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WARM SEASON HEAT TRANSFER MODEL FOR DAIRY CATTLE IN NATURALLY VENTILATED BARNS OF VARIED DESIGN

By

Richard Robert Stowell

A DISSERTATION

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ABSTRACT

WARM SEASON HEAT TRANSFER MODEL FOR DAIRY CATTLE IN NATURALLY VENTILATED BARNS OF VARIED DESIGN

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Richard Robert Stowell

Improved methods of predicting when cows will experience heat stress are needed. A model is presented that can be used to characterize the static, steady-state thermal energy status of a row of dairy cows located within a naturally ventilated building prior to any decline in feed intake or milk production. Characteristics of the cattle, design features of the barn and weather conditions are primary inputs that can be varied in the model.

The computer model ANTRAN was written in FORTRAN for use on PC based computer systems. A two-dimensional analysis was employed to evaluate heat and mass flows from the animals for specified environmental conditions. The model is comprised of equations and other model components available from previous research in the animal and physical sciences, as well as engineering. Model results were compared to data from controlled environment studies on dairy cattle at the University of Missouri to validate the model. The use of ANTRAN was then demonstrated by modeling two extreme summer conditions for a row of cows in barns of varied design.

Modeled hair and skin surface temperatures compared closely to those measured from cows in a controlled environment, to within 0.5 °C for conditions where air speeds in the animal zone were about 2 m/s. Differences between the modeled heat transfer rates and those measured in environmental chambers were attributed to the

decline in feed intake, and thus metabolic heat production, of the cows in the controlled environment study. The cows in that study were exposed to hot conditions over several days, whereas ANTRAN predictions result from modeling over a shorter time frame.

Sample trials using the model ANTRAN under both 'hot & breezy' and 'calm & muggy' mid-Michigan summer conditions compare variations of several barn design alternatives, including barn width, sidewall height, sidewall open area and roof insulation. Model output is shown in the form of heat loss of the animals and the resulting thermal energy status of the cows. Heat loss is segregated according to the mechanism of heat transfer. The modeled impacts of varied parameters on heat flow and net heat load on the animals are also discussed.

To my family and friends in the Lord Jesus Christ

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TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

LIST OF MODEL VARIABLES

Symbol	Description	Basic units
A	area	m^2
C_v	opening effectiveness	---
D	mass diffusivity of water	m^2/s
d	diameter	m
F	view factor	---
H	height	m
h	heat transfer coefficient	$W/m^2\,K$
h_m	mass transfer coefficient	m/s
h_{fg}	latent heat of vaporization	J/kg
I	analogous current (flow)	W or kg/s
J	radiosity	W/m^2
k	thermal conductivity	W/m K
L	length	m
M	mass	kg
$\dot{m}$	mass flow rate	kg/s
Nu	Nusselt number	---
p	pregnancy state	days
Q	volumetric flow rate	m^3/s
q	heat transfer rate	W
R	analogous resistance	W/K or s/m^3
Re	Reynolds number	---
s	sweating rate	kg/s
T	temperature	K
t	thickness	m
U	free stream velocity	m/s
v	air velocity	m/s
W	weight	kg
w	humidity ratio	kg/kg
x	distance	m
Y	milk yield	kg

Symbol	Description	Basic units
ε	emissivity	---
ρ	density	kg/m^3
σ	Stefan-Boltzman constant	$W/m^2\,K$
θ	attack angle	degrees
ν	kinematic viscosity	m^2/s

Subscript	Description
a	ambient air
b	body core
cs	hair coat surface
cond	conduction
"cond"	effective conduction
conv	convection
diff	diffusion
eff	effective
evap	evaporation
hc	hair coat
i, n	integer count
lat	latent
m	mass
meta	metabolic
rad	radiant
resp	respiration
sk	skin
s, surf	body surface
swt	sweat
t	trunk
tiss	peripheral tissue
va	vapor in air
vasc	vascular
vsat	vapor at saturation
w	water
∞	free stream

1 - INTRODUCTION

Heat stress is a major concern for dairy farmers trying to maintain production and profitability in an increasingly competitive industry. Losses in milk yield, reproductive efficiency and herd health can be substantial during warm weather.

Even the best of available indices of heat stress suffer from incompleteness. Also, the usefulness of indices in designing common dairy facilities (generally, naturally ventilated buildings lacking precise environmental control) is quite limited. While the use of naturally ventilated dairy facilities without significant environmental control may suggest overly simplistic design, economic realities have determined that such facilities are the current housing standard even for progressive operators with high producing and often large herds.

A substantial research need exists to determine which design features of these barns most affect the cows' thermal environment and under what conditions additional investment in these features is appropriate. To accomplish this, the barn and enclosed cows must be evaluated as a system with emphasis placed on the thermal state or energy balance of the productive component of the system, the cows.

2 - OBJECTIVES

The overall goal of this work was to develop a tool to use in evaluating barn design features based on the housed animals' potential for heat loss compared to their normal metabolic heat load. Specifically, the objectives of this research were:

1) To develop a computer model that calculates the thermal energy state of a single continuous row of cows in a naturally ventilated barn given the heat transfer characteristics of the animals, pertinent structural features of the building and ambient summer weather conditions.

2) To partition cows' potential for heat loss amongst the prevailing heat transfer mechanisms.

3) To model the impact of some primary barn design features on the heat exchange capacity of dairy cattle for selected summer weather conditions in Lansing, MI.

3 - REVIEW OF LITERATURE

The underlying assumption in any study of this nature is that, for every animal, environmental conditions exist wherein the animal's full genetic productive capability is realized provided that nutritional needs are met. Over the last half-century, a substantial amount of research has been conducted in an attempt to define optimal environments for livestock production and to quantify the production losses that result in actual practice from non-ideal environmental conditions.

3.1 Physiology and the Thermal Environment

Homeostasis. The physiological nature of warm blooded species is to achieve a state of homeothermy; i.e., to maintain a constant body temperature. Dairy cattle, like most other animals and man, are dynamic beings possessing physiological mechanisms for responding and adjusting to their environment. These mechanisms are only completely effective in maintaining homeothermic conditions within some limited range of environmental conditions. When conditions are outside of this range, the cow's body temperature must change and productive capabilities are diminished.

Thermoneutral zone and thermal stress. Brody (1948) and Curtis (1983) provide thorough analyses of temperature regulation processes. Metabolism provides animals their primary internal source of heat. Metabolic heat is released when food is oxidized during digestion. Animals digest food to maintain bodily functions, to support growth, to produce additional products or work, and to reproduce, generally in that order of priority.

Every animal has a unique thermoneutral or comfort zone. Within this zone of environmental temperatures, an animal under basal (maintenance needs only) conditions generates near-minimum quantities of metabolic heat to maintain homeothermy. Also, within this temperature zone, physical adaptations allow normal activities to continue with minimal change in metabolic activity. Hey (1974) adds the criterion that body temperature remains normal and that sweating mechanisms are not activated.

When defining an appropriate environmental temperature, the temperature of the ambient air is generally used. Ambient air temperature is not an explicit indicator of the environment, however, since the thermoneutral zone is defined by an energy balance and the mechanisms of heat transfer are only partially determined by ambient air temperature. Environmental temperature is really a perceived temperature that may vary considerably from that of the ambient air depending on the nature of the surroundings and the animal.

The size and location of the thermoneutral zone for the mature dairy cow is primarily a function of milk production level, although differences in breed, weight, pregnancy and individual animals also exist (Kibler and Brody, 1950a and 1951; Ragsdale, et al., 1950). Extensive study of cows in environmentally controlled conditions at the Animal Psychroenergetic Laboratory at the University of Missouri revealed that the thermoneutral zone for lactating Jersey and Holstein cows is approximately 40 to 60 °F (Kibler and Brody, 1949). The comfort zone of dairy cattle encompasses temperatures that are considerably cooler than those for non-ruminants.

Thermal stress has been defined as any environmental situation that provokes an adaptive response to regulate body temperature (Curtis, 1983). Under this broad definition, livestock in real surroundings are constantly under some level of thermal stress. Thermal stress is commonly regarded as any situation in which physical adaptations are unable to maintain a constant body temperature. During these conditions, animals must make chemical adaptations and innately sacrifice productive activities for self-preservation.

When environmental temperature is below the thermoneutral zone, animals increase feed consumption and metabolic heat production to keep the body temperature from falling. Usually, growth and desired performance can be maintained at temperatures near thermoneutral in this process. At even colder temperatures, as well as at temperatures above the thermoneutral zone, change in metabolic activity is unable to offset the thermal effects of the environment without sacrificing growth or production.

If the environment becomes extremely cold, or if cold conditions prevail for extended periods of time, body temperature will begin to fall and the animal must take more drastic actions to survive. Most cattle are extremely cold tolerant. Webster (1974) states that when cattle are in good condition and have adequate access to food, it is almost impossible to impose acute cold stress so severe that heat loss exceeds the ability to maintain metabolic rate and, consequently, homeothermy.

Unlike under cold conditions, most animals have very limited capability to relieve the stress on the body from a hot environment without sacrificing production and the resulting margin between stress-free and life-threatening conditions is comparatively quite small. Hot environmental conditions can impose substantial stress on dairy cattle.

3.2 Consequences of Heat Stress in Dairy Cattle

Thermal stress due to hot conditions will be referred to as heat stress throughout the remainder of this text. Fuquay (1981) presents a thorough review of the effects of hot weather on animal production. Growth, conception and survival rates are negatively affected by hot environmental conditions for almost all livestock species (Mangold, et al., 1967; Nienaber, et al., 1985 and 1987). Production almost always declines because the animal reduces feed intake in order to reduce heat production (Kibler and Brody, 1953; Johnson, et al., 1962).

Milk production losses. The negative effects of heat stress are a prominent concern for many dairy operations. According to Ragsdale, et al. (1948, 1949 and 1951), feed consumption, milk production and body weight fall off gradually above 70 °F for both Jerseys and Holsteins, then rapidly decline at environmental temperatures above 75 to 80 °F for Holsteins (a large, higher producing breed) and above 80 to 85 °F for Jerseys (a smaller, lower producing breed).

Johnson (1965) delineates the effects of heat stress on high producing cows. As milk production has risen rapidly over the last half-century, absolute and percentage losses in milk production are now even greater during hot weather. Today, herd-wide losses of 30 percent or more in daily milk production during periods of persistent hot weather are not uncommon.

Production increases that are achieved from the normal use of bovine somatotropin (BST) do not appear to be diminished, nor do they induce any added thermal stress on the cow during hot weather, even though the feed intake of injected cattle is higher than that of control animals (Johnson, et al., 1991 and Lotan, et al., 1993). Higher efficiency of conversion of feed to milk in animals given BST may explain why they are not under greater thermal stress. A marked decline in milk production was still evident in all high producing cows exposed to hot, oppressive conditions whether or not the cows were receiving BST.

Poor reproductive performance. Conception rates of swine (Mahoney, et al., 1970) and cattle (Gwazdauskas, et al., 1973 and 1975) are reduced in hot environments. Research has demonstrated that heat stress shortens and reduces the intensity of expression of estrus, reduces the efficiency of fertilization and increases early embryonic mortality. Thatcher and Collier (1986) present a fairly complete description of the effects of heat stress on reproduction in the dairy cow.

Declines in milk production and reproductive performance can be attributed to the survival instincts of the cow. To maintain its body temperature, the cow may be forced to discontinue non-essential heat generating processes in addition to inducing greater evaporative cooling from its respiratory and body surfaces (Kibler, et al., 1949 and McDowell, et al., 1969).

Other health problems. Several other problems exist during hot weather. In an attempt to increase evaporative cooling, cows often congregate around waterers. They will even go so far as to splash themselves with drinking water or lie in manure-filled alleys. Aside from the undesirable implications for those handling and milking such cows, flies and pathogens seem to find the combination of warm, moist and dirty conditions around lethargic cows a haven for growth. Hogan, et al. (1989) measured an increased prevalence of environmental mastitis cases in commercial dairy herds during the summertime. The environmental conditions that commonly prevail during hot weather and the stressed nature of cows during such times likely contribute to this greater incidence of mastitis.

The negative impacts of heat stress on health and reproductive performance are more difficult to describe and assess in economic terms than are milk production losses, but they are thought by many to be longer lasting and of equal or greater economic cost. Fortunately, modifications that improve the environment for milk production often alleviate these problems as well.

3.3 Measures of Heat Stress and Their Application

Generally, indices of heat stress are either related directly to the animal involved, in terms of productive characteristics, physiological parameters or behavior, or to the surrounding environmental conditions. McDowell (1962) assessed the application and appropriateness of many such indices.

Milk yield. Declines in feed intake and milk production are longstanding measures of heat stress in livestock. Almost every research project involving heat stress will begin by detailing production losses. Yousef and Johnson (1966) reported that dairy cows reduced their feed intake, heat production and milk yield by 27%, 22% and nearly 70%, respectively, in a 32 °C environment compared to levels at 18 °C.

Many dairy operators realize their cows are experiencing heat stress only after feed intake and milk production are noticeably reduced. The financial implications of lost milk production during and after heat stress suggest that production will retain its dominant position as an indicator of heat stress in field situations.

Production is fairy easily measured and is commonly monitored. The response to rising environmental temperatures is readily observed. Disadvantages of using production measures, however, include the substantial thermal and time lags before a response is shown, the influence of other non-thermal factors, and the fact that limited causal information is generated.

Physiological measures. Physiological parameters are often examined to determine the heat tolerance of animals and man. They can also be used as indicators of heat stress, especially amongst animals possessing similar heat tolerance characteristics that are housed in varied thermal environments. Several physiological parameters have been considered for use as evidence of heat stress.

Metabolic heat production. Metabolic activity is a realistic indicator of thermal stress. Accurate measurement of heat production requires use of calorimetry in a controlled environment. In practice, heat production can be estimated by measuring feed intake and composition.

Body temperature. In his review of research on heat stress in dairy cattle, Bianca (1965) concluded that, if properly used, deep-body temperature is the best single physiological criterion of heat tolerance (or conversely, state of heat stress) in cattle. Also, McDowell (1962) writes, "The research to date shows rather clearly that for engineering studies the best indicator of animal comfort is body temperature."

Although Curtis (1983) reports that the temperature of blood in the aorta is the best single indicator of average body temperature, rectal temperature is the most commonly measured and employed body temperature estimate. Kibler and Brody (1954a) stated that, "rectal temperature is the best available index of an animal's ability to maintain a balance between heat production and heat dissipation." Brody, et al. (1955) provide rectal temperature data for dairy cattle from various regions of the United States, (average "Midwest normal" was 101.6 °F) and correlated rectal temperatures with several other body temperatures.

Some other temperatures may also be used to represent internal body temperature with reasonable accuracy. One is tympanic membrane (ear drum) temperature. Wiersma and Stott (1983) developed a technique to measure tympanic membrane temperature. Korthals, et al. (1995) and Macaulay, et al. (1995) analyzed temperature regulation in swine and dairy calves, respectively, by monitoring the temperature at this location. Another body temperature measure is intravaginal temperature. Zartman, et al. (1983) illustrate the feasibility of measuring intravaginal temperature remotely in dairy cows.

Respiratory activity. Respiration rate has frequently been used to assess heat stress. Webster (1974) postulates a simple clinical test of a cow's state of thermal comfort; if respiration rate is greater than 80 per minute the animal is very warm. In general, elevated respiratory activity indicates that the animal is not maintaining its heat balance and, being overheated, is trying to restore the balance (McDowell, 1962).

Other physiological parameters. The pulse rate of dairy cattle decreases with increasing temperature while skin and hair coat surface temperatures are elevated.

Pulse rate is not well-correlated with heat stress, however, displaying considerable variation amongst individuals and being strongly influenced by other stressors. Roller and Goldman (1969) concluded that pulse rate is almost useless as an indicator of heat stress in swine. Skin temperature would be a good indicator of stress because all heat loss from the body surface must be transmitted via a temperature gradient across peripheral tissue. Unfortunately, difficulties in obtaining timely, accurate measurements of skin and hair temperature limit their value as indices of stress.

Blood composition, as measured by the acid-base balance (Blincoe and Brody, 1951; Dale and Brody, 1954) and volume (Dale, et al., 1956) have been used to delineate the level of stress present in dairy cows in hot conditions. Leucocyte (white blood cell) counts in milk and blood progesterone levels have also been used for this purpose (Wiersma and Stott, 1969).

Some physiological parameters, such as body temperature and metabolic heat production rate, are accurate indicators of a heat stress condition. The usefulness of most physiological parameters is limited in a number of ways, however. Most are not easily or accurately measured in the field, especially if continuous readings are desired. If not measured frequently, changes in readings are not revealed until productive processes are already in decline so monitoring of feed consumption and milk production provide similar information in dairy cattle.

The application of telemetry, the transmission of remote sensor measurements via radio waves, is rapidly improving the usefulness of basic physiological parameters in characterizing animal stress. A major advantage of telemetry systems is that measurements can be taken from unrestrained animals (Rawson, et al., 1965). Reece and

Deaton (1969) and Nakamura, et al. (1993) provide examples of using telemetry to monitor livestock temperatures. Harris and Siegel (1967) additionally measured heart rate with the body temperature of chickens by telemetry.

Another limitation of physiological measures is that they are truly responses to the environment, providing little causal information. They may be quite appropriately used to calibrate and test other indices that are more causal in nature.

Environmental measures. Whereas physiological parameters indicate thermal stress by measuring the animal's response to the environment, environmental measures provide information regarding what conditions elicited an animal response.

Ambient air temperature. Most investigations of thermal stress or seasonal variation in livestock health and production begin with consideration of the effects of ambient temperature. The effect of ambient air temperature on the growth of various livestock species is discussed by Ragsdale, et al. (1957), Hellickson, et al. (1967) and Curtis (1983), among others.

In their summary of Psychroenergetic Laboratory research, Yeck and Stewart (1959) provide abundant data on the responses of dairy cattle to varied environments. Many of the studies that are summarized in their report correlated responses with ambient air temperature. Key reports on the effects of temperature include:

Ragsdale, et al. (1948) - milk production and feed consumption;
Kibler and Brody (1949) - heat production and cardiorespiratory activity;
Kibler, et al. (1949) - heat production and cardiorespiratory activity;
Thompson, et al. (1949) - water consumption;
Thompson, et al. (1951a) - insensible weight loss and moisture vaporization;
Thompson, et al. (1951b) - hair and skin temperature and partitioning of heat loss;
Blincoe and Brody (1951) - blood composition; and
Thompson (1954) - heat exchanges in dairy barns.

Yousef and Johnson (1966) and McDowell, et al. (1969) provide additional evaluations of the effect of high ambient temperature on dairy cattle function and production. In summary, hot environmental conditions lower feed intake, heat and milk production, and pulse rate, while raising respiration rate and rectal temperature.

Humidity. For many environmental conditions, air temperature alone cannot adequately describe the resulting animal response. Several environmental factors may need to be considered jointly or incorporated with ambient temperature into an index of stress. Humidity is often the primary variable included in such measures.

