5-16 applies. Data to quantify moisture production from the birds is in Appendix 5-1 and program PLUS may be used to determine the relevant humidity ratios.

At 23 C moisture production from 1.8 kg leghorn laying hens is estimated as

$$(m_{water})_{kg} = 0.97 + (5/10)\ (1.19 - 0.97) = 1.08\ mg/kg\text{-}s.$$

There are 30,000 hens, averaging 1.8 kg, thus, the total moisture production within the airspace is estimated as

$$m_{water} = (30{,}000)\ (1.8)\ (1.08) = 58{,}320\ mg/s = 0.05832\ kg/s.$$

The humidity ratios needed for Equation 5-16 are:

$$W_o = 0.000393\ kg/kg, \text{ and}$$
$$W_i = 0.013919\ kg/kg.$$

The required mass flow rate of air is

$$\begin{aligned} m_{air} &= (0.05832\ kg/s)/(0.013919\ kg/kg - 0.000393\ kg/kg) \\ &= 4.3\ kg/s\ (\text{dry air}). \end{aligned}$$

Correcting the required air mass flow rate for humidity is a sufficiently small change as to be disregarded considering the lack of precision in other design parameters, but will be included here for illustration.

Each kg of dry air holds 0.013919 kg of water, thus, the exhaust ventilation rate must be 4.3 + 0.013919 = 4.314 kg/s. If an exhaust ventilation system is used, the applicable air density is at indoor conditions and is 1.03 kg/m³. The volumetric ventilation is, therefore,

$$\dot{V}_{air} = (\overset{4.36}{\cancel{4.314}}\ kg/s)/(1.03\ kg/m^3) = \underset{4.23\ m^3/s}{\cancel{4.44\ m^3/s}},$$

should be

$4.3 * (1 + 0.013919)$
$= 4.3598517$

or 0.14 L/s-bird.

It would be a useful exercise to develop a graph for the required ventilation rate to maintain the indoor design conditions as a function of the outdoor temperature at several outdoor relative humidity values. For example, develop the graph for the required ventilation rate for outdoor temperatures between - 30 and 20 C, at outdoor relative humidities of 0%, 50% and 100%. Explain any difficulties which might arise.

5-6. Components of the Mass Balance, Carbon Dioxide

Carbon dioxide is frequently ignored as a design parameter for animal housing in general engineering practice. Experience has shown that ventilation to control temperature and moisture is usually sufficient to control carbon dioxide (and other gaseous contaminants of the air). However, carbon dioxide is an asphyxiant and should be of concern at high levels. The level for concern, however, is not clear. Rules exist for human exposure to carbon dioxide. For example, long exposure to levels over 10,000 ppm should be avoided. Some engineering recommendations for animal housing suggest a maximum continuous exposure level of 2500 ppm. The carbon dioxide level where health and productivity are affected is not known; the level of 2500 ppm was chosen to be sufficiently low that effects would not be anticipated. Further research is required before safe levels will be known with certainty.

Carbon dioxide is frequently added in greenhouses when light levels are high and there is little or no venting. At low light levels, light is limiting and carbon dioxide is not; ambient carbon dioxide suffices for growth.

The amount of carbon dioxide required for optimal photosynthesis is open to question. Common practice is to supplement the level to somewhere in the range from 800 to 1500 ppm. Sensing carbon dioxide within the plant canopy is important. If air movement is limited within the canopy, carbon dioxide depletion may be serious even though the level measured outside the canopy seems sufficient. For example, carbon dioxide levels as low as 150 to 200 ppm have been detected within dense plant canopies on bright days. Photosynthesis is seriously affected in this range. Fans to mix the greenhouse air have been found to help prevent depletion within canopies.

5-6.1. Carbon Dioxide Produced By Animals. Carbon dioxide is a by-product of metabolism, as is heat and moisture. The ratio between carbon dioxide production and total animal heat production is fixed by the biochemical processes of metabolism. Additional carbon dioxide may be produced from decomposition of wastes and digestive processes of ruminants, but such additional production is small compared to that respired by the animals.