Riek and Lee (1948) performed some of the earlier measurements of physiological and behavioral reactions of dairy cows to hot, humid conditions; reporting that, as ambient temperature rises, the proportion of heat lost via evaporation rises rapidly. Thompson, et al. (1951a) reported that the ratio of evaporative cooling to heat production increased from about 10 percent at 0 °F to nearly 100 percent at 100 °F. Subsequent research by Ragsdale, et al., Kibler and Brody, and Thompson, et al. (all in 1953) found that productive processes, temperature regulation and vaporization of moisture of the cow, respectively, are not affected by humidity at low temperature, but are impacted to an increasingly greater extent by humidity as ambient temperature rises above 65 °F.

In a later study, dairy cattle were exposed to warm environments of varying temperature and humidity to determine heat and vapor dissipation rates (Cargill and Stewart, 1966) as well as skin and hair temperatures (Shanklin and Stewart, 1966) . It was concluded that increasing humidity at high temperatures increased heat stress in the cows considerably.

Logically, a measure that combines the effects of ambient temperature and humidity would facilitate improved assessment of the potential for heat stress. In the design of buildings for humans, ASHRAE (1989) makes use of an effective temperature ET* to describe the thermal environment and any stresses imposed by the environment. ET* is a rationally derived index that equates various psychrometric (temperature-water vapor) conditions of air that produce similar heat loss from the skin. Comfort zones overlain on the psychrometric chart are available for a variety of conditions. Aside from its application for humans, there has been interest given toward developing effective temperature relationships for swine in hot environments (Beckett, 1965), but not as much interest for dairy cattle.

Another commonly used thermal comfort index for hot weather is the heat stress index, the ratio (on a percent basis) of the evaporative heat loss from the skin required for thermal equilibrium to maximum possible heat loss to the environment. Other temperature and humidity indices include (Rosenberg, et al., 1983): humiture, temperature humidity or comfort index, humidex, and sultriness. There are also indices available to evaluate the evaporative potential from sweaty or wet human skin.

Several simple indices have been developed for livestock species. Isotolerance curves for man and some livestock species using weighted temperatures include:

Humans: $0.15\ T_{db} + 0.85\ T_{wb}$ (Provins, et al., 1962);

Dairy cattle: $0.35\ T_{db} + 0.65\ T_{wb}$ (Bianca, 1962);

Swine: $0.75\ T_{db} + 0.25\ T_{wb}$ (Roller and Goldman, 1962); and

Turkeys: $0.74\ T_{db} + 0.26\ T_{wb}$ (Xin, et al., 1992), where T_{db} is dry-bulb and T_{wb} is wet-bulb temperature. The weighting factors included in these expressions provide meaningful insight to the user concerning the relative importance of ambient temperature and humidity to animal comfort. However, the index expressed above for dairy cattle has not received much further attention or use. No reason for this lack of use with dairy cattle could be found in the literature.

Berry, et al. (1964) presented data on the decline in milk production (as given by Johnson, et al., 1962) based on least squares equations of dry- and wet-bulb temperatures as well as the temperature humidity index THI (per Thom, 1959). The results using THI, which gives equal weight to increasing dry- and wet-bulb temperatures as shown in Equation 3.3.1, were as accurate as equations considering them separately.

$$THI = 0.4(T_{db} + T_{wb}) + 15 \qquad (3.3.1)$$

where T_{db} and T_{wb} are dry- and wet-bulb temperatures, respectively, in degrees Fahrenheit.

Data for THI values are available through the U.S. National Weather Service. The prevailing equation for the temperature humidity index utilizes the air's dewpoint temperature T_{dp}, in degrees Celsius, as shown in Equation 3.3.2 (ASAE, 1991).

$$THI = T_{db} + 0.36\ T_{dp} + 41.2 \qquad (3.3.2)$$

Kibler (1964) confirmed that eight physiological responses of dairy cattle were associated with changing values of THI. Cargill and Stewart (1966) concluded that a THI value of 75 was the limiting design criterion for environmentally controlled summer dairy shelters. Appendix A shows the expected state of stress that has been associated with various THI values.

A primary advantage of THI is that it was accepted by the U.S. Weather Bureau as a comfort index for humans. Thus, it is widely monitored and is commonly used in other applications. Its has limitations in practice, however. It is neither a temperature nor a ratio, which may lead to conversion errors and effectively removes any physical meaning from its value. Also, there is considerable debate and confusion concerning its application in a varying environment.

Air movement. Convective heat and mass transfer from the animal's surface, described in a following chapter, depend on air movement to maintain thermal and vapor pressure gradients. Increased air velocity alters surface temperatures and the partitioning of heat losses from the animal, with convective and evaporative losses increasing and radiant losses decreasing. Effects of air velocity on the performance of swine (Bond, et al., 1965; and Riskowski and Bundy, 1990) and broilers (Drury and Siegel, 1966) have been reported.

Milk production is impacted by air velocity only in hot weather. Brody, et al. (1954a) found that milk production and feed consumption were unaffected by differing air velocities (0.5 to 9 mph) from 18° to 80 °F, but, at 95 °F, milk production and feed consumption were not suppressed as much with the higher velocities.

Air velocity also has differing impact on heat loss at high and low temperatures. Thompson, et al. (1954) and Kibler and Brody (1954a) found that increasing air velocity shifts the vaporization curve toward higher temperature and reduces rectal temperature, thereby extending the range of physiologically tolerable temperatures. Cooler hair and skin temperatures resulted from increased convective (sensible) cooling with higher air velocity at temperatures below 80 °F. Total heat loss remained fairly constant with increased air velocity at these intermediate air temperatures, however, as reduced vaporization offset the higher sensible heat losses. Finally, the effect of increasing velocity is non-linear as the greatest impact occurred after raising the air velocity from 0.5 to 5 mph.

Timmons and Hillman (1993) demonstrated that heat stress may be worsened in livestock by increasing air movement at extreme temperatures (above 35 °C for poultry). This has not been demonstrated in cattle, however, probably since cattle have greater natural vaporization potential from their body surface.

Currently, no suitable index incorporates the effect of air velocity on heat stress in dairy cattle. Integrating the sensible and latent heat content of air, Suggs (1966) proposed using enthalpy and air velocity to characterize the heat loss and physiological responses of animals to the environment. However, the approach has not been advanced with dairy cattle although the enthalpy-velocity relationships that were developed showed potential for use in describing heat stress conditions.

Radiant heat exchange. Radiant heat transfer, also described later, may either intensify or mitigate heat stress. Solar irradiation is a significant heat source during daylight hours. At night, net radiant heat flow is skyward. Infrared (terrestrial) heat transfer may be a source or a sink depending on the temperature of the surroundings.

Kibler and Brody (1954b) found that milk production, feed and water consumption, and body weight were all depressed when cows were exposed to increasing radiation intensities (5 to 180 Btu/ft^2/hr). The trend was consistent at all experimental air temperatures (45, 70 and 80 °F).

The effects of radiant heat exchange have been addressed in a number of ways. The mean radiant temperature, MRT, is a commonly utilized parameter for assessing the impact of radiant heat load on humans and animals. MRT is defined as the uniform surface temperature of an imaginary black enclosure in which a subject, also assumed to be a blackbody, exchanges the same radiant heat as in the actual environment. MRT is calculated using black globe temperature and knowledge of the subject and the environment for heat exchange.

Studies have shown that MRT, used in conjunction with ambient temperature and air speed, is quite useful for evaluating cold stress and in calculating effective environmental temperatures for swine in cool conditions (Gunnarson, et al., 1967 and Hoff, et al., 1993a). The effects of humidity, however, are not accounted for in such an analysis. Thus, the usefulness of MRT and subsequently calculated effective environmental temperature are limited during warm weather.

Efforts have been made to develop a sensor that includes the effect of temperature, humidity, air speed, and radiant heating or cooling. Johnson and Kirk (1981) report on the wet-bulb globe temperature (WBGT) and the Botsball sensor. Compared to other currently available sensors, such a sensor would probably provide the truest measure of the thermal environment. However, maintenance difficulties and large errors that would be expected with the use of a sensor that must remain moist, "black" and fast-responding in real environments have likely limited its use.

Buffington, et al. (1981), studied the use of the black globe humidity index BGHI to help assess the impact of radiant heating on heat stress in dairy cows. The BGHI is obtained by replacing the dry-bulb temperature in the equation for THI with black globe temperature. BGHI effectively accounted for the milk production difference (10.9 percent) between shaded and unshaded cows during a Florida summer when there was no measurable difference in air temperature, humidity or THI.

BGHI is a definite improvement over the basic THI because it does include the effects of air movement and the surroundings. However, it is limited in some of the same ways as THI, and, for modeling purposes, by a lack of readily available and applicable black globe temperature data.

Suggs (1967) attempted to extend the relationship of stress to enthalpy and air speed (described previously) to include the effects of radiant heat exchange with the surroundings. An equation utilizing the mean radiant temperature of the environment was found to be in harmony with experimental data. However, when rewritten to utilize enthalpy, the equation did not accurately describe conditions of heat stress, most likely due to uncertainties involved in describing the body surface.

Behavioral measures. Several behavior patterns are good qualitative indices of heat stress. Some, such as frequency of feeding or drinking, are quantifiable, but highly variable among animals (Regan and Mead, 1939; Thompson, et al., 1949). Others, such as body positioning, salivating profusely, lapping or slopping drinking water, increased urination and bunching together, are very dramatic, but are quite difficult to quantify or assess. Continued advancement of monitoring systems, such as infrared activity monitoring (Pederson, 1994), might aid in effectively using behavioral measures as indices of heat stress.

Behaviors can also be utilized to reduce the effects of heat stress. Harrison, et al. (1993) developed and evaluated a conductive zone body cooling apparatus for use in behavioral thermoregulation.

In general, behavioral measures of heat stress have similar limitations as physiological parameters. Primarily, they are difficult to monitor and provide limited causal information.

3.4 The Role of Modern Naturally Ventilated Dairy Barns

The livestock shelter serves to moderate the thermal environment of the housed animals and to improve the environment in other ways so that the animals remain healthy and productive. Additionally, barns should enable farm operators to effectively manage the animals and increase labor efficiency. Scott (1984) provides a comprehensive review of livestock buildings.

The distinction between so-called cold and warm housing facilities deserves some attention. The difference lies in the interior thermal environment desired during winter months. Warm housing requires tight construction, proper insulation and persistent control (Kammel, et al., 1982; MWPS, 1989; and Choiniére and Monroe, 1990) to maintain indoor temperature significantly above that outdoors while providing adequate ventilation for moisture control. Cold housing, on the other hand, relies on greater ventilation rates to remove moisture with only a minimal temperature difference (5 °C or less, Bickert and Stowell, 1990). The choice of housing can have serious implications regarding construction costs and management requirements.

Generally, a cool environment is most suitable for today's higher producing cows, and the benefits provided by warm housing for milk production are often quite limited (Brody, 1956). In fact, there is increasing evidence that dairy cattle are healthier and more productive, and that dairy producers are financially better off when naturally ventilated cold housing is employed (Drehmann, 1994). Warm housing entails the need for greater air exchange to accommodate the higher heat and moisture dissipation requirements of cattle in warm environments. Hindhede, et al. (1982) report on the production and health of dairy cattle in various housing systems.

Designing dairy facilities for hot weather is now recognized as having great importance for farms in temperate climates. Consider that the thermoneutral zone of dairy cows is significantly cooler than that of other livestock species. Also, as milk production increases, cows usually have greater need to dissipate metabolic heat.

Rigorous control over the local animal environment provided by naturally ventilated dairy facilities is difficult to achieve because the driving forces for ventilation vary with changes in the weather. Also, in a freestall barn, the cows have great liberty to move about and to seek out locations within the barn that are comfortable. During warm weather, however, control of ventilation is irrelevant since heat exchange is facilitated by maximal air flow rates and over-ventilation is not a concern.

A variety of resources are available to assist producers in the design and layout of such facilities (Barth, 1988; Martin and Bucklin, 1994; Bickert and Stowell, 1994; Graves and McFarland, 1995; Bickert, et al., 1995). Some primary functions stand out, however, as being important to the heat balance of the cows.

Shade. The roof of a livestock shelter provides shade while reducing the entrance of precipitation into the building. The benefits of reducing the solar heat load on dairy cows are considerable (Buffington, et al., 1983) in terms of lower body temperatures and environmental black globe temperatures, as well as higher feed intake, milk production, conception rate and calf birth weights compared to cows without shade. Other studies have highlighted similar benefits (Roman-Ponce, et al., 1977). Wiersma (1982) identified shade as the single most significant improvement to the cows' hot weather environment.

Design recommendations for shade structures are provided by Wiersma (1982) and Buffington, et al. (1983). Primary considerations include barn orientation, radiative properties of the shade material, covered area, height, ventilation of the shade surfaces and the use of insulation.

Air exchange. Ventilation is required to remove heat, moisture and other air contaminants from the animal space. The barn design must provide at least the minimum ventilation rate based on animal weight and the season. Bickert, et al. (1995) outline appropriate ventilation rates for dairy cattle.

Air exchange can be accommodated through utilization of wind and thermal buoyancy forces (ASHRAE, 1989). Both of these naturally occurring forces effectively contribute to year-round ventilation (Bodman, 1983; Bickert and Stowell, 1993) though wind-induced ventilation is of greater importance during warm weather. Bruce (1977) details the role and application of natural ventilation in livestock systems and suggests that the function of natural ventilation systems be given a high priority in the design phase of building construction. Holmes and Graves (1994) offer a fairly comprehensive summary of the design considerations that need to be evaluated during construction of naturally ventilated buildings for dairy cattle.

Wind-induced flow. Wind generates pressure on building surfaces and will create air flow through any wall or roof openings. Maximizing open area in sidewalls is widely recommended to increase summer air exchange and to improve convective and evaporative cooling (Bickert, 1988). Choiniére and Monroe (1990) also recommend that portions of the gable endwalls be opened during the summer months. Stowell and Bickert (1994) demonstrated benefits from providing these additional openings when winds approached the barn at poor angles (parallel to ridge).

Dairy facilities that utilize wind-induced ventilation maintain cooler interior temperatures and are better able to maintain high levels of milk production during hot weather (Failla, et al., 1987; Stowell and Bickert, 1992b). Wind-induced ventilation is primarily a function of wind speed, direction of the wind relative to the openings, the size and effectiveness of openings, and siting with respect to obstructions (Hellickson, 1983; Albright, 1990).

Some efforts have been made to mechanize and control opening size (Brockett and Albright, 1987; Choiniére, et al., 1990; van't Klooster, 1995). Because dairy cattle can tolerate wide fluctuations in ventilation rate, however, costly automatic controls or frequent manual adjustments are considered unnecessary for dairy housing.

Thermal buoyancy. Differences in inside and outside temperature create a chimney or stack effect that moves air out through openings in the roof of a barn bringing fresh air in at lower levels, usually through the eaves during the winter. Air exchange via the stack effect is especially valuable during cold weather (Bruce, 1975 and 1981) for removing moisture and other air contaminants without inducing drafts.

Ventilation from the stack effect depends on the size and location of openings and the difference in the temperature between the air inside and outside the barn. The temperature difference that is maintained is a function of the animal heat production, level of insulation and ventilation rate (Bruce, 1973). Foster and Down (1987) provide a review of the equations that have been developed to calculate ventilation rate via the stack effect.

Access to feed and water. It is important to provide reasonable access to feed and water within livestock facilities. Good design encourages livestock to maintain feed intake for high production.

Thompson, et al. (1949) stressed the importance of making cool water available to cattle in hot weather for body temperature regulation. Seif, et al. (1973) found that dairy cows retain body fluids for evaporative heat loss even when water intake is greatly restricted. Thus, lack of water will result in reduced milk production.

The thermal environment in feeding and resting areas is especially important for maintaining milk production during warm weather. Cows are encouraged to eat and drink in a cool, comfortable place. Also, cattle will avoid traveling more than minimal distances to feed and water in hot weather since, although quantitatively small, the energy expended in walking is an additional heat load (Blaxter, 1962). Design assistance is available in this area as well as in structural and ventilation areas (Bickert, et al., 1995).

Clean, dry and comfortable resting place. The dairy cow is usually provided a free stall area and bed where she can rest comfortably in a clean, dry environment. The cows' use of the stalls, cleanliness, health and productivity all are improved when cool, comfortable environmental conditions are provided in this area during warm weather. Generous air flow is essential to maintain air quality and to dry bedding and floor surfaces.

Considerable information in regard to the design and management needs of these facilities is available (Bickert and Ashley, 1991; McFarland and Gamroth, 1994; Bickert, et al., 1995; McFarland and Graves, 1995). Key design components are also contained in a standard for the design of free stalls (ASAE, 1991).

3.5 The Energy Balance Equations

The stresses on animals in hot environments are more difficult to quantify than those in cold conditions using an absolute index of heat stress (Webster, 1974). Rather than trying to derive some single measure to describe the thermal environment or the animal's response to the environment and then attempting to transpose that laboratory measure to fit real conditions of heat stress, Mount (1968) promotes an alternative -- estimate the transfer of heat via each of the four channels (radiation, convection, conduction and evaporation) separately. Then, knowing the magnitudes of the components and the balance thereof with heat generation and storage, a physiological assessment of the animal in its environment can be made with better understanding and confidence. Birkebak, et al. (1966) also present a strong case for modeling animal systems using basic heat loss equations and properties. Such an approach has the advantage of providing causal information resulting in greater confidence in application of results and conclusions.

An animal body, a livestock building or most any component of either can be represented as a control volume. A control volume is simply a spatially defined system. The energy balance of any body or system is calculated by applying the law of conservation of energy to its control volume. This principle states that the rate at which energy (both mechanical and thermal) enters or is generated within a control volume minus the rate at which energy is leaving must equal the rate at which energy is stored within the control volume (Incropera and DeWitt, 1985). The energy balance, in its most general form, is shown in Equation 3.5.1.

$$\dot{E}_{in} + \dot{E}_{g} - \dot{E}_{out} = \dot{E}_{st} \tag{3.5.1}$$

where: $\dot{E}_{in}$ = rate at which energy enters the control volume;

$\dot{E}_{g}$ = rate of energy generation within the control volume;

$\dot{E}_{out}$ = rate at which energy leaves the control volume; and

$\dot{E}_{st}$ = rate of energy storage within the control volume.

Within a control volume, a thermal energy (heat) balance can be calculated as well. The net flow of heat across the surfaces that form the boundaries of the control volume, $q_{in} - q_{out}$, and the heat generated within the control volume, q_g, are heat gains. Thus, the heat balance can be expressed as the sum of all heat gains equals heat storage.

3.6 Mechanisms of Heat Exchange

The transfer of heat across a surface occurs through four mechanisms: conduction, convection, radiant heat exchange and change of phase. The first three mechanisms are sensible heat transfer processes that require a temperature difference between the surface and a surrounding medium. Latent heat exchange occurs when there is a material change of phase. Latent heat transfer does not depend directly upon a difference in temperature and is sometimes called insensible heat transfer. The mechanisms of heat transfer have received considerable attention in their general application. The fundamental principles that follow are described in detail within Incropera and DeWitt (1985).

Conduction. Conduction is a process of molecular interaction. It is defined by Fourier's law given in Equation 3.6.1. The rate of heat transfer depends on a material's thermal conductivity, the cross section surface area normal to the flow of heat and the temperature gradient in the direction of heat flow.

$$q_{cond} = kA_s(dT/dx) \tag{3.6.1}$$

where: q_{cond} = sensible heat flow across surface (W);

k = thermal conductivity (W/m K);

A_s = contact surface area (m^2); and

dT/dx = temperature gradient at surface (K/m).

Conduction is an elementary process that plays an integral role in most heat transfer processes. Although heat flow by conduction may be stated as being negligible in many situations, conduction dominates heat transfer in convective boundary layers and is almost always the limiting heat transfer mechanism in some portion of real systems.

Convection. Convection involves the transfer of heat from a surface by a moving fluid. Newton's law of cooling, given in Equation 3.6.2, states that heat flow from a surface is proportional to the surface area as well as the difference in temperature between the surface and the free stream. The proportionality factor is called the convection heat transfer coefficient. This coefficient is a function of fluid properties (such as density, viscosity, thermal conductivity and specific heat), the surface geometry and flow conditions. The process involved in determining the heat transfer coefficient is often called the problem of convection.

$$q_{conv} = hA_s (T_s - T_a) \tag{3.6.2}$$

where: q_{conv} = convective heat tansfer rate (W);

h = convection heat transfer coefficient (W/m^2 K);

A_s = exposed surface area (m^2);

T_s = surface temperature (K); and

T_a = ambient or free stream air temperature (K).

Flow conditions and Reynolds number. Convective heat transfer is highly dependent on the presence and form of boundary layers. Both velocity and thermal boundary layers exist just above the interface of the surface and the fluid.

The velocity boundary layer forms due to viscous shear (Streeter and Wylie, 1979) with the fluid velocity varying from zero at the surface to free stream velocity at some distance away from it. The shape and extent of the velocity boundary layer depend on the relative magnitude of inertial forces in the free stream compared to viscous forces along the surface and on the unique characteristics of the body in terms of geometry and surface roughness.

When external forces drive the fluid motion, the transport phenomena occur by forced convection. The Reynolds number is the dimensionless parameter used to describe the relative importance of inertial and viscous forces in such flows. Equation 3.6.3 is used to evaluate the Reynolds number for objects in a free stream of ventilation air.

$$Re_L \equiv \rho U_\infty L / \mu = U_\infty L / \nu \tag{3.6.3}$$

where:
- Re_L = Reynolds number;
- ρ = air density (kg/m^3);
- U_∞ = free stream velocity (m/s);
- L = characteristic length (m);
- μ = dynamic viscosity of air ($N\ s/m^2$); and
- ν = kinematic viscosity (m^2/s).

In general, a flow regime with low Reynolds number is dominated by shear forces at the surface and is called laminar flow. The velocity boundary layer in laminar flow is parabolic. The depth and shape of the boundary layer depend on the surface geometry, fluid viscosity and position along the surface.

Turbulent flow results when inertial forces dominate and the Reynolds number is high. In turbulent flow, eddies (random rotational movements) that exist or develop in the flow overcome the viscous action near the surface. This effectively narrows the laminar boundary layer and increases the velocity gradient at the surface. A small buffer layer and comparatively broad turbulent boundary layer are supplanted between the laminar and free stream flow conditions.

Turbulent mixing action greatly enhances the transfer of heat from a surface. The effect of the viscous sublayer and other surface boundary conditions are minimized in highly turbulent flow to the point of being negligible. A description of heat transfer in systems with highly turbulent flow is beyond the scope of this research, but is presented in substantial detail by Leont'ev (1966).

Generally, turbulence is increased by raising the fluid velocity or through addition of disruptions on the surface. Hosni, et al. (1991) evaluated several rough-wall surfaces and reported that the heat transfer rate was 40 to 75% greater with roughened compared to smooth surfaces at equivalent Reynolds number.

Free stream turbulence, commonly referred to as turbulence intensity, impacts heat transfer in a manner similar to that of rough surface conditions. Kestin, et al. (1961) and Kestin (1966) illustrate through a series of studies that, as turbulence intensity increases, the heat transfer rate produced increases from otherwise equivalent Reynolds number flows.

Zukauskas (1972) presents a thorough presentation of factors affecting the rate of heat transfer from tubes in crossflow that considers Reynolds number, surface roughness, tube arrangement and turbulence intensity among other factors. An interesting finding of this research was that inner tubes within a bank of tubes commonly had higher heat transfer rates than leading edge tubes. This result was attributed to the effects of greater turbulence intensity of the flow as the fluid stream proceeded into the bank of tubes.

Nusselt number and convective heat transfer coefficients. A distinct thermal boundary layer also exists wherein the difference between the local fluid temperature and surface temperature, $T - T_s$, varies from zero at the solid surface to a maximal value of free stream temperature minus surface temperature, $T_\infty - T_s$, some distance away from the surface. The thermal boundary conditions and convective heat transfer rate are evaluated using similarity techniques and the Nusselt number, Nu. The Nusselt number relates the convection heat transfer coefficient to conduction across the thermal boundary layer as shown in Equation 3.6.4. Large Nusselt numbers are associated with high rates of convective heat transfer whereas a low Nusselt number means heat transfer is limited by the rate of conduction across the thermal boundary layer.

$$Nu_L \equiv hL/k \tag{3.6.4}$$

where: Nu_L = Nusselt number;

h = convective heat transfer coefficient (W/m^2 °C);

L = characteristic length (m); and

k = thermal conductivity of fluid (W/m °C).

Empirical relationships for the Nusselt number have been developed for a variety of configurations and flow conditions, usually as functions of the Reynolds and Prandtl numbers. The Prandtl number, Pr, is the ratio of the momentum and thermal diffusivities, ν and α, and is approximately constant at 0.7 for air from 20 to 40 °C. Baseline information can usually be obtained by using average Nusselt numbers that have been experimentally derived for a flat plate, smooth cylinder or sphere in cross-flow.

Flat plate: $\bar{Nu}_x = 0.664\ Re_x^{1/2}\ Pr^{1/3}$ for laminar flow, $Pr > 0.6$, or (3.6.5)

$\bar{Nu}_L = (0.337\ Re_L^{4/5} - 871)\ Pr^{1/3}$ for mixed flow,

$0.6<Pr<60$, $Re_L <10^8$. (3.6.6)

Cylinder: $\bar{Nu}_D = C\ Re_D^m\ Pr^{1/3}$ for $0.4<Re_D <4x10^5$, $Pr > 0.7$, where the (3.6.7)

constant m is experimentally derived, and is generally near 0.5.

Sphere: $\bar{Nu}_D = 2 + (0.4\ Re_D^{1/2} + 0.06\ Re_D^{2/3})\ Pr^{0.4}\ (\mu_\infty / \mu_s)$ (3.6.8)

for $3.5<Re_D<7.6x10^4$, $0.71<Pr<380$ and $1.0<(\mu_\infty / \mu_s)<3.1$.

Flow also occurs due to thermal buoyancy forces that exist in the presence of a vertical temperature gradient. In this case, the Nusselt number is a function of the Rayleigh number, which is a function of the Grashof and Prandtl numbers. Although free convection is almost always creating air flow around a body in real environments, its effect is negligible if $Gr_L / Re_L^2 << 1$. If this condition is not satisfied, free convection must be accounted for by applying a relationship similar to Equation 3.6.9. Morgan (1975) provides an extensive overview of heat transfer from smooth horizontal cylinders by both natural and forced convection.

$$\bar{Nu}_L = C\ Ra_L^n \tag{3.6.9}$$

wherein $Ra_L \equiv Gr_L\ Pr$ and:

Nu = Nusselt number;

Ra = Rayleigh number;

C & n = experimental constants; and

Pr = Prandtl number.

The Grashof number is defined in Equation 3.6.10.

$$Gr_L \equiv g\beta(T_s - T_\infty)\ L^3/\nu^2 \tag{3.6.10}$$

where the new terms are:

g = gravitational constant (m/s^2); and

β = volumetric thermal expansion coefficient (K^{-1}).

Radiation. Radiant heat exchange does not require any medium for heat flow. Radiation is a thermal property that is associated with a material's absolute surface temperature and the radiant energy equations require the use of absolute temperature (degrees K).

Blackbodies, solar and terrestrial radiation. Radiation is emitted by all surfaces in proportion to the fourth power of their temperature according to the Stefan-Boltzmann law, Equation 3.6.11, with a spectral distribution also determined by absolute temperature (per the Planck distribution). A black-body is an ideal surface that absorbs all incident radiation and diffusely emits the maximal amount of radiation possible for its temperature.

$$E_b = \sigma T_s^4 \quad (3.6.11)$$

where: E_b = emissive power (W/m^2);

σ = Stefan-Boltzmann constant (5.67×10^{-8} $W/m^2\ K^4$); and

T_s = absolute temperature of surface (K).

The magnitude and spectral range of solar irradiation (radiation incident on a surface) are unique in environmental systems. Solar radiation is comprised of very short wavelength energy (most in the 0.1 to 4 μm range) due to the sun's high surface temperature (effective black-body temperature of about 5800 K). As received by the earth's atmosphere, it is also very directional in nature. By comparison, radiation given off by the comparatively much cooler natural surroundings on earth is not as powerful and is comprised of longer wavelength (4 to 100 μm) energy. Such radiation is called infrared radiation. An object's radiant heat load may be affected by both long and short wavelength radiation. It will be affected differently by each, however, due to differences in their spectral and directional natures.

Emissivity and Kirchoff's law. An important consideration is that real objects and their surroundings are not ideal radiant surfaces. Emissivity is the ratio of radiation emitted by a real surface to that emitted by a blackbody at the same temperature. Emissivity may vary with both direction and wavelength. The fraction of incident radiation that is absorbed by a surface is called absorptivity. The remainder of incident radiation will be either reflected by or transmitted through the surface.

The radiant heat exchange of an object with its surroundings and of the surroundings with the atmosphere are dependant on the emissive and absorbing nature of the surfaces involved. For many real (grey) surfaces, absorptivity is approximately equal to emissivity according to Kirchoff's law, though Kirchoff's law only applies when the components of irradiation have similar wavelengths. Because of the large difference in spectral composition, solar and infrared irradiation may be absorbed and reflected differently by a surface. The simplified equation describing the radiant heat exchange for an object in a perfect enclosure is given in Equation 3.6.12.

$$q_{rad} = A_s \sigma(T_s^4 - T_{es}^4)/[1/\varepsilon_s + A_s/A_{es}(\varepsilon_{es} - 1)] \qquad (3.6.12)$$

where: q_{rad} = net radiant heat transfer rate (W);

A_s, A_{es} = surface area of object and enclosure, respectively (m^2);

T_s, T_{es} = object and enclosure surface temperature, respectively (K); and

ε_s, ε_{es} = emissivity of object and enclosure surface, respectively.

View factors and heat exchange with real surroundings. Calculation of the heat exchange between distinct surfaces utilizes the concept of the view factor. The view factor, F_{ij}, is a relationship that delineates how well surface i "sees" surface j as a fraction of unity (unity implying that all the radiation leaving surface i is intercepted by j). Sparrow and Cess (1978) describe the theory behind view factors in great detail. Values for a variety of such factors have been determined for common two- and three-dimensional surface arrangements (Howell, 1982).

Two useful relationships exist to aid in calculating view factors. The reciprocity rule, in Equation 3.6.13, allows one view factor to be determined from knowledge of the reciprocal view factor and surface areas. The summation rule states that the sum of view factors for any one surface to all of the remaining enclosure surfaces must be unity.

$$A_i F_{ij} = A_j F_{ji} \qquad (3.6.13)$$

For blackbody surfaces, the gross flow of radiant energy from surface i to j equals the product of the source surface area and black-body emissive power times the appropriate view factor. The net radiative heat transfer rate, q_{ij}, equals the difference in gross heat flows between surfaces which can be expressed as a difference in temperatures raised to the fourth power as shown in Equation 3.6.14.

$$q_{ij} = A_i F_{ij} \sigma(T_i^4 - T_j^4) \qquad (3.6.14)$$

This relationship assumes the object is fully enclosed within surroundings of constant surface temperature and emissivity. In real environments, this is seldom true. Radiant heat exchange is also complicated by the fact that real surfaces are not perfect emitters or absorbers. Radiosity is a term used to define the total energy leaving a surface. The radiosity of a surface equals the sum of radiation emitted plus radiant energy reflected by the surface as in Equation 3.6.15. The irradiation may have to be broken into components if it includes both short- and long-wave radiation because many materials have significant differences in reflectivity to radiation at differing wavelengths.

$$J_i = \varepsilon_i E_{bi} + (1 - \varepsilon_i) G_i \tag{3.6.15}$$

where: J_i = radiosity of surface (W/m^2);

ε_i = surface emissivity;

E_{bi} = black-body emissive power, σT_i^4 (W/m^2); and

G_i = surface irradiation (W/m^2).

The net total radiant heat flow from any surface within the enclosure, q_i, can be calculated through a network solution or by summation of component surface exchanges as shown in Equation 3.6.16 or 3.6.17 where the summations are over the n total surfaces of the enclosure. Radiation heat transfer textbooks (Siegel and Howell, 1972; Sparrow and Cess, 1978) contain descriptions of techniques for approaching enclosure problems.

$$q_i = \sum A_i F_{ij} (J_i - J_j) \tag{3.6.16}$$

$$q_i = \varepsilon_i A_i \left[\sigma T_i^4 - \sum (F_{in} J_n)\right] \tag{3.6.17}$$

Evaporation. Evaporation has great potential for cooling because of the large latent heat of vaporization at skin temperature (approximately 2400 kJ/kg). The evaporative heat transfer rate is the product of the latent heat of vaporization and the mass transfer rate as in Equation 3.6.18. Evaporative heat transfer occurs through both diffusion and convective evaporation of moisture from a surface. The nature of evaporative processes and equations describing them are conveniently presented in the ASHRAE (1989) Handbook of Fundamentals and the following relationships were obtained primarily from that reference.

$$q_{evap} = \dot{m}_w h_{fg} \tag{3.6.18}$$

where: q_{evap} = evaporative heat transfer rate (W);

$\dot{m}_w$ = mass transfer rate of moisture (kg/s); and

h_{fg} = latent heat of vaporization (J/kg).

Diffusion and Fick's law. Diffusion is a passive process that occurs in the presence of a vapor pressure gradient. Fick's Law for diffusion of mass is analogous to Fourier's Law for conduction of heat. The diffusion of water vapor from a free water surface into air is a function of surface area, the mass diffusivity of water vapor in air and the vapor pressure gradient at the surface as given in Equation 3.6.19. The mass diffusivity of water vapor into air is approximately 330 m^2/hr.

$$\dot{m}_w = A_s D_v d(\rho_w / d\rho)/dy \tag{3.6.19}$$

where: $\dot{m}_w$ = mass transfer rate (kg/s);

A_s = evaporative surface area (m^2);

D_v = mass diffusivity of water vapor into air (m^2/s);

ρ_w, ρ = density of water vapor and moist air, respectively (kg/m^3); and

y = length of gradient (m).

Convection of mass and analogies to convective heat transfer. The convection of mass is analogous to convective heat transfer with the substitution of a mass transfer coefficient and a vapor density differential between the saturated surface and the free stream as shown in Equation 3.6.20.

$$\dot{m}_w = \bar{h}_m A_s (\rho_s - \rho_\infty) \quad (3.6.20)$$

where: $\dot{m}_w$ = convective mass transfer rate (kg/s);

$\bar{h}_m$ = average convective mass transfer coefficient (m/s);

A_s = evaporative surface area (m^2);

ρ_s = saturation vapor density at surface (kg/m^3); and

ρ_∞ = free stream vapor density (kg/m^3).

For convective heat transfer, $\bar{Nu} = g(Re, Pr)$ and $\bar{Sh} = g(Re, Sc)$ for low mass flow rates and similar geometry, where: $\bar{Sh}$ is the average Sherwood number; Sc is the Schmidt number; and the function g is the same in each case. The form of the Sherwood and Schmidt number are given in Equations 3.6.21 and 3.6.22. Thus, if Nusselt relations have been developed for given flow and geometry conditions, the Sherwood number and resulting mass transfer coefficient can be found in a similar manner.

$$\bar{Sh} \equiv h_m L/D_v \quad (3.6.21)$$

$$Sc \equiv \mu/\rho D_v \quad (3.6.22)$$

where: $\bar{Sh}$ = average Sherwood number;

Sc = Schmidt number;

h_m = convective mass transfer coefficient (m/s);

L = characteristic length L (m);

D_v = mass diffusivity (m^2/s);

μ = dynamic viscosity of the fluid medium ($N\ s/m^2$); and

ρ = fluid density (kg/m^3).

The Lewis relation states that when the ratio of thermal to mass diffusivities, α/D_v, equals unity in laminar flow or any value in turbulent flow, the ratio of heat to mass transfer coefficients, h/h_m, equals the humid specific heat, c_{pm}. The humid specific heat is given by Equation 3.6.23. The result of these derivations is that a much simplified method of evaluating water vapor transfer is achieved for many real situations.

$$c_{pm} = (\rho_s/\rho_\infty)c_p \tag{3.6.23}$$

where: c_{pm} = humid specific heat (kJ/kg K);

ρ_s, ρ_∞ = vapor density at the surface and free stream (kg/m^3); and

c_p = specific heat capacity of the fluid (kJ/kg K).

The heat and mass transfer analogies defined above are only explicitly valid for low mass flow rates. A high mass flow rate unduly influences the velocity sublayer and subsequently affects heat transfer from the surface. When high mass flow rates exist, the system must be evaluated based on the blowing parameter and advanced turbulent flow theory (Leont'ev, 1966).

3.7 The Energy Balance Equation for Animals

Monteith (1974) presented an approach that is generally used when modeling animals and animal systems. In this approach, the surface heat flows are expressed as heat losses rather than gains; i.e., in terms of heat leaving minus heat entering the body, $q_{out} - q_{in}$. In this framework, heat gains from internal heat generation minus losses across the body surface must equal heat storage in the body.

The heat generated within the body by warm blooded animals is equal to the total energy produced by metabolism minus the rate of mechanical work. Heat generation in animals is called metabolic heat production. Monteith (1974) and Curtis

(1983) provide fairly comprehensive overviews for performing an energy balance on animals. Accordingly, since $q_g - q_{loss} = q_{st}$, $q_g = q_{meta}$ and $q_{loss} = q_{sens} + q_{lat}$,

$$q_{meta} - q_{sens} - q_{lat} = q_{st} \tag{3.7.1}$$

where: q_g = rate of heat generation within the body (W);

q_{loss} = net heat transfer rate from the body (W);

q_{st} = rate of heat storage within the body (W);

q_{meta} = rate of metabolic heat production (W);

q_{sens} = rate of sensible heat transfer from the body (W); and

q_{lat} = rate of latent heat transfer from the body (W).