One liter of carbon dioxide is produced, on the average, for every 24.6 kJ of total heat added to the environment by an animal. Total heat production of animals can be estimated using data in Appendix 5-1.

5-7. Uses of the Mass Balance, Carbon Dioxide

The minimum ventilation rate required to maintain a specified carbon dioxide level in animal housing is constant over the range of possible outdoor conditions. The ambient carbon dioxide level is approximately 345 ppm.

Example 5-13

Problem: Consider the poultry house described in Example 5-11. Calculate the carbon dioxide level when ventilation is 4.3 m^3/s.

Solution: The mass balance of Equation 5-2 applies when rearranged to solve for the carbon dioxide concentration within the airspace. Well-mixed conditions are assumed and the concentration of carbon dioxide in the exhaust air will be the same as the concentration within the space.

The mass of carbon dioxide in air can be calculated by

$$M_{CO_2} = M_{air}(C_{CO_2})_{mass},$$

where M is mass and $(C_{CO_2})_{mass}$ is the mass concentration, kg/kg, of carbon dioxide.

The carbon dioxide partial volume in air can be calculated similarly by

$$V_{CO_2} = V_{air}(C_{CO_2})_{volume},$$

where V is volume and $(C_{CO_2})_{volume}$ is the volumetric concentration, m^3/m^3, of carbon dioxide.

Carbon dioxide production depends on total heat production, which at 23 C is

$$q_{kg\ of\ bird} = 6.8 + (5/10)\ (6.6\text{-}6.8) = 6.7\ W/kg.$$

The 30,000 birds average 1.8 kg, thus, total heat production within the building is

$$q_{total} = (30{,}000)(1.8)(6.7) = 361{,}800\ W.$$

The conversion factor from total heat to carbon dioxide production leads to a carbon dioxide production rate of

$$\dot{V}_p = (361.8\ kW)/(24.6\ kJ/L) = 14.7\ L/s = 0.0147\ m^3/s.$$

The carbon dioxide balance, when expressed on a volumetric basis, is (for constant pressure)

$$\dot{V}_p + \dot{V}_{vo} = \dot{V}_{vi},$$

where

$$\dot{V}_{vo} = \dot{V}_{air}(C_{CO_2})_{volume},$$

$$= (4.3\ m^3/s)\ (C_{CO_2})_{volume}, \text{ and}$$

$$\dot{V}_{vi} = \dot{V}_{air}(C_{CO_2})_{volume} = (4.3\ m^3/s)\ (0.000345),$$

$$= 0.001\cancel{484}\ m^3/s,$$

for ambient air at 345 ppm carbon dioxide concentration.

The balance for carbon dioxide is, thus,

$$0.0147 \text{ m}^3/\text{s} + 0.001484 \text{ m}^3/\text{s} = (4.3 \text{ m}^3/\text{s})(C_{CO_2})_{\text{volume}},$$

and V_p outdoor indoor

$$(C_{CO_2})_{\text{volume}} = 0.003764 = 3764 \text{ (approximately 3800) ppm.}$$

This concentration of carbon dioxide is well below the 5000 ppm limit used by OSHA for human occupation. However, if research were to discover that a lower limit is preferred, a minimum ventilation rate might be based on carbon dioxide rather than moisture control.

This example could also be approached using mass rather than volume and it would be a useful exercise to do so. The factor to convert carbon dioxide volumetric to mass concentration is the ratio of molecular weights of carbon dioxide to air, which is 1.519. The mass concentration of carbon dioxide in ambient air is, thus, approximately 524 mg/kg.

345 × 1.519 =

Example 5-14

Problem: A greenhouse at sea level with a volume of 3000 m^3 experiences an air infiltration rate of 0.75 air changes (ac) per hour. Carbon dioxide is added to enhance plant growth. At what rate (kg/s) is carbon dioxide lost from the greenhouse by exfiltration when the greenhouse carbon dioxide level is 1000 ppm?