The form of Equation 3.7.2 follows by substituting the individual mechanisms for sensible heat exchange, acknowledging evaporation as the prevailing means of latent heat exchange in a warm environment, and expressing heat storage in terms of the body's heat capacitance and a changing body temperature.

$$q_{meta} - q_{cond} - q_{conv} - q_{rad} - q_{evap} = Mc_p\, dT_{body}/dt \tag{3.7.2}$$

where: q_{cond} = conductive heat flow from the body (W);

q_{conv} = convective heat flow from the body (W);

q_{rad} = radiant heat flow from the body (W);

q_{evap} = evaporative heat flow from the body (W);

M = body mass (kg);

c_p = average specific heat of the body (J/kg K);

T_{body} = body temperature (K); and

t = time (s).

If an animal is in thermal equilibrium, there is no change in body temperature and no storage of heat. This is called the steady state condition. Under steady state conditions, the rate of metabolic heat production equals the rate of heat loss from the surfaces of the body.

3.8 Modeling Heat Transfer Mechanisms in Dairy Cattle

The distinctive avenues and mechanisms for heat transfer and body temperature regulation in animals need consideration to develop an accurate model of heat production and losses. In the following sections, research conducted on metabolic heat production and heat transfer across peripheral body tissue, from the skin, through and from the hair coat, and from the lungs are examined. Relationships that have been developed to model the individual mechanisms of heat transfer are presented and assessed where appropriate. Comprehensive reviews of the heat transfer mechanisms employed by man and animals are provided by Monteith and Mount (1974) and Rosenberg, et al. (1983).

Metabolic heat production. The production of heat during metabolism is very important in the cow's thermal energy balance. During metabolism, a fraction of the caloric energy intake of the consumed feed is converted to heat. The actual rate of metabolic heat production depends on the age, weight, level of production, current and previous levels of nutrition, sex, breed, health and environment (NRC, 1981) of the cows.

Maintenance needs. Energy is required to maintain essential bodily functions. An animal's metabolic energy requirement is generally proportional to it's surface area and consequently to a fractional power of its weight. This power is approximately equal to 0.7 for a wide range of animal species (Blaxter, 1962). In ruminants, such as cattle, roughly 4 to 12% of the total caloric energy intake for maintenance and growth is given off as sensible heat. The heat production rate (equalling heat loss at thermoneutrality) of fasting cattle is about 70-90 W/m^2 (Webster, 1974). This elementary heat loss is referred to as the basal metabolic rate. Maintenance needs usually account for about half of the total metabolic heat production in lactating cattle at typical production levels.

Pregnancy. The mature dairy cow requires minimal energy for growth, but additional energy needs are required for fetal development in pregnant animals. Energy requirements for fetal growth increase in a nonlinear manner as the pregnancy progresses.

Milk production. Production of milk requires significant quantities of metabolic energy. Lactation converts the metabolizable energy of food consumed in excess of maintenance needs to milk with an efficiency of about 70% (Blaxter, 1962). Blaxter also provided a formula for determining metabolizable energy needs that accounts for milk fat content. A high yielding cow producing 35 kg of 3.5% fat milk per day consumes 26,500 kcal (111 MJ) of energy to accommodate lactation. While only a fraction of this energy is given off as heat, the vastness of the gross energy expenditure for milk production in high producing cows results in lactation being a major contributor to the total amount of metabolic heat produced in these cows.

Under normal conditions, the energy efficiency of milk production, defined as the percentage of the total metabolizable energy intake that is used to produce milk, is about 20% (Kibler, et al., 1966) and has been measured to be as high as 40% in high producing cows (Brody and Cunningham, 1936). Obviously, high milk production, as is available today with improved breeding, feeding, management and facilities, places a substantially greater requirement on the cow to dissipate metabolic heat (Tyrrell, et al., 1988; Tillotson and Bickert, 1994).

Metabolic heat production models. Yeck and Stewart (1959) presented metabolic heat production data that were widely used in the past. More recently, other formulations have been developed to account for the perceived underestimation of heat production rates obtained from their data. Webster (1974) provides metabolic heat production rates for cattle in a thermoneutral environment at various production levels. Four recently developed equations for total and component metabolic heat production rates under thermoneutral conditions are available for comparison (CIGR, 1984).

$$\phi_{at} = 5.52\ m^{0.75} + 23.4\ Y \tag{3.8.1}$$

$$\phi_{at} = 6.36\ m^{0.73} + 22.2\ Y + 12.5\ e^{0.01p} \tag{3.8.2}$$

$$\phi_{at} = 6.6\ m^{0.73} + 21.7\ Y + 1.6\text{x}10^{-5}\ p^3 \tag{3.8.3}$$

$$\phi_{at} = 5.2\ m^{0.75} + 30\ Y + 1.6\text{x}10^{-5}\ p^3 \tag{3.8.4}$$

where: ϕ_{at} = total heat production, W;

m = body mass, kg;

Y = milk yield, kg/day; and

p = number of days pregnant.

The selection of an appropriate equation depends on the bias placed on each component and on how conservative the estimate must be. Of note, Equation 3.8.1 does not include a pregnancy term, Equations 3.8.2 and 3.8.3 yield nearly identical heat production values and Equation 3.8.4 provides estimates of heat production due to milk formation that are too high based on measured efficiencies of milk synthesis. The proportions of energy required for maintenance, pregnancy and milk production as partitioned by Equation 3.8.2 for a cow three months pregnant at various levels of milk production are illustrated in Figure 3.8.1.

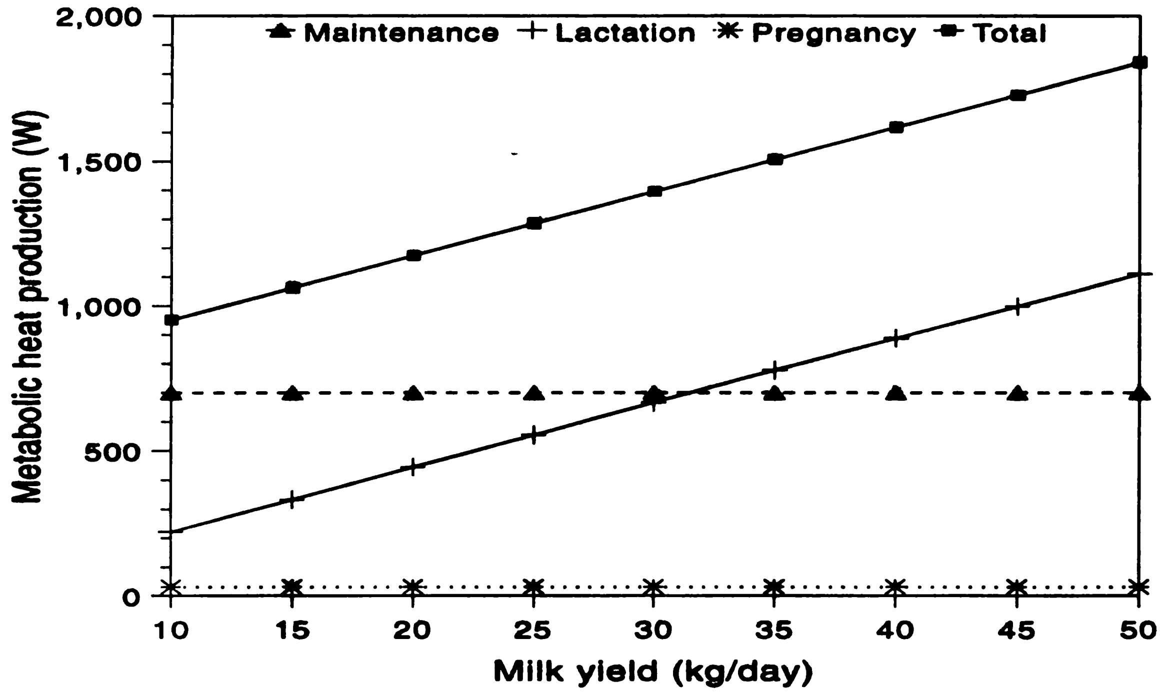


Figure 3.8.1 Effect of milk yield on modeled metabolic heat production of 625 kg cows that are 3 months pregnant (per CIGR, 1984).

<u>Response to environment.</u> Metabolic heat production is quite sensitive to the environment, decreasing rapidly in hot surroundings (Kibler and Brody, 1949; Kibler, et al. 1949; Yousef and Johnson, 1966). Cattle reduce feed intake to limit metabolic heat production in order to maintain thermal equilibrium in a hot environment with the

result that milk production also decreases. High humidity at high temperatures accelerates these declines (Johnson, et al. 1962).

Body surface area. Since all four of the mechanisms for heat transfer are surface phenomena, accurate calculations of the transfer of heat and mass to the surrounding environment requires a reasonable approximation of a cow's surface area. The Meeh equation, Equation 3.8.5, is generally used to estimate surface areas of livestock based on body mass or weight. The values of the constants, m and b, for dairy cattle are listed in Curtis (1983) as 0.15 and 0.56, respectively, for SI units.

$$A_s = mW^b \tag{3.8.5}$$

where: A_s = total surface area, m^2;

m = appropriate Meeh constant;

W = body weight, kg; and

b = specific exponential empirical constant.

For example, the surface area of a typical mature 450 kg Jersey cow is 4.6 m^2 according to the Meeh equation compared to 5.6 m^2 for a 650 kg Holstein. Surface area increases at a slower rate than does body mass during growth. McDowell, et al. (1953) specifically studied the effects of breed differences in surface area on the heat tolerance of dairy cattle. They determined that heat tolerance differences could not be attributed to differences in surface area since the vastly differing animal breeds studied had equivalent surface area to body weight ratios. They also evaluated correlations of surface area to body weight and determined that the correlations had correlation coefficients of between 0.70 and 0.75. The surface area of animals is also highly variable among individuals. Thus the Meeh equation is a best estimate that is used in the absence of direct measurements.

The surface area required for each component equation of heat transfer differs slightly depending on whether the skin, hair coat, or some surface in the hair coat is specified. A single value of surface area is often used when modeling dairy cattle in summer condition, however, due to the small thickness of the hair coat. Hillman and Gebremedhin (1995) measured the summer hair coat loft of the dairy cow as extending approximately 3 mm beyond the surface of the skin. The summer hair coats measured by Chastain and Turner (1994) were even thinner (1.4 mm, on average).

Animals will adjust the amount of surface area that is exposed to the environment in an effort to regulate heat flow from the body. Crouching and huddling behaviors are common in cold environments. In a hot environment animals generally will try to expose maximal body surface area to the air to increase the rates of convective heat and mass transfer. DeShazer, et al. (1970) found that sensible heat loss from laying hens increased by 20 to 40% when the birds changed from a sitting to standing position. If the radiant heat load on an animal is substantial, however, it may again crouch or attempt to crowd together with others to reduce the effective surface area exposed to the source of radiant energy. This reaction is often observed in animals exposed to direct sunlight.

Conduction across body surfaces. Conduction is a significant mechanism of overall heat transfer when there is direct bodily contact between the animal and solid surfaces having different temperature. In warm animal environments, heat loss by conduction is usually considered negligible. This assumption is acceptable for the conductive heat flow from a cow in a barn during the summer since i) the cow spends much of its time standing (to increase convective cooling as well as to perform normal activities), ii) air is a poor conductor, and iii) regardless of the cow's position, the temperature of potential cow contact surfaces (i.e. the stall bed, partitions, etc.) are not substantially different from the cow's surface temperature so little or no temperature gradient exists to drive conductive heat flow.

Convection of heat from the hair coat. Convection plays an important role in determining heat loss in cattle. Even if the magnitude of sensible heat loss is small during warm weather, the convective properties of the air stream also drive the evaporation of moisture from the animal's body which can produce substantially greater heat loss.

Before body heat can be lost to the air, it must be transferred from within the body to the surface of the hair coat. Several mechanisms of heat exchange are at work and a variety of models have been developed to describe the thermal phenomena involved in this process. Ingram and Mount (1975) provide insightful details on many of the mechanisms of heat exchange and highlight a number of the equations that follow.

<u>Heat transfer across peripheral body tissue.</u> All of the heat that flows from the outer body surface must first be conducted across the outer body tissues. The cow, through vasoconstriction and vasodilation, has some control over the blood supply and resulting insulation value of its peripheral tissues (Goodall and Yang, 1954). As the cow environment warms, vasodilation is engaged, sending more blood to and through the cutaneous blood vessels and the insulation value of the peripheral tissue is lowered

toward some minimal insulation level. Dowling (1964) determined that the skin thickness is not of major importance to heat flow from cattle. Henriques and Moritz (1947) evaluated the conduction through various tissue components, determining that muscle conducted heat at three to four times the rate of subcutaneous fat. The minimum tissue insulation of adult cattle is reported to be 0.05 $m^2\,°C$ hr/kcal (Curtis, 1983). An equation for determining tissue insulation from sensible heat flow from the skin is given by Blaxter (1962) in Equation 3.8.6.

$$I_t = (T_b - T_s)/M \qquad (3.8.6)$$

where: I_t = tissue insulation ($m^2\,°C/W$);

T_b, T_s = rectal and mean skin temperature, respectively (°C); and

M = unitized rate of metabolic heat production (W/m^2).

Cui and Barbenel (1990 & 1991) developed a numerical model of heat transfer and temperature distribution in the skin and surface tissue. This model was used to evaluate the effects of various parameters of the tissues on the skin when heat was removed from the body by convection and conduction. They found skin blood flow and dermal conductivity to be the main cutaneous parameters of importance to heat transfer.

Heat loss through hair coats. The thermal insulation provided by a hair coat is a primary mechanism for temperature regulation in most homeotherms. Turner and Schleger (1959) considered coat changes and conditions to be of greatest impact in the control of heat tolerance in cattle.

Cena and Clark (1978) review insulating properties of animal coats and human clothing. They present the insulating action of a hair coat as one part of a resistance circuit acting in series with parallel external resistances, namely surface convective and radiant resistances. Also, the hair coat provides resistance to both sensible and latent heat (i.e. moisture) flow. Equation 3.8.7 defines the heat balance at the skin surface.

$$M = \frac{(T_s - T_c)}{I} + \frac{h_{fg}(\rho_{vs} + \rho_{va})}{R_V} \tag{3.8.7}$$

where: M = net metabolic heat flux to be dissipated from the body (W/m^2);

T_s = temperature of the skin below the hair coat (°C);

T_c = temperature of the hair coat surface (°C);

h_{fg} = latent heat of vaporization (J/kg);

ρ_{vs}, ρ_{va} = vapor density at skin and hair coat surfaces, respectively (kg/m^3);

I = insulation, or thermal resistance, of the hair coat (m^2 °C/W); and

R_V = diffuse resistance of the hair coat (s/m).

The resistance values can be further refined by first principles into their basic components as shown in equations 3.8.8 and 3.8.9.

$$I = \ell_{hc}/k \tag{3.8.8}$$

$$R_v = \ell_{hc}/D_v \tag{3.8.9}$$

where: ℓ_{hc} = thickness of the hair coat (m);

k = effective thermal conductivity of the hair coat (W/m °C); and

D_v = mass diffusivity of water vapor into air (m^2/s).

The hair coat serves to trap air and the insulation value of the coat subsequently comes from this trapped air. Equation 3.8.10 (as presented by Webster, 1974) predicts the thermal insulation value of cattle coats as a function of coat depth using data from two distinct studies under relatively calm (air movement of 0.2 m/s) and cool conditions.

$$I_e = (118 + 132\, f - 16.4\, f^2)\ x10^{-3} \tag{3.8.10}$$

where: I_e = external insulation for sensible heat flow, °C m^2/W and

f = coat depth, cm.

Berry, et al. (1965) measured the thermal insulation of cattle coats and reported the average conductance to be 3.6 Btu/m^2 hr °F. Tylutki, et al. (1993) recently determined the thermal insulation of cattle having summer and winter hair coats in a variety of housing situations. The insulation values for both hair coat types were near 2.0 °C m^2 d/MJ for the experimental conditions.

The conductance or, conversely, insulation of the hair coat is a function of a number of coat characteristics. Berry, et al. (1965) found that conductance was linearly correlated with hair weight per unit surface area and was greatly influenced by hair orientation (the ratio of hair coat depth to hair length), but was not significantly affected by hair diameter or the number of hairs per unit surface area. Tregear (1965) confirmed that, in still air, there was no consistent relation between conductance and hair density.

The hair follicle density of dairy cattle is dependent on breed with the total number of hairs determined at birth. This fact establishes the basis behind measurements (Turner, et al., 1962) that suggest that hair follicle density is negatively correlated with body weight (and subsequently with surface area). The importance of hair density appears to be more critical in the ability of the cow to sweat (Carter and Dowling, 1954) than in providing thermal insulation.

When the rate of heat flow across the hair coat is experimentally measured, the apparent thermal conductivity (per Equation 3.8.8) is always greater than that of still air. Tregear (1966) calculated that the rates of heat conduction along the fibers of animal coats were trivial due to the small percentage of surface area occupied by hair (0.1 to 1% typically). Evidently, the effect of other heat transfer mechanisms, in addition to conduction, must be included in these values in order to obtain rates of heat transfer higher than could be obtained by conduction alone.

Radiative heat exchange is a significant means of heat transfer within the hair coat. Research by Cena and Monteith (1975a) showed that radiation transfer in hair coats was affected by the coat structure in a manner similar to that of commercially developed insulations composed of randomly oriented inorganic fibers. Upon determining so-called radiant conductivities, they concluded that the addition of radiant heat flow was sufficient to explain the high measured coat thermal conductivities. The remaining differences are attributed to free convection within the coat (Cena and Monteith, 1975b). If desired, an analysis of heat transfer via free convection could be performed using Nusselt and Grashof number relationships.

When only the total sensible heat flow through the hair coat is required, it is far more convenient to use a single parameter called the effective thermal conductivity. This value includes the effects of conductive, radiant and convective heat transfer within the hair coat. Chastain (1991) recently measured the dry effective thermal conductivity of the bovine as 0.039 W/m °C. He also determined that the effective thermal conductivity of a saturated hair coat was 0.23 W/m °C.

Effect of radiation wavelength. The effects of shortwave radiation on heat transfer in hair coats will vary from those of infrared due to, not only the difference in magnitude of the energy fluxes experienced, but primarily to the differences in albedo. Cena (1974) reviews the nature of solar insolation on animals. The emissivity of white cattle, black cattle and grass are approximately 0.50, 0.90 and 0.70, respectively, for the range of wavelengths in solar radiation compared to 0.95 for each with infrared radiation (Curtis, 1983). The impact of reflected radiation must also be considered to determine the fraction of reflected radiation that is redirected to the skin surface (an apparent transmissivity of the coat). If this fraction is measurable, the net heat loads on the body may be equivalent even though a light colored hair coat is not warmed as much by the radiation. If significant penetration of the hair coat by radiation occurs, the radiant properties of the skin will become more important (Stolwijk and Hardy, 1964).

Effect of wind. Hair coat insulation is decreased by the effects of wind, the extent depending on the specific type of coat and direction of air flow. Coat insulation was reduced by a factor of 2 to 3 when simulated winds were increased from still air to 12 m/s for a variety of fur samples tested by Moote (1955). Tregear (1965) obtained similar results and additionally determined that hair density significantly increased the depth of penetration by wind and, thus, the hair coat conductance in animal coats. However, cattle coats were found to be among the coats little affected by wind.

Effects of body size. Since heat loss is a function of exposed surface area, the animal's body size and shape are important. These relationships are illustrated in

Equations 3.8.11 and 3.8.12, per Cena and Clark (1978). Heat loss may be increased by a thicker hair coat in small animals and on the appendages of larger species since the effect of a larger Reynolds number may outweigh the benefits of added insulation.

$$I = (r/k)\ \ln(1 + \ell_{hc}/r) \text{ for cylinders} \tag{3.8.11}$$

$$I = (r/k)\ (\ell_{hc}/(\ell_{hc} + r)) \text{ for a sphere} \tag{3.8.12}$$

where: I = thermal insulation of hair coat (m^2 °C/W);

r = radius of animal body (m);

k = effective thermal conductivity of hair coat (W/m °C); and

ℓ_{hc} = thickness of hair coat (m).

Convective properties of the hair coat. Arkin, et al. (1991) compared the convective heat transfer properties of the dry hair coat of a cow to that of a smooth metal surface. Convective heat transfer coefficients with the hair coat were lower than those of the metal plate at low Reynolds number (velocities below 1.65 m/s), but were progressively larger than those with the metal plate at higher Reynolds number. The effect of the summer hair coat may be to induce early transition from laminar to turbulent heat transfer. This conclusion is in agreement with those developed from controlled studies of heat transfer from circular cylinders having varied degrees of mechanical roughness (Achenbach, 1977).

Chastain (1991) determined experimentally that the external flow velocities did not penetrate the summer hair coat of a cow and that the hair coat behaved in similar fashion to surface roughness (equivalent to 4 mm sand roughness). Early transition to supercritical flow occurred at Re_D = 96,000, which, for the experimental cow modeled as a cylinder, required a velocity past the animal of just over 2 m/s.

A limited number of equations exist for estimating convection from the body of a cow. The correlation developed by Wiersma and Nelson (1967), as shown in Equation 3.8.13, is most widely accepted currently for use in predicting the convective heat transfer from cattle in a forced air stream. The equation is valid for cows modeled as cylinders within a limited range of air velocities where the Reynolds number, convective heat transfer coefficients and total convective heat transfer rate are given by Equations 3.8.13, 3.8.14 and 3.8.15, respectively.

$$\bar{Nu} = 0.65\ Re_D^{0.53} \text{ for } 8 \times 10^3 < Re_D < 1.5 \times 10^5. \qquad (3.8.13)$$

$$Re_D = U_\infty D_T / \nu \qquad (3.8.14)$$

$$\bar{h} = \bar{Nu}\ (k/D_T) \qquad (3.8.15)$$

where:

$\bar{Nu}$ = average Nusselt number;

Re_D = Reynolds number for cylinder model;

U_∞ = free stream velocity (m/s);

D_T = trunk diameter (m);

ν = kinematic viscosity (m^2/s);

$\bar{h}$ = total convective heat transfer rate ($W/m^2\ K$); and

k = thermal conductivity of air (W/m K).

Convection from the hair coat can be found using a modified form of Equation 3.6.2.

$$q_{conv} = \bar{h} A_s (T_{cs} - T_a) \qquad (3.8.16)$$

where: T_{cs} = hair coat surface temperature (K).

Forced convection generally dominates free convection at air speeds greater than 0.2 m/s. Under calm conditions free convection dominates. A number of basic equations are available to estimate the free convective heat transfer coefficient from cylinders (Rosenberg, et al., 1983). One expression used for horizontal cylinders is shown in Equation 3.8.17 for cases where $1.5x10^{-4} < D^3(T_a - T_b) < 15$.

$$h_{fc} = 1.2\ [(T_a - T_b)/D]^{1/4} \tag{3.8.17}$$

where: h_{fc} = free convective heat transfer coefficient (W/m^2 °C);

T_a, T_b = air and body surface temperature, respectively (°C); and

D = diameter of cylinder (m).

Monteith (1974) recommends that convective heat flow from animals be performed in one of three ways depending on the relative magnitudes of the Reynolds and Grashof numbers as partitioned by $0.1\ Re^2 \leq Gr \leq 16\ Re^2$.

1) If Gr falls below the given range, calculate heat flow by forced convection.

2) If Gr is above this range, calculate heat flow by free convection.

3) If Gr lies within this range, calculate h_c for free and forced convection and then use the higher value to estimate heat flow.

Radiant heat exchange. In any radiant energy transfer model, appropriate view factors must be determined. Various methods have been proffered for animal modeling purposes. Hart (1958) proposed a graphical method called orthographical projection. This approach modeled objects within an enclosure, such as a cow in a barn, as cubical points. This method models spherical objects effectively, but requires that the projected orthogonal areas be added by planimetry.

Perry and Speck (1962) later developed geometric (view) factors for cows and surrounding interior surfaces. The cows were modeled as equivalent spheres for each surface. The effects of a neighboring cow were also evaluated. The radius of the spherical model varied with each surface since real cows are larger when viewed from the side. If appropriate view factors can be obtained for the modeled animal, the characteristics of the emitted radiation can be defined and the emissivities for the individual surfaces can be determined. Then the basic principles of radiant exchange in an enclosure can be used as described previously.

Evaporative heat loss. Evaporative heat loss from animals, as illustrated in Equation 3.8.18, occurs through three avenues: diffusion, convective evaporation of body surface moisture and respiratory evaporation.

$$q_{evap} = q_{diff} + q_{surf} + q_{resp} \qquad (3.8.18)$$

where: q_{evap} = total evaporative heat loss (W);

q_{diff} = heat loss via vaporization of diffused cutaneous vapor (W);

q_{surf} = heat loss via evaporation of body surface moisture (W); and

q_{resp} = heat loss via evaporation from the respiratory tract (W).

Moisture diffusion. Scott, et al. (1983) described the evaporative diffusion heat loss from animal skin by Equation 3.8.19 and determined that the loss of heat via diffusion in cattle is only important at low ambient temperatures. It is insignificant at higher temperatures compared to the other mechanisms of evaporation (Murray, 1966).

$$q_{diff} = M_m A_s L_t (P_{vs} - P_{va}) \qquad (3.8.19)$$

where: q_{diff} = heat loss via vaporization of diffused cutaneous vapor (W);

M_m = permeance coefficient of skin to water (s/m);

A_s = evaporative skin surface area (m^2);

L_t = latent heat capacity of moisture (J/kg);

P_{vs} = partial pressure of water vapor at skin temperature (Pa); and

P_{va} = partial pressure of water vapor at air temperature (Pa).

The latent heat capacity of water decreases with increasing surface temperature according to Equation 3.8.20.

$$L_t = h_{fg} - c_p(T_s - T_a) \qquad (3.8.20)$$

where: L_t = latent heat capacity of water (J/kg);

h_{fg} = latent heat of vaporization of water at T_s (J/kg);

c_p = specific heat of water vapor (J/kg °C); and

T_s, T_a = skin and air temperature (°C), respectively.

Moisture movement within the hair coat. Bulk movement of water vapor within hair coats depends on the maintenance of an overall vapor pressure gradient across the coat thickness, the actual location of evaporation and the diffusive nature of the coat.

Naturally occurring evaporation normally takes place from the skin since the conduction of water by hair fibers is usually low. Allen, et al. (1970) reported that the hair close to the skin of cattle rarely contained free water.

According to Cena and Clark (1978), the transfer of moisture through the hair coat by free convection is controlled by the gradient of virtual temperature rather than air temperature within the coat. At any location, virtual temperature is defined by Equation 3.8.21.

$$T_v = T(1 + 0.38\, P_v/P) \qquad (3.8.21)$$

where: T_v = virtual temperature of hair coat (K);

T = air temperature, in degrees K;

P_v = partial pressure of water vapor (Pa); and

P = atmospheric pressure (Pa).

The appropriate relationships for mass transfer are given in Equations 3.8.22 and 3.8.23. The Lewis number, Le, is the ratio of the molecular diffusivities for heat and water vapor in air. A value of Le = 0.89 is appropriate for the ambient conditions within living spaces (ASHRAE, 1989).

$$Nu = 1.66\ (T_{vs} - T_{va})^{0.7} \tag{3.8.22}$$

$$Sh = Nu\ Le^{0.7} \tag{3.8.23}$$

where: Nu = Nusselt number;

T_{vs} = virtual temperature of evaporative surface (K);

T_{va} = virtual temperature of air (K);

Sh = Sherwood number; and

Le = Lewis number.

The effectiveness of evaporation as a mode of heat dissipation is reduced when evaporation of sweat or other moisture occurs from surfaces away from the skin. The hair coat of cattle changes considerably over the seasons, becoming less dense, shorter and generally less restrictive, but coat insulation still impedes the desired transfer of heat and moisture by convection during warm weather. Man, having considerable amounts of bare skin, has an advantage during summer in that regard.

Sweating rates and limitations on surface mass transfer. The cow's physiological source of body surface moisture is the sweat gland. Hair and sweat gland numbers in cattle are given by Carter and Dowling (1954). Under natural conditions (i.e., no additional moisture applied), the mass transfer rate is limited to the maximal sweating rate. Cena and Clark (1978) state that latent heat transfer through hair coats will more often be controlled by the rate at which sweat is generated than by the diffusion resistance of the coat. Data are not presented, however, to explain their rationale.

Allen, et al. (1970) reported that the sweating rate of cattle varied with ambient temperature from 28 to 438 g/m^2 hr and that most evaporation of sweat in the hair coat occurs at the skin surface. However, lower maximal rates (150 to 330 g/m^2 hr) of moisture evaporation from the skin were measured in other studies (Kibler and Brody, 1950a; Allen, 1962 and McLean, 1963a). Table 3.8.1 summarizes measured maximal rates of vaporization from the respiratory tract and the outer body surface.

Table 3.8.1 Maximal vaporization rates of Holstein dairy cattle.

Source	Respiratory tract		Outer body surface	
	(g/m^2/hr)	(g/hr)	(g/m^2/hr)	(g/hr)
Kibler & Brody (1953)	85		160	
Kibler & Brody (1954b)		270		610
Kibler, et al. (1966)		305	110	
Ingram & Mount (1975)	40		230	

Regardless of the maximal sweating rate, the evaporative heat loss from the cow's skin without the extraneous addition of moisture is limited as shown in Equation 3.8.24. If water is added to the hair coat, the heat required for evaporation is no longer taken directly from the skin surface. Rather, it is taken from either a liquid interface above the skin (when large water droplets are sprinkled onto the coat) or the air surrounding the hair coat (when misting is utilized).

$$q_{surf} \leq s_{max} A_s h_{fg} \tag{3.8.24}$$

where: q_{surf} = evaporative heat loss from the body surface (W);

s_{max} = maximal sweating rate (kg/m^2 s);

A_s = body surface area (m^2); and

h_{fg} = latent heat of vaporization (kJ/kg).

Convection of surface moisture. Chastain (1991) presented a combined equation for mass transfer from the surface of a cow as shown in Equation 3.8.25. An effective mass diffusivity is required to account for the additional resistance to vapor diffusion of the porous, fibrous hair coat. Chastain also modeled evaporation from within a cow's hair coat as a moving boundary defined by the position of the evaporative interface.

$$\dot{m}_w = A_s(C_s - C_a)/(R_{m,d} + R_{m,c}) \tag{3.8.25}$$

where: $\dot{m}_w$ = rate of mass transfer of moisture (kg/s);

A_s = evaporative hair coat surface area (m^2);

C_s = vapor concentration at skin surface (kg/m^3);

C_a = vapor concentration of ambient air (kg/m^3);

$R_{m,d}$ = diffusive resistance to mass transfer (s/m); and

$R_{m,c}$ = convective resistance to mass transfer (s/m).

The value of the resistances are determined using the following equations:

$$R_{m,d} = \ell_{hc}/D_{eff} \tag{3.8.26}$$

$$R_{m,c} = 1/h_m \tag{3.8.27}$$

where: ℓ_{hc} = depth of the hair coat (m);

D_{eff} = effective mass diffusivity of water to air (m^2/s); and

h_m = convective mass transfer coefficient (m/s).

<u>Respiratory moisture loss.</u> As air is inspired it is usually warmed to the temperature of the respiratory tract and humidified to saturation while the tract lining is simultaneously cooled and dried slightly. Deep within the lungs, the air reaches saturation at body temperature. As it is exhaled, it cools slightly and returns some of the added moisture to the respiratory tract. The result is that both sensible and latent heat are lost during respiration. During warm conditions, latent heat loss is of greatest importance. An equation for respiratory heat loss by Scott, et al. (1983) is shown in Equation 3.8.28.

$$q_{resp} = \rho Q_r h_{fg} (w_r - w_a) \qquad (3.8.28)$$

where: q_{resp} = evaporation via respiration (W);

ρ = average air density (kg/m^3);

Q_r = respiratory ventilation rate (m^3/s);

h_{fg} = latent heat of vaporization (J/kg); and

w_r, w_a = humidity ratios of expired and ambient (inspired) air (kg/kg).

Respiratory ventilation rate depends on the respiration rate (breaths/min) and tidal volume (volume/breath). A variety of values has been obtained for the maximal rate of air movement through the lungs of cattle and for the resulting moisture and heat loss from respiration. Several of these maximal values are shown in Table 3.8.2 along with typical basal values that prevail in thermoneutral conditions.

McLean (1963b) found that respired air was 90 to 100% saturated. However, Stevens (1981) determined that the assumption of virtually saturated air was only valid if the temperature of the respired air was taken as being cooled below body temperature. He developed the following equations to model the respiration rate, tidal volume,

Table 3.8.2 Respiration rates (maximal:typical basal) of Holstein dairy cattle.

Source	Resp. rate (breaths/min.)	Vent. rate (L/min.)	Water loss ($g/m^2/hr$)	Heat loss ($Cal/m^2/hr$)
Kibler, et al. (1949)	150:25	---	---	---
Kibler & Brody (1950b)	150:25	260:110	---	---
Kibler & Brody (1953)	140:35	310:140	70:30	50:20
Kibler & Brody (1954b)	90:25	240:100	---	35:15
Kibler (1964)[a]	95:25	180:100	---	---
Kibler, et al. (1966)	100:35	190:90	(300:150 g/hr)	---
Ingram & Mount (1975)	170:--	---	41:--	---

[a] Ambient air temperature was limited to 90 °F, thus maximal values are understated.

and respiratory vapor loss from Holstein cows based on expired air being saturated at some calculated temperature. The correlation coefficient of the predicted data correlated with measured vapor loss data reported in the Missouri AES Environmental Physiology series of bulletins was 0.743. The individual equations had higher levels of accuracy.

$$\ln(RR) = 2.966 + 0.0218\, t_c + 0.00069\, t_c^2 \tag{3.8.29}$$

$$TV = 0.0189\, RR^{-0.463} \tag{3.8.30}$$

$$EXT = 17.0 + 0.3\, t_c + e^{(0.01611\, RH\, +\, 0.0387\, t_c)} \tag{3.8.31}$$

where: RR = respiration rate, breaths/min;

t_c = ambient temperature, °C;

TV = tidal volume, m^3/breath;

EXT = expired air temperature, °C; and

RH = relative humidity, %.

3.9 Implementation and Control of Thermoregulatory Activities

Animals clearly adjust to their environment. In this section, activities associated with acclimatization and acclimation are discussed. Then the mechanisms responsible for controlling temperature regulation activities are investigated. Lastly, methods for modeling the adaptive measures of cows are presented. Foundational material is available in a review by Bianca (1965).

Acclimation entails those focussed modifications that may be made in response to a single environmental variable (e.g. solar heat load) or a temporary stressful condition. Acclimatization involves more elaborate modifications of physical thermoregulatory mechanisms that occur over time to address the entire system of variables in the animal's environment (Folk, 1974). Changes in the weather and the season are complex processes that elicit acclimatization from animals.

Worstell and Brody (1953) reported that gradually rising dry-bulb temperatures induced suddenly accelerated respiration and vaporization rates in European breed dairy cows. Rectal temperature began to rise at about 21 °C followed by depression of feed intake, heat and milk production, and pulse rate. Kamal, et al. (1962) confirmed that reduced metabolic heat production is a primary component of acclimatization as manifested by lower feed intake, milk production and growth. Seasonal variation in production of Holstein cows in naturally ventilated barns was recently reported by Stowell and Bickert, 1992b).

Acclimatization. A number of physical and physiological changes are made by cows on a seasonal basis. Among the more important changes that may be made include converting the hair coat to its summer condition and tuning up the capabilities to sweat, elevate the flow of blood to the vascular system and decrease body tissue insulation. Sometimes seasonal changes in body temperature and feed intake for milk production may also be employed as preparation for hot conditions. Results of several studies (Bianca, 1959a and 1959b; Moran, 1970; DeShazer, et al., 1970) suggest that acclimatization to a hot environment reduces the severity of heat stress.

The cow modifies the character of its hair coat to more suitably meet the demands of the season. Webster, et al. (1970) demonstrated that cows having their winter hair coats were extremely heat stressed in an environment that was warm (20 °C), but the environment was only marginally stressful for acclimatized cattle.

Moran (1970) found that the body temperatures of two breeds of cattle were closely associated with the mean daily dry-bulb temperature. Body temperature fluctuated within a roughly 3 °F range on a seasonal basis.

Acclimation. Diurnal variations are important and have received attention in the study of heat stress (Drury, 1966; Feddes and DeShazer, 1988; Xin and DeShazer, 1991 and 1992). In dairy cows, Yeck (1955) found that average daily temperature could be used satisfactorily in place of a prescribed temperature in relationships that were previously developed to predict total heat dissipation from a constant temperature environment. Moisture dissipation rates, however, may be substantially underestimated for warm, diurnal conditions using this approach. Therefore, Yeck recommended that mean daily temperature, or the mean attenuated by the daily maximum or minimum temperature, should be used during the summer to assess the thermal stress in cows.

A companion study found that heat production, cardiorespiratory activities (Kibler and Brody, 1956) as well as moisture vaporization (Yeck and Kibler, 1956) responded to diurnal temperature patterns in a manner consistent with those for cool or warm constant temperatures. Respiration rate, rectal temperature and moisture vaporization lagged ambient temperature by one to two hours in their response. This lag in body temperature response has recently been observed in Holstein cows in a naturally ventilated free stall barn (Stowell and Bickert, 1992a).

The upper critical temperature signifies the condition above which the animal is unable to regulate heat loss without experiencing a rise in body temperature or initiating a reduction in heat production. Thermolability is the measure taken by some animals in which body temperature is allowed to rise by a small amount for a short duration (Webster, 1974). Benefits of thermolability to the cow are temporary heat storage within the body mass and increased heat loss from the body due to a greater temperature difference between the body mass and skin surface.

Most animals utilize increased respiratory activity as a way to adapt to hot conditions. Dairy cows have the ability to increase respiratory ventilation rate by a factor of about three. They sweat and pant at moderate levels, as contrasted with sweating, non-panting humans and non-sweating, panting animals like dogs and swine.