Solution: For contrast to Example 5-12, this example will be solved using mass rather than volume. Assumptions are needed, as is usually the case in engineering design. Assume the greenhouse air will be at approximately 20 C and 70% relative humidity (typical), and the air will be well-mixed.

Air enters the greenhouse with a carbon dioxide concentration of 524 mg/kg, and leaves with a concentration of 1519 mg/kg. The approximate density of air at 20 C and 70% relative humidity at sea level is 1.18 kg/m^3. The mass flow rate of air through the greenhouse is, thus,

$$m_{air} = (1.18 \text{ kg/m}^3)(3000 \text{ m}^3)(0.75 \text{ ac/hr})(3600 \text{ s/hr})^{-1},$$
$$= 0.7375 \text{ kg/s}.$$

The change of carbon dioxide concentration is 1519 mg/kg - 524 mg/kg, or 995 mg/kg. Thus, the loss of carbon dioxide, in terms of mass, is

$$m_{vo} = (0.7375 \text{ kg/s})(995 \text{ mg/kg}) = 733.8 \text{ mg/s} = 0.0007338 \text{ kg/s},$$
or 2.64 kg/hr.

By comparison, carbon dioxide uptake by a greenhouse crop with a full canopy in good sun should be approximately 0.7 mg/m^{2}s. A greenhouse with a volume of 3000 m^3 will have a floor area of approximately 1000 m^2. The carbon dioxide uptake by the crop will, thus, be approximately 0.0007 kg/s, which is approximately the rate at which carbon dioxide is lost by exfiltration.

0.7 g/s

42 g/分

SYMBOLS

A	area, m^2
C	concentration, ppm
F	perimeter heat loss factor, W/mK
I	irradiation, W/m^2
K	extinction coefficient, mm^{-1}
L	thickness, m or mm
m	mass flow rate, kg/s
m	rate of mass production, kg/s
M	mass, kg
n	index of refraction
P	perimeter, m
q	heat transferred, or produced, W
R	unit area thermal resistance, m^{2}K/W
t	temperature, C
U	unit area thermal conductance, W/m^{2}K
V	volume, m
$\dot{V}$	volumetric flow rate or volumetric production rate, m^3/s
W	humidity ratio, kg/kg
x	spacial variable, m
α	absorptance
ϕ	angle of refraction
λ	wavelength, microns
ρ	density, kg/m^3
ρ	reflectance
τ	transmittance
ϕ	angle of incidence

EXERCISES

1. Determine the sensible, latent, and total heat production from a flock of 30,000 white leghorn laying hens (averaging 1.9 kg body mass) housed at 24 C.

2. Determine the coefficients for a second order polynomial to represent the sensible heat production of the laying flock described in Exercise 1 above, including the heat added by lights. Lighting within the housing space is

designed at 40 W/m^2, and the building has a floor area of 1200 m^2. Compare predicted heat production to the data used to determine the coefficients of the polynomial.

3. Determine the coefficients of a second order polynomial to describe the carbon dioxide production from a herd of 100 dairy cows (averaging 560 kg body mass) as a function of air temperature.

4. Direct beam solar radiation strikes a single glazed window (ordinary window glass 3 mm thick). What is the greatest angle of incidence at which at least half the light passes through the glass?

5. A swine barn housing 1000 growing pigs (averaging 55 kg body mass) is located at an elevation of 800 m. Desired indoor conditions are 18 C and 60% relative humidity. Develop a graph of the required ventilation rate for moisture control (m^3/s, based on inside conditions) for outdoor conditions of 80% relative humidity and a temperature range of - 20 to + 5 C.

6. For the conditions described in Example 5 above, develop a similar graph of ventilation rate as a function of outdoor temperature when ventilation is calculated based on temperature control. Assume the (UA+FP) value for the building is 800 W/K.

7. For the barn described in Examples 5 and 6 above, will supplementary heat be required to maintain the barn at 18 C indoors when it is - 20 C outdoors? If so, how much?

8. For the conditions described in Example 5 above, develop a graph of supplemental heat required as a function of outdoor air temperature if ventilation is for moisture control at all times.

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