Acclimation activities are not activated without cost, as most actually increase the metabolic energy requirement for maintenance and some pose a hazard to the animal's health if maintained for extended periods (Bianca and Findlay, 1962). An increase in respiration rate raises the metabolism of the muscles that work to maintain panting in cattle by as much as 270% compared to basal muscle metabolism (Hales, 1976). The increase in respiratory muscle metabolism can be predicted using Equation 3.9.1.

$$\Delta q_{mr}''' = 0.649\ (Q_r/0.9)\ q_{mb}'''\ (\text{for } \bar{T}_c > 38.6) \tag{3.9.1}$$

where: $\Delta q_{mr}'''$ = muscle metabolism increase from increased respiration, W/m^3;

q_{mb}''' = basal muscle metabolism of normal respiration, W/m^3;

Q_r = respiratory ventilation rate, $m^3/hr/m^2$; and

$\bar{T}_c$ = average body core temperature, °C.

The rise in body temperature associated with thermolability necessarily increases the metabolic rate according to the van't Hoff effect which relates the rate of thermochemical reactions to the temperature at which the reaction occurs. Metabolism is directly affected as shown in Equation 3.9.2 (Blaxter, 1962).

$$H_p = H_p^* e^{k\Delta T} \tag{3.9.2}$$

where: H_p = heat production at elevated body temperature (W);

H_p^* = heat production at thermoneutral temperature (W);

k = van't Hoff coefficient (°C^{-1}); and

ΔT = increase in body temperature (°C).

Cows stand during hot weather to expose more body surface area for convective and evaporative cooling. This action requires an additional expenditure of energy, albeit the amount is comparatively very small (1 or 2% of total expenditure, Blaxter, 1962).

Origins and site of thermoregulatory control. Temperature regulation has been the subject of considerable research. A very brief review of the research is necessary to better define where and when the physiological temperature control mechanisms are activated. How the control takes place is of lessor importance for this research.

Reviews of thermoregulation (Curtis, 1983; Ingram and Mount, 1975) converge toward consensus that thermoregulation is under central, exterior and peripheral control. Central control occurs in the hypothalamus. This organ receives neural stimuli from other sites and monitors core temperature in a manner similar to a central room thermostat in a house. When body temperature deviates from the desired setpoint or outside stimuli are received, the hypothalamus activates appropriate mechanisms to restore homeothermy. The hypothalamus utilizes both the nervous and hormonal networks to bring about the desired reactions.

The spinal cord and possibly other internal body organs facilitate and accentuate the activity of the hypothalamus. It is believed that the spinal cord, called external because it is located outside the brain, primarily initiates, transmits and disperses thermoregulatory signals that are processed by the hypothalamus. However, when the hypothalamus is nonfunctional, thermoregulation of the body is still maintained at a reduced capacity. The spinal cord has been credited maintaining this activity.

Peripheral organs, such as the skin, tongue, lung surface tissue, etc., are primarily responsible for sensing local temperatures, sending signals to and responding to feedback from the hypothalamus. However, it has been demonstrated that some peripheral temperature control mechanisms are stimulated directly by peripheral control sites.

Modeling thermoregulatory control. In modeling heat flow in animals, reliable information is needed regarding the source of control and the primary stimuli for certain acclimatory responses. For example, the onset of a physiological activity and the rate at which it proceeds could be controlled by the hypothalamus, control sites in the skin or a combination of the two. Also, the stimuli may be absolute temperature, a change in temperature, the magnitude of a surface vapor pressure gradient, exposure to light or a combination of stimuli. Control of physiological activities is obviously a complex subject. Although considerable research has been performed in this area, the intricacies of the control of thermoregulatory actions are not yet well understood.

Feed and metabolizable energy intake. Feed consumption, which is a primary determinant of the release of energy during metabolism, is likely under both voluntary and involuntary control. Blaxter (1962) summarizes research that establishes a clear relationship between feed intake and function of the hypothalamus -- the thermoregulatory control center. Hypothalamic control means that feed intake is automatically (i.e., involuntarily) adjusted to meet the thermal energy demands of the body. This largely explains the fall in feed intake that inevitably results from persistently hot conditions. Considerably less is known about the nature of the stimuli and control strategy that initiate adjustment of feed intake.

Voluntary control of appetite, on the other hand, has more to do with feed quality and the physical-chemical processes in an animal's stomach. Blaxter also shows that the food and metabolizable energy intake rises when high quality feed is available, regardless of the species or stomach system. Feed quality can be defined by the proportion of metabolizable energy present in the food -- with high quality feed having a greater metabolizable energy content. In this case, feed intake is not a direct function of the animal's thermal environment or thermal energy status, but rather of some level of satiety.

Since the metabolic heat increment of a foodstuff is inversely related to its metabolizable energy content, high quality feed can supply an animal's energy needs with less release of heat to the body. Nutritional management strategies are beyond the scope of this research. However, it is clear that feeding strategies are another very important tool that can be used to limit heat stress in cattle during hot weather.

Sweating. Sweating has been shown to be under the control of peripheral thermoregulation sites as well as central control. In cattle, this is clearly evident as sweating is initiated at about 18 °C compared to 29 °C in man, even though no change in body temperature occurs until ambient temperature rises considerably (Kibler & Brody, 1950a).

Respiration. The control pattern and mechanisms of control are considered by Murray (1986). The response of many animals to hot conditions has been shown to include increased respiratory activity which benefits the animal by increasing the total heat loss from the lungs. The studies of respiratory activity in dairy cattle that are referenced in Table 3.8.2 generally attempted to use environmental parameters such as ambient air temperature as variables controlling respiration. The results obtained provide useful insights, but, because of the considerable scatter in the data, relationships that were developed for assessing control of respiration are not very accurate.

When ambient environmental conditions are utilized as the stimuli for a respiratory response, the inherent assumption is that control occurs within the lungs or at the surface of the animal's body. A few studies have specifically related respiratory parameters to body temperature. Bianca and Findlay (1962) presented the measured rise in respiration rate, respiratory volume and resulting ventilation rate of bull calves exposed to hot environmental conditions as a function of rectal temperature. The results showed that each of the measures was fairly accurately predicted by rectal temperature. Hales (1976) developed relationships to predict the respiratory ventilation rate per unit of surface area as a function of rectal temperature based on the results of the work of Bianca and Findlay.

$$Q_r'' = 0.90 \text{ (for } \bar{T}_c \leq 38.6) \tag{3.9.3}$$

$$Q_r'' = 0.90 + 1.322\ (\bar{T}_c - 38.6)^{0.42} \text{ (for } 38.6 \leq \bar{T}_c \leq 40.5) \tag{3.9.4}$$

$$Q_r'' = 2.64 + 0.75\ (\bar{T}_c > 40.5) \tag{3.9.5}$$

where: Q_r'' = respiratory ventilation rate per unit surface area, $m^3/hr/m^2$ and

$\bar{T}_c$ = average body core temperature, °C.

Roller and Goldman (1969) found that in swine, respiration rate was well correlated with rectal temperature and that the standard error of the relationship was virtually the same for differing numbers of animals tested; i.e., there was little variability among animals. This approach assumes that respiration is centrally controlled and that equivalent respiratory responses could be achieved by an unlimited combination of environmental conditions

3.10 Models of Animal Interactions with their Thermal Environment

Stevens (1982) promotes the concept of making models isomorphic, which means that the components of models should progressively be designed and integrated to more closely resemble the components of the physical and physiological systems being modeled. Such models have the advantages of i) having measurable inputs and outputs at each component modeling stage of the overall model, ii) allowing for independent development of the component models and iii) facilitating the swift improvement of the model by updating individual components of the model. Several varieties of models have been developed to describe how animals interact with their environment.

Production. Early models attempted to predict the growth rate and production of livestock in various thermal environments. Modeling work continues in this area as the need remains to describe the productive processes of ever higher producing livestock. Such models have been developed by Greninger, et al. (1982) for layers, Nienaber, et al. (1982) for swine, and Timmons and Gates (1988) for Tom turkeys.

Thermal load exerted by environment. Hahn, et al. (1961) used scale models of livestock housing to evaluate the relative importance of shelter components toward the resulting interior thermal environment. By using black globe thermometry and a radiant energy balance, the heat loads contributed by various enclosure surfaces were determined.

Other models predict interior thermal conditions of livestock housing (Diesch and Froehlich, 1988). A few have incorporated an index of thermal stress into the model, while others have predicted the heat deficit or surplus that exists for optimum production. Reece and Lott (1982) developed a computer model to predict fuel consumption needs of broiler houses based on chicken characteristics and stocking density, climatic conditions, and house design.

Oliveira and Esmay (1982) developed a model to predict temperature-humidity and black globe humidity indices within various cattle barn enclosures. This model evaluated the performance of barns based on the calculated indices as a function of solar radiation, shade material, walls, eave height, floor type, air velocity, inside and outside air temperatures and dewpoint temperature. The steps that were followed were:

1) determine shape factor of each part of the surroundings with respect to the sphere;
2) determine the radiosity of each part of the surroundings with respect to the sphere;
3) multiply the shape factors by the respective radiosities; and
4) add the parts to obtain the total radiant heat load.

Unique aspects of this model were that it i) accommodated a variety of building configurations and materials, ii) included radiant heat load through the sidewall ventilation openings and iii) modeled an animal as a sphere to eliminate questions concerning orientation with respect to the radiant heat load.

A limiting feature of the model was that no attempt was made to determine the net heat exchange for the animal or enclosure surfaces. All surface temperatures were estimated based on weather conditions and construction materials. When evaluated against real field measurements, the model overestimated the value of the heat stress indices. The critical calculation was determined to be that of the surface temperature of the roofing materials.

Further investigation suggests that the model underestimated the cooling capacity of ventilating air, both on the enclosure and animal surfaces. Also, the model gave little or no insights concerning the actual or potential evaporative cooling from the animal. For most livestock species, and dairy cattle in particular, this is of considerable importance.

Energy balance models. Several models have been developed to perform energy balances on animals. McArthur (1982) proposed a method of modeling the energy exchange of animals with their environment that has the advantage that knowledge of skin temperature is not required. Beckett (1965) modeled the internal heat generation and external heat loss of 150 lb pigs. Their model partitioned heat loss amongst several transfer mechanisms and did so with reasonable accuracy.

Resistance circuit analogies are commonly used to model heat transfer in animal systems (Jordan and Barwick, 1965). Hoff, et al. (1993b) modeled the radiative and convective interactions of the newborn piglet with its surroundings. The piglets were modeled as cylinders having an established core temperature. The resistance to heat flow from the core to the skin was defined as tissue resistance and included the convective resistance of the cardiovascular system and the standard resistance to conduction offered by the vascular tissue and skin. Then the radiant heat exchange of newborn piglets in various enclosures was calculated by solving the matrix of equations resulting from summation. Use of this model was limited to ambient temperatures below 30°C for the piglets, however, because it neglected the effects of evaporation. Such a model has little direct value for use with hot weather dairy housing where evaporation is very important.

Turner, et al. (1987a) modeled the dynamic heat transfer from cattle using finite element analysis. The model was quite intensively designed in that it analyzed the impact of differing components (muscle, fat, water, etc.) of the body individually. It included provisions for modeling thermoregulatory control, establishing relationships for predicting respiratory ventilation rate, the increase in muscle metabolism due to panting or shivering, sweating rate, vasoconstriction/vasodilation and feed intake. The model was found to correctly simulate the trends of core temperature response of beef cattle exposed to high temperature environments (Turner, et al., 1987b). However, considerable error was present in estimating the magnitude and phase of the response. The error in the predictions was attributed primarily to the fact that the model did not account for warming of the ingesta although it is not clear from the data how this feature would improve the correlation of the predicted and measured data.

Chastain (1991) developed a model to evaluate a direct evaporative cooling cycle in terms of the heat flux to or from the cow's body relative to the heat loss by evaporation. The model considers convection, evaporation and any radiant heat loads placed on the animal.

Ehrlemark and Sällvik (1996) modeled the heat and moisture dissipation from cattle based on thermal properties of the animals and the heat load of the surroundings. Their model, ANIBAL, predicts the heat loss of cows based on a relative measure, the thermal load index, they defined as shown in Equation 3.10.1.

$$TLI = 100(t_{amb} - LCT)/(t_{body} - LCT) \tag{3.10.1}$$

where: TLI = thermal load index;

t_{amb}, t_{body} = ambient and body temperature, respectively (C); and

LCT = lower critical temperature (C); and

ANIBAL was developed to account for deficiencies in existing heat loss prediction equations of high producing cattle, principally underestimation of evaporative losses of cows in warm environments. The model they produced provided satisfactory results in this regard compared to values derived from experiments. The model did not incorporate a housing type or varied environment, however, as is desired in this study.

3.11 Ventilation Rate in Naturally Ventilated Buildings

Air flow through naturally ventilated buildings can be induced by either wind or thermal buoyancy forces. Wind-induced ventilation is of primary interest for this study and is highlighted in greater detail in this section. Thermal buoyancy will be overwhelmed by the action of the free stream if free stream velocities are sufficiently high. The critical air velocity for modeling heat loss in animal shelters was determined to be 0.12 m/s (Hoff, et al., 1993a). Above this value, convective heat loss is governed by standard Reynolds number relations.

Wind effect. Air enters openings due to the presence of wind pressure. The dynamic pressure and resulting air flow rate through wall openings are given by ASHRAE (1989) in Equations 3.11.1 and 3.11.2, respectively.

$$\Delta P_v = C_p^{1/2} \rho v^2 \tag{3.11.1}$$

where: ΔP_v = wind pressure (Pa);

C_p = surface pressure coefficient;

ρ = air density (kg/m^3); and

v = wind speed (m/s).

$$Q_w = C_v A v \tag{3.11.2}$$

where: Q_w = air flow rate (m^3/s);

C_v = effectiveness coefficient; and

A = free area of inlet openings (m^2).

<u>Open area.</u> Bottcher, et al. (1986) studied a scale-model naturally ventilated building to evaluate the ventilation rate, pressure coefficients and opening resistance as open area in the sidewalls varied from 0 to 75%. They determined that a straight-line relationship existed between the mass flow rate through the structure and the wall open area as shown in Equation 3.11.3. Pressure drops measured across the barn and opening resistance decreased in a non-linear fashion with increased wall open area, but since they act against each other when determining flow rate, the ventilation rate rose linearly.

$$\frac{\dot{m}}{\rho V_{ref} LH} = -0.006 + 0.00334\,A \qquad (3.11.3)$$

where: $\dot{m}$ = mass flow rate of air through the building (kg/s);

ρ = mass density of air (kg/m^3);

V_{ref} = reference velocity (m/s);

L = building length (m);

H = building ridge height (m); and

A = percentage opening area.

Bottcher and Willits (1987) modeled the flow of ventilation air around and through a peaked roof building using numerical methods. The analysis was cumbersome for high Reynolds number flow, but, for low Reynolds numbers, the results matched well those given by algebraic solutions and ASHRAE recommendations.

<u>Opening effectiveness and barn orientation.</u> The effectiveness C_v of an opening is primarily a function of the opening's discharge coefficient, C_D, when the direction of flow is parallel to the open area's surface normal (i.e. flow is perpendicular to

opening). The average value of C_v was 0.82 in the experiments performed by Bottcher, et al. (1986). As the experiments were performed with air flowing directly at the sidewalls and other parameters were controlled, $C_v = C_D$. Discussion with Bottcher (1995) revealed that this value was unrealistic for practical applications since both the opening characteristics and flow conditions were more favorable than could be expected in the field. He recommended that a value of 0.5 to 0.6 be used (similar to that given by ASHRAE above).

When the flow approaches in a direction different from normal to the opening, net flow through the opening will be reduced. This effect has been incorporated into the opening effectiveness term in a number of ways. Fundamental principles of fluid flow specify that the reduction factor is sin θ, where θ is the angle the wind vector makes with respect to the planar opening surface. This would require that there be no flow across a barn for wind directions parallel to the axis of the barn (i.e., $\theta = 0°$). Measurements in real facilities, however, indicate that there is always some minimal level of flow across buildings with parallel winds due to leakage and other real construction imperfections.

Barrington, et al. (1994) recommended the use of Equation 3.11.4 which produces a range of C_v values that vary from 0.15 to 0.40 as wind direction changes from parallel to perpendicular to the sidewall openings. Ventilation rates for modeled building orientations calculated using this equation varied from 7.0 to 37.0 m^3/s on a seasonal basis when evaluated against weather data. Equation 3.11.4 was developed for buildings that used tilt panels to regulate the opening size. The panels increase the opening's resistance to air flow and therefore have a lower discharge coefficient than do clear openings.

$$C_v = 0.15 + 0.25 \; |\sin \theta| \qquad (3.11.4)$$

Zhang, et al. (1989), developed and validated a model for natural ventilation that included the combined action of both wind and thermal forces. The resulting model demonstrated that thermal buoyancy effects dominated when wind speed was below 0.5 m/s and that wind induced flow dominated when wind speed was greater than 3 m/s.

Stack effect. Thermal buoyancy is the driving force in the free convection or as its commonly referred to as the stack effect. Bruce (1982) developed an equation for determining airflow velocities due to the stack effect. Timmons, et al. (1984) present nomographs for determining ventilation rates from an open building ridge using descriptive geometric parameters for the ridge, the temperature difference between inside and outside air, and the absolute outside air temperature. Down, et al. (1990) verified the conditions under which the equations developed by Bruce were valid. Pearson (1993) presents an improved equation for use in calculating the ventilation rate achieved via the stack effect. His equation utilizes pressure drop coefficients that, if known for the given opening, provide more accurate predictions of air flow.

Air distribution. The interior air flow patterns during ventilation via the stack effect have been modeled recently. Timmons and Baughman modelled ventilation in a livestock structure using dimensional analysis. Tanguy and DuPuis (1986) utilized a finite-element model to assess distribution patterns in naturally ventilated buildings having an interior heat source.

Effect of natural ventilation design features. A number of studies have been performed in the laboratory and in the field to measure the impact of barn features on the quality of ventilation achieved. Stowell and Bickert (1994) presented results of monitoring the thermal environment of sixteen free stall barns. They found that the gross amount of open area provided per cow was the chief determinant of maintaining low inside air temperature. Barn features that directly influence this parameter are sidewall height, percent of sidewall open and the stocking density. Barn orientation and gable open area were found to be related in their impact on the thermal environment of the barn.

3.12 Radiant Heat Exchange of Buildings

The heat contributed to the environment within a building by solar radiation can be substantial. At night, heat loss to the sky may also greatly affect the energy balance of a facility. The effects and modeling requirements of each of these conditions are considered in this section.

Solar radiation. The effect of solar radiation on building heat load is dependent on the astronomical factors that determine the amount and nature of radiation that reaches the building, the orientation of building surfaces with respect to the sun's rays, and the radiative characteristics of the building surfaces.

<u>Astronomical factors.</u> Due to the earth's shape, axial tilt with respect to the sun and rotation about its axis, radiation is received from the sun at differing angles with respect to horizontal at differing locations on the earth. The resulting angle is commonly called the solar elevation angle. As the solar elevation decreases from

normal, the net radiant flux incident on the surface also decreases according to Lambert's Cosine Law (Rosenberg, et al., 1983 throughout subsection).

$$I = I_0 \sin \beta \tag{3.12.1}$$

where: I = flux density on unit horizontal surface (W/m^2);

I_0 = flux density on unit normal surface (W/m^2); and

β = solar elevation angle (°).

The solar elevation angle is a function of the time of day, location on the earth and season. It is calculated according to Equation 3.12.2.

$$\sin \beta = \cos(l) \cos(h) \cos(D) + \sin(l) \sin(D) \tag{3.12.2}$$

where: l = latitude;

h = hour angle; and

D = solar declination.

The solar declination is the sun's angular distance north or south of the equator and determined by Equation 3.12.3.

$$D = 23.5 \cos[2\pi(d - 172)/365] \tag{3.12.3}$$

wherein d = the day (integer) of the year.

The amount and nature of the radiation that reaches the earth's surface is also greatly affected by the conditions of the atmosphere. Clouds, water vapor and particulate matter absorb and scatter some of the direct beam solar radiation.

Trenberth (1992) illustrates that in the radiation balance of the earth, roughly half of the incoming solar radiation reaches the earth's surface (global solar radiation) and another quarter is absorbed by the atmosphere. Of the solar radiation reaching the earth, some is received directly and the remainder is scattered by the components of the atmosphere. Scattered radiation is called diffuse because it reaches the earth at nearly all hemispherical angles. On a clear day, about 10-30% of the solar radiation reaching the earth is diffuse scattered radiation, whereas, on overcast days, virtually all of the incoming solar radiation is scattered. Clouds and the gases that comprise the atmosphere selectively absorb solar radiation at the very short (UV and shorter) and longer infrared wavelengths (Peixoto and Oort, 1992), meaning most of the remaining incoming solar radiation is in the visible and infrared range.

Details on radiation energy balances, the mechanisms of atmospheric scattering and absorption as well as equations for use in modeling radiation incident to the earth's surface are provided by Rosenberg, et al. (1983), Peixoto and Oort (1992) and Trenberth (1992).

Longwave atmospheric radiation. The atmosphere is always emitting radiation in accordance with its temperature. Clouds, and the atmosphere in general, tend to insulate the earth. In addition to intercepting solar radiation during the day, they absorb nearly 90% of the terrestrial radiation emitted by the earth at all times (Rosenberg, et al., 1983). The absorbed energy is stored in latent form and then is emitted as longwave radiation, with much of this re-emitted energy being directed back toward the earth. The total radiation flux (direct and diffuse solar plus longwave atmospheric) incident on the earth is called total hemispherical radiation.

Rosenberg submits an equation developed by Swinbank (1963) to predict longwave atmospheric radiation from the air temperature measured in a standard shelter 2 m aboveground. The accuracy of Equation 3.12.4 has been confirmed to be reasonable for modeling night-time radiation and dry atmospheric conditions, but greater errors are involved when used during the day or in humid conditions.

$$q_{lw}'' = 5.31 \times 10^{-13}\ T^6 \qquad (3.12.4)$$

where: q_{lw}'' = longwave sky radiation flux, W/m^2 and

T = shelter air temperature, K.

The earth's surface, including buildings, are at temperatures on the order of the atmosphere's. Thus, terrestrial radiation is emitted in the longwave range and Equation 3.12.5 relates the net flux of thermal longwave energy between the sky and a surface.

$$q_{lwnet} = q_{lw} - \sigma T_s^4 \qquad (3.12.5)$$

where: q_{lwnet} = net longwave radiation flux (W/m^2);

σ = Stefan-Boltzman constant (5.67×10^{-8} W/m^2 K^4); and

T_s = absolute surface temperature (K).

Bond, et al. (1967) demonstrated a means of measuring the various components of radiation that are incident on buildings. Representative zenith radiation measurements during summer California conditions (8/18/64, clear sky, noon) indicated the diffuse solar component was about 50 Btu/hr/ft^2 compared to 125 for the longwave atmospheric. This ratio would likely change considerably in more humid regions where the diffuse solar component can be much larger.

Effectiveness of shading devices. Provision of shade is one of the primary roles of the livestock barn. The effectiveness and benefits offered to cattle by various shading materials and arrangements may vary widely. Neubauer and Cramer (1965) performed tests of shading materials arranged in several orientations. They determined that the slope and orientation greatly influenced the all-day (average) surface temperature. Steeper slopes and exposure to the north sky resulted in the lowest daytime heat gain. A slope of 70° from horizontal experienced the smallest temperature rise, being an order of magnitude less than that of the flat arrangement. At night, the shades with less slope did lose more thermal energy to the night sky, however. These findings were incorporated into suggested roof and shade configurations.

Shade height. Givens (1965) determined that low shades (6 ft) were preferable to taller (9 and 12 ft) shades in the southeastern United States based on black-globe temperature measurements. [Black-globe thermometers have a temperature sensor enclosed within a black, conductive sphere. Black-globe temperature is a measure of the maximal radiant heating that may occur in a given environment.] He attributed the added heating effect in the tall shades to the cloudy conditions that prevail in that area. Cloud cover increases the proportion of radiation that is diffusely received by the earth and the tall shades evidently permitted more of this radiation to enter the building.

Bond, et al. (1967) monitored the radiation load under galvanized steel shades of similar heights as those of Givens' study, but in clear sky, dry California conditions. The measurements from the center of the shade indicated that the diffuse shortwave energy from the zenith sky at noon was 35 Btu/hr/ft^2 under a 12 ft shade compared to 15.6 under a 6 ft shade. It was evident from measurements and prediction that animals under a tall shade are exposed to greater amounts of diffuse solar energy. Substantial amounts of terrestrial longwave emission entered the buildings since more longwave radiation was consistently measured under the 12 ft shade when the longwave emission from the taller shade should have been less than that from a shorter shade. They also determined that 20-25% of the total radiation received by a surface under shade is diffuse solar radiation and half of that is reflected by the ground. They noted that the benefit of the taller shade should increase if animals were able to follow the shadow throughout the day.

Garrett, et al. (1967) tested this theory using the same shade facilities and beef cattle. The cattle were found to indeed be able to take advantage of the larger shadow cast by taller shades if they moved with the shadow. The advantage is most obvious under ambient conditions that otherwise imposed mild to medium heat stress on the animals. Rectal temperature, surface temperature and respiration rate were each significantly reduced under the taller shade compared to no shade and a 6 ft shade height. Under more severe conditions, the differences in the physiological responses were more variable and were not statistically significant.

Roof materials. Shade materials are generally characterized by their reflective and transmissive properties. Bond, et al. (1969) investigated the effect of radiant heating on three building materials -- unpainted plywood, white painted plywood and embossed aluminum. They concluded that a building's material surface does present a significant source of radiant heating to nearby animals on a sunny day. The more reflective aluminum and white painted surfaces reduced the resulting heating load.

While nonconductors, such as white enamel paint, are highly reflective to radiation in the shorter visible wavelengths, they are known to be virtually black to infrared radiation (Sparrow and Cess, 1978). Also, aging of roofing surfaces can reduce their reflectance through oxidation or simply by being contaminated. Birkebak, et al. (1964) reported that surface roughness decreases both the total hemispherical and specular reflectance of metallic surfaces, although not equally. When metals oxidize, their surfaces generally become pitted and rough.

Various coatings have been applied to roofs in an attempt to add or restore surface reflectance to the roofing. There is evidence that some of these coatings do reduce the solar heat gain to a livestock building (Bottcher, et al., 1990), although their longevity and effectiveness in naturally ventilated buildings with large sidewall openings is still unknown.

Dolby and Jeppsson (1993) recently reported on the thermal environment inside naturally ventilated dairy facilities having transparent roof covering. They determined that the interior thermal environment was well maintained as long as the ventilation rate was reasonably high. Considerable variation in the daily environment was evident in their results although it was not directly attributed to the effects of radiant heating and cooling.

Insulation is frequently installed below the roof sheeting. During hot weather, insulation under the roof is expected to reduce radiant heating of the roof on housed animals. The effect has been questioned for barns with tall open sidewalls, however. There appears to be little value of having reflective insulation for reducing radiant heat exchange in livestock buildings because it soon becomes covered with dust (Bottcher, et al., 1990). Generally, many problems must be addressed when installing insulation in naturally ventilated livestock buildings if it is to perform as desired (Muehling, 1967).

Convective cooling. The effective roof surface temperatures are a function of both the radiative and convective thermal environment. Movement of air over the roofing material tends to counteract the effects of radiant heating or cooling, thus limiting the difference between the temperature of the roof and ambient air.

When convective heat transfer from a roof is modeled as that from an inclined surface under laminar flow, equations developed by King and Reible (1991) for calculating Nusselt numbers may be applied. It is difficult to apply these equations to buildings, however, since real flows are generally turbulent and because total building geometries don't match that of a flat plate in a freestream.

Braud and Nelson (1962) performed model studies of the convective cooling effect of metal roofs that were exposed to both radiant heating and convective heat transfer (generally cooling). Their study examined thin, uninsulated roofs over symmetrical gable roof shelters with the wind direction normal to the eaves. Dimensional analysis was utilized to develop an equation relating the convective heat transfer rate to the incident radiant energy intensity as a function of ambient temperatures, the Reynolds number and position on the roof. After validating their model with field data, they concluded that shelters having a high roof with low absorptivity had a thermal advantage due to greater exposure to stronger winds and less solar heating.

More recently, Pieters et al. (1994) developed a static one-dimensional model describing heat transfer by conduction, convection, radiation, and phase change to, through and from single greenhouse covers. This model was utilized to determine the influence of condensation and evaporation on static heat losses from greenhouses.

4 - MODEL DEVELOPMENT

The model of heat transfer for the animals, in this case dairy cattle, was created by first developing a conceptual model of the various thermal interactions between the animal and the environment within the barn. This conceptual model considered the pertinent effects of weather conditions on the barn environment.

A few basic assumptions and constraints were established before developing the conceptual model. Next, a physical representation of the animals was selected. Then the production and dissipation of heat and moisture from their bodies were characterized. Theoretical and empirical relationships developed previously by others to describe heat flow from cows were utilized extensively. The combined actions of the various heat transfer mechanisms operating simultaneously were then modeled.

With the animals' heat loss mechanisms modeled, defining and modeling the animals' surroundings and general thermal environment remained. Weather was not modeled. Instead, real weather data were utilized as inputs. The weather inputs were included in the estimation of ventilation air characteristics and radiant heating impacts.

Following the development of the conceptual model, the appropriate theoretical and empirical relationships were incorporated into a computer program that quickly calculates the modeled cows' thermal energy status. The computer model was named ANTRAN (from "ANimal heat TRANsfer model").

4.1 Basic Assumptions of the Conceptual Model

Primary model assumptions were, i) mass and energy flows are two-dimensional, ii) air flows through the barn due only to wind forces, iii) air enters and exits the enclosure through the sidewall openings and iv) steady-state conditions prevail.

Two-dimensional model. A two-dimensional constraint was applied to both the animal model and the housing model. Figure 4.1.1 illustrates the two-dimensional nature of the modeled physical system. Livestock buildings are commonly modeled in this manner, converting the three-dimensional directional nature of wind to solely transverse flow of air across the width of the building. This assumed condition accurately represents buildings that are much longer than they are wide (as might exist in barns housing large numbers of cattle) as well as buildings that are situated with sidewalls facing the wind. Less accurate model results are obtained for barns of shorter relative length under real, varying weather conditions.

The cows are modeled as horizontal cylinders oriented perpendicular to the flow of ventilation air across the barn envelope. Specific implications of the cylindrical model assumption for dairy cattle are detailed in Section 4.2. The two-dimensional model assumption implies that the net flow of heat in the axial direction (along the length of the barn and cow rows) is zero.

Wind-induced flow. The assumption that the flow of ventilation air due to thermal buoyancy forces can be neglected is valid only if the Grashof number is very small compared to the Reynolds number (see discussion following Equation 3.6.9). Under relatively calm conditions, this assumption will not hold true and a different flow regime must be applied. That analysis is left for further study.

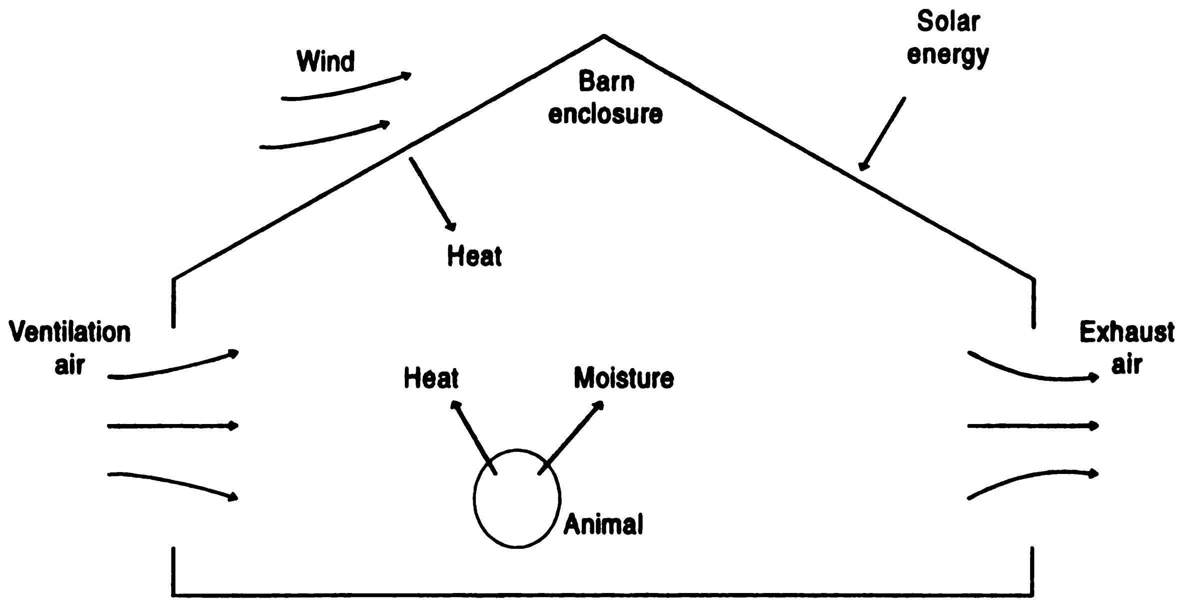


Figure 4.1.1 The two-dimensional nature of the animals and the naturally ventilated barn as modeled in ANTRAN.

Exclusive airflow through sidewalls. Real barns often have openings in the ridge of the roof and in endwalls. Flow through the endwalls has already been precluded from consideration in ANTRAN by the two-dimensional constraint. When warm season ventilation is due to wind forces, a ridge opening usually serves as an exhaust vent. The roof was assumed to be of gable construction (having uniform slope in both directions joining at the peak) and closed at the peak. The closed peak assumption was made to eliminate the consideration of numerous available ridge designs and because flow through the peak under conditions governed by wind was assumed to have minimal effect on the airflow in the vicinity of the animals. The presence of a ridge opening and the design of the opening would need to be considered for a more detailed evaluation of roof temperature or of calm weather conditions.

Steady-state conditions. The steady-state assumption eliminates the need to account for heat storage or to have specific knowledge of the time-varying nature of the animals' physiological reactions to the environment. A dynamic model would have to consider both the heat storage capacity of the building materials and the capacity of the animals to retain heat.

The steady-state model assumption neglects storage of heat in the animal's body when an increase in body core temperature occurs. Since any rise in body temperature will typically be slight and the objectives of this study are concerned with heat exchange potential once body and surface temperatures have essentially equilibrated rather than during any rise in body temperature, calculation of heat storage within the animal was considered to be of minimal importance and was not performed.

The model also assumes that all building component surfaces, except those of the roof, are at ambient air temperature unless explicitly assigned a different temperature. The roof is usually under additional radiant heat load and ANTRAN handles the roof separately. Since air temperature may change more rapidly than the temperature of other building components that have substantial thermal mass; e.g., the floor/ground system, building component temperatures are not precisely represented by the model. However, only the surface temperatures are necessary for calculating heat exchange with the animals and it was assumed that the surface temperature of these components would rapidly approach inside air temperature.

Little data are available on the thermal condition of floor surfaces in naturally ventilated buildings. Dolby and Jeppsson (1993) did find that in well ventilated buildings having transparent coverings, the daily average temperature of the upper region of the alley floors closely followed average ambient outdoor air temperature regardless of the season. In this study, it was assumed that floor surface temperature would be roughly equal to inside air temperature for all conditions. The effect of varying surface temperature or of the effect of moisture evaporation from the cow alleys were left for later consideration.

4.2 Representation of the Animals within the Barn

The cows were modeled as horizontal cylinders that are standing and are aligned end-to-end in rows extending continuously along the length of the barn. All animals were assumed to be homogeneous in size, shape and dimension. Figure 4.2.1 illustrates the positioning of a row of animals within the modeled barn enclosure.

Cylindrical form. The cows were modeled as cylinders for a number of reasons. First, modeling the animals as horizontal cylinders fits the two-dimensional nature of ANTRAN very well. Other modeled shapes, such as spheres or ellipsoids, possess a depth dimension that complicates modeling the animals. Also, most of the heat transfer relationships that have been developed and accepted for use with cattle, as well as other livestock, utilized a cylindrical animal form. Lastly, the cow is simply well-described physically by a cylinder, as are many other animals.

One disadvantage of the cylindrical form in a two-dimensional model is that the orientation of the animals must be pre-established. Another is that the distinct effects of appendages, such as the legs, are either ignored or muddled into the combined effect of an extended trunk. Although the effects of appendages and their substantial surface area play a role in heat transfer from the animal body, Wiersma

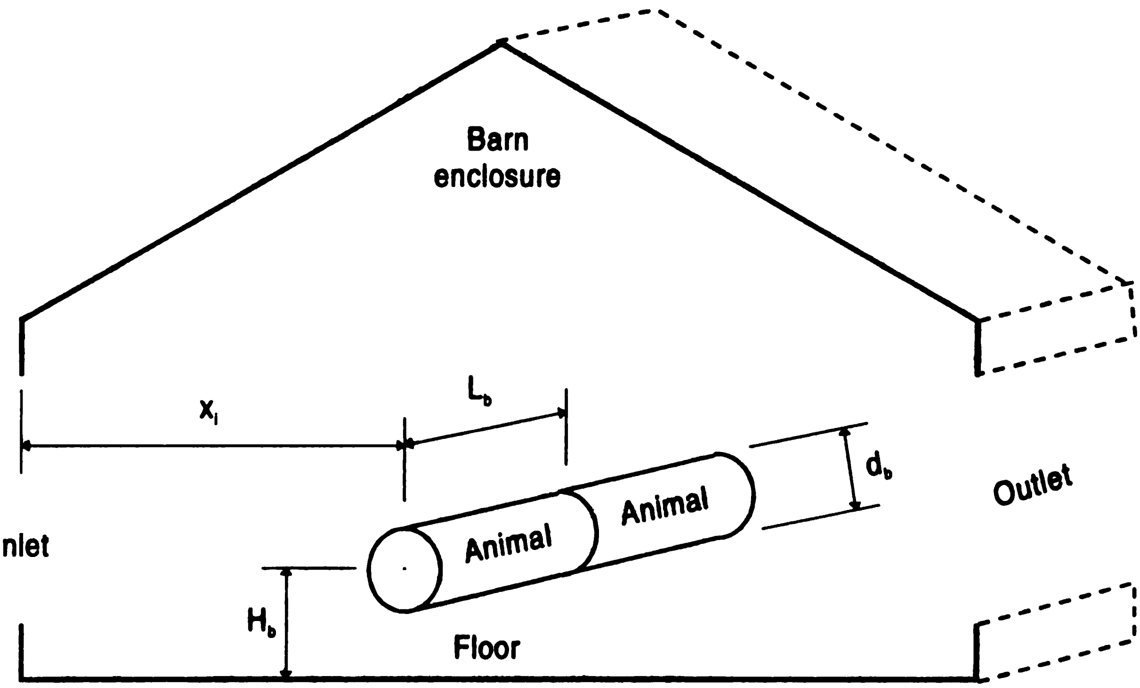


Figure 4.2.1 Illustration of the modeled cow dimensions and the location of the row of cows within the barn enclosure.

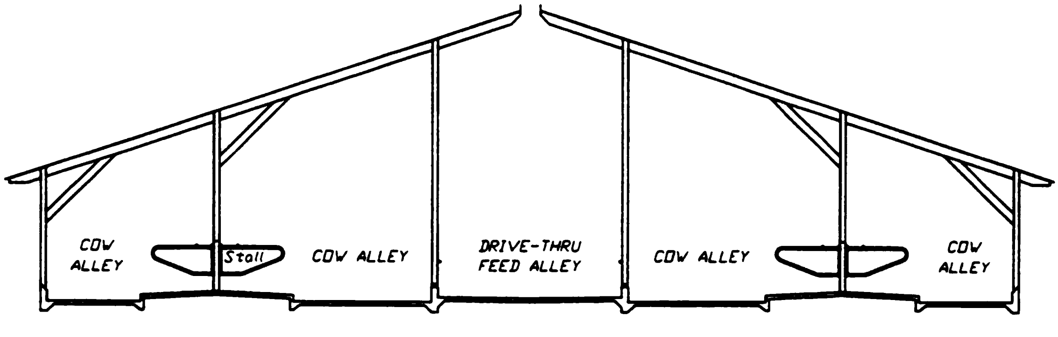
(1967) implied that equivalent results are obtained from heat transfer studies with physical cow models by using a single cylinder having an equivalent total surface area and a diameter equal to that of the modeled animal's mean trunk diameter.

The diameter and length of each cow were modeled according to Wiersma's suggestion. That is, the diameter was modeled as the mean trunk diameter of the cow and the modeled length was initially calculated as the total surface area divided by the cylinder circumference. Mean trunk diameter can be obtained from ASAE Standards (1991) for several livestock species either directly or based on trunk circumference.

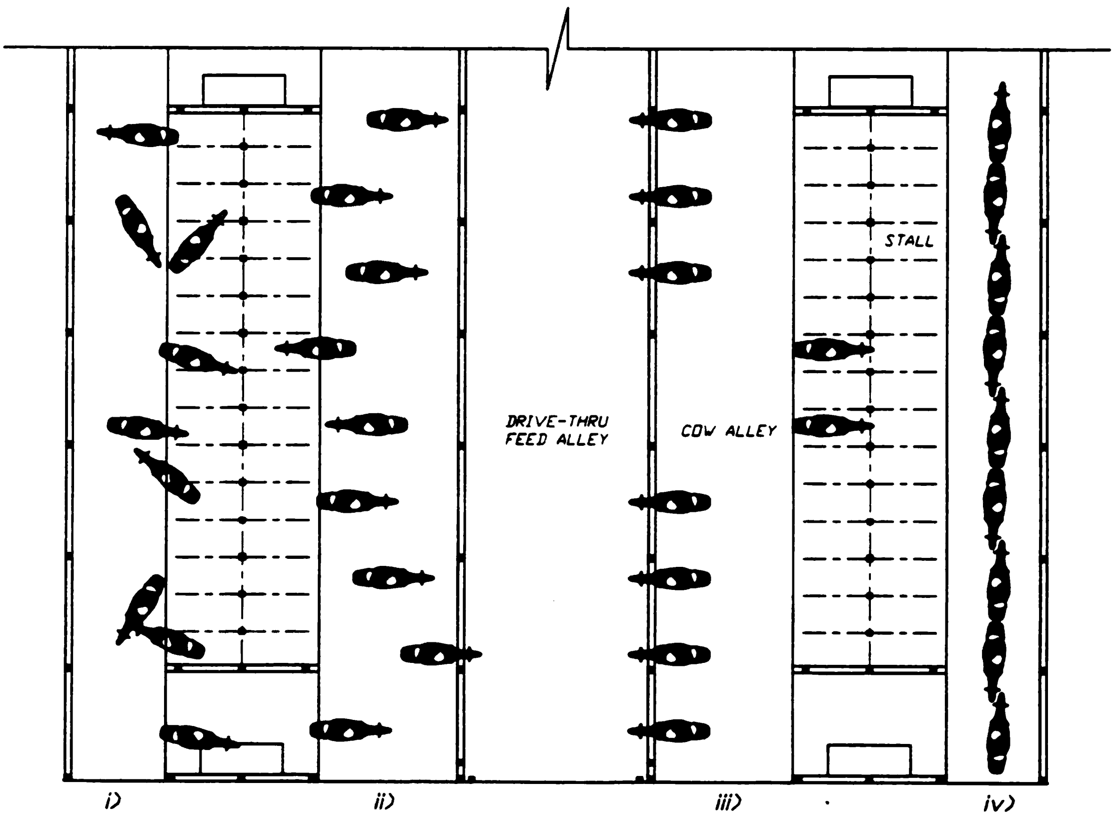
A question arose as to whether surface area should be assigned to the individual cylinder ends and, if so included, whether the ends would participate in heat loss from the animal. The assumption was made that the model cow's total surface area should be divided between both the outer cylinder area and the area required to cover the ends of the cylinder. This assumption was made because animals logically should be modeled as closed bodies and also because animal lengths were more realistic when modeled in this manner. Then it was assumed that the cylinder ends would not participate in heat exchange since they were butted up against the ends of adjacent animals and were not exposed to air flow.

Standing position. The cows are assumed to be in a standing position. Real cows do not always stand up. Actually, the purpose of the stalls in the barn is to provide a comfortable place for the cows to lie down. However, cows do spend a substantial portion of the day standing. Even more time is spent standing during hot weather as the cows attempt to expose more body surface area to air movement. Thus the standing position representation may be accurate for potentially stressful environmental conditions but doesn't hold for continuous or mild weather modeling purposes.

Row alignment. The assumption that the cows are lined up end-to-end or head-aside-head in one or more continuous rows cannot immediately be justified as an accurate description of real conditions in livestock buildings. Figure 4.2.2 illustrates four possible ways that cows might align themselves in a free stall barn. The first distribution suggests that the cows randomly choose a location within the barn and also assume a random orientation. In reality, however, they can only locate themselves within the alleys or stall spaces, so the completely random arrangement is an unrealistic representation.



BARN CROSS SECTION



OCCUPIED FLOOR PLAN

Figure 4.2.2 A free stall barn cross section and an occupied floor plan showing possible representations of cow position and alignment within the barn; i) completely random, ii) randomly-located transverse alignment, iii) rows of transversely aligned cows and iv) in-line rows.

The arrangement of stalls and the placement of feed in typical free stall barns affects where and how a cow positions herself within the barn. Because of these features, cows will tend to assemble in rows running the length of the barn and align themselves aside one another with their bodies perpendicular to the sidewalls. The most accurate model of cow positioning is likely some variation of the second and third schemes shown. Unfortunately, the more realistic representations pose several difficulties in modeling heat loss from the animals via any one heat transfer mechanism, much less via several in combination. Hence, the simplified in-line representation of the cows was chosen. In real free stall barn environments, there is no rigid constraint to make the cows distribute themselves along the length of the barn. In fact, cows tend to bunch into huddled masses during repressively hot conditions. However, the assumption was made that the regular distribution of stalls and feeding space along the length of the barn justifies a continuous row representation for most summer environmental conditions and for basic thermal energy status comparisons.

In its current form, ANTRAN models only one row of animals within the barn enclosure. It was not assumed that all the animals that would normally be housed in a livestock barn would, nor could, position themselves into a single lengthwise row. The single row assumption is purely a starting point from which to develop improved methods of modeling the entire herd of housed animals.

4.3 Modeling the Thermal Energy Status of the Cows

Certain assumptions had to be made to model the thermal energy status of the animals. Acclimatization, acclimation and the method of determining the cows' present thermal state were considered in the development of the model. The resulting process block diagram is presented in Figure 4.3.1.

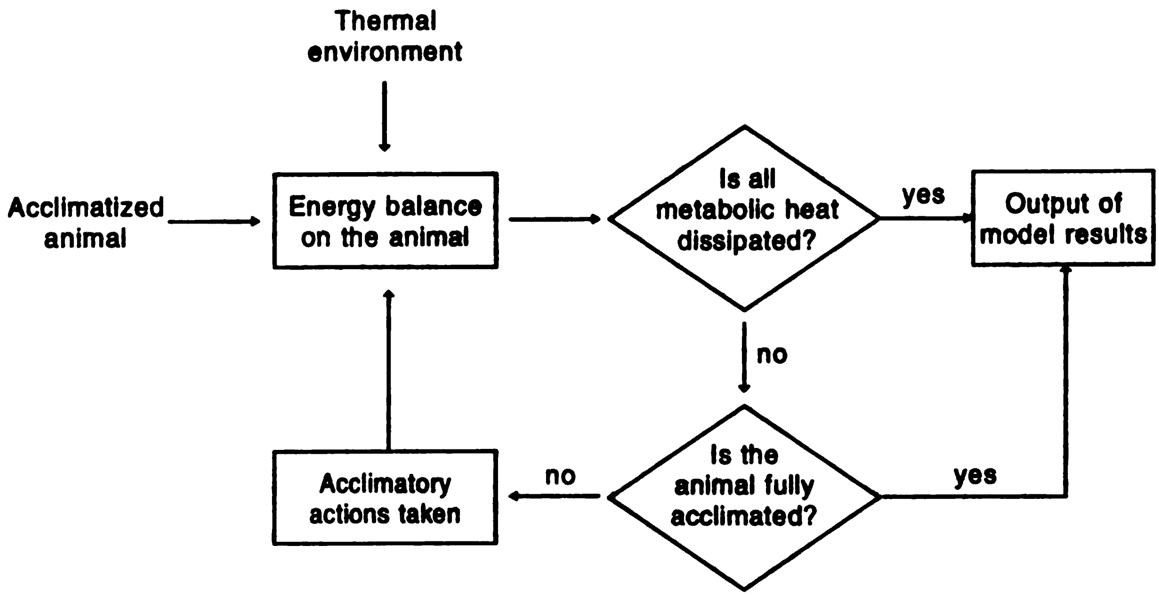


Figure 4.3.1 Block diagram of the decision process used in modeling the thermal energy status of the animals.

Acclimatization. The animals were assumed to be acclimatized to warm conditions. For dairy cows, this means that the hair coat is in summer condition, minimal tissue resistance and maximal sweating rates are readily maintained, and the cow is not constrained from further acclimating to the surroundings. Although tropical breeds of cattle may make seasonal adjustments in body temperature, it was assumed that this effect was not common in European breeds in temperate regions, so body temperature adjustment was modeled as a mechanism of acclimation only.

Thermal energy status. A fundamental assumption incorporated into AN-TRAN was that feed intake and metabolic heat production are not immediately reduced by hot conditions. An animal's thermal energy status in the model is indicated both by the skin temperature that can be maintained and by the difference between the heat produced by the cow and the heat that can be dissipated to the environment at normal pre-stress metabolic rate. Thus, if a critical temperature difference cannot be maintained between the body core and skin surface or if heat production exceeds dissipation, then a rise in body temperature is impending. If such a situation exists even when the animals have taken full acclimatory actions, physiologic reality dictates that heat production must be reduced and milk yield will be adversely impacted.

It is important to understand that the underlying assumption throughout the model is that heat production is not reduced and that homeothermy may not be achievable at all times. Stated another way, ANTRAN emphasizes the projected heat dissipating capacity of the animal within its environment and the projected net heat load on the animal if metabolic activity is not reduced.

This approach was taken largely because, although there are a number of relationships that have been developed to predict feed intake and milk production in warm environments, the limitations on the appropriate use of available relationships were generally substantial. No relationships were found that were adequately demonstrated to be accurate outside the environmental conditions in which they originated and none was suitably tied to skin temperature or a projected thermal energy imbalance.

As shown in Figure 4.3.1, heat flow from the body and the thermal energy status of the animals were initially modeled at thermoneutral acclimatized conditions. If the cow's thermal energy status reveals that a surplus of thermal energy exists, then the energy balance was performed again with the cow being in a more acclimated state. This process is continued until either a thermal energy balance is achieved or all acclimatory actions are exhausted by the animal.

Acclimation. It was assumed that, beyond sweating, the principal means of acclimation utilized by the cows were elevated respiratory activity and an allowance for some rise in body temperature. Although some research reports suggest that respiratory activity responds directly to the environment and a rise in respiration rate precedes any rise in body temperature, no suitable relationship exists to confidently define when the two mechanisms are exercised. Also, a larger body of literature suggests that elevated respiration rate is a coincident signal that homeothermy is already not being maintained. Consequently, respiratory ventilation rate and body temperature are increased simultaneously if additional dissipation of body heat is required, each being constrained within some maximum allowable level.

It was assumed that if the cow can maintain homeothermy without expending resources to acclimate to the surroundings, then milk production is fully maintained. If cows must acclimate to the environment, then milk production may still be maintained. But, more likely, feed intake will fall to help offset the need to maintain the mechanisms of acclimation and, consequently, milk yield will fall slightly also. The interactions involved in these situations and the related implications on the production and health of the cow are not well understood (Beede, 1995). Certainly, if the cow is not able to increase the rate of heat dissipation through physiological acclimation, heat production must be reduced and the cow's health and productivity will likely suffer.

4.4 Modeling Heat Transfer from the Animal's Body

Heat flow from the body was assumed to occur via the respiratory system and from the exterior body surface of the animal as shown in Figure 4.4.1. Here, the various forms of heat transfer from the body surface are lumped together into q_{surf} for simplicity. Other minor avenues of heat loss were considered negligible. One such avenue that has been considered by others (Turner, et al., 1987a) is heat loss when ingested food and water are warmed to body temperature before being excreted. As with other purely conduction processes, it was assumed that this minor mode of heat loss would become negligible under warm conditions. If the drinking water or feed was chilled, this assumption would need reconsideration.

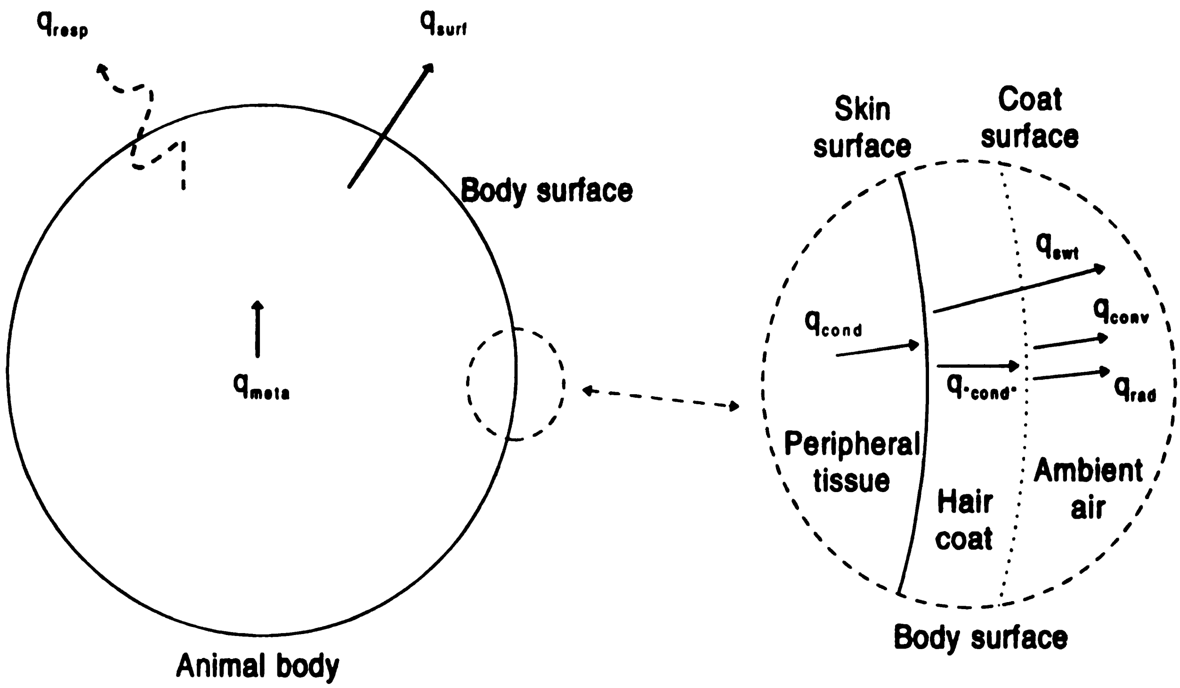


Figure 4.4.1 Schematic of modeled heat flow from the cow detailing heat loss from the two-dimensional body surface.

Uniform body core temperature. Metabolic heat was assumed to be produced and distributed throughout the body at a constant rate and in a manner that produces a uniform core temperature. Metabolism does not really occur at a constant rate, but the digestion process in lactating cattle is extended throughout the day sufficiently to justify that assumption. Warm blooded animals have a cardiovascular system that supports a large heat-carrying capacity. This makes the uniform body temperature assumption plausible. It must be recognized, however, that body extremities, especially in appendages, may not be at body temperature. It has already been stated that the cylindrical representation of the animals restricts the modeling of heat loss from appendages. It was assumed that the variability of temperature within the body is, at most, slight during warm environmental conditions.

Respiratory heat loss. Both sensible and latent heat loss from the lungs are accounted for within ANTRAN. In warm conditions, heat loss is largely in latent form, but it is convenient to model sensible heat loss from respiration so it was included. The expired air was assumed to be saturated at a temperature slightly cooler than body temperature per the research of Stevens (1981) described earlier.

Respiration rate and respiratory ventilation rate were not assumed to be functions of ambient temperature alone, however, as was assumed by Stevens. The equation for estimating expired air temperature was quite accurate ($R^2 = 0.96$), but those for respiration rate and tidal volume were less accurate ($R^2 = 0.94$ and 0.81, respectively). The assumption employed in the formulation of ANTRAN was that respiratory activities are not strictly a function of one or more ambient conditions, but are a function of the thermal energy status of the cow. Therefore, respiratory activity is modeled as a mechanism of acclimation that is directly related to body temperature.

Heat loss from the surface of the body. In the model, respiratory heat loss is subtracted from metabolic heat production. Any excess heat must be dissipated across the body surface. It was assumed that this heat is carried in the arterial blood stream to the vascular tissues and that this process does not limit heat flow from the body. Heat is then transferred to the peripheral tissues below the skin by the vascular blood supply system. From there, the entire balance of residual body heat must be transferred by conduction across the peripheral tissue.

Heat is lost from the skin by sensible mechanisms and by evaporation of surface moisture. It was assumed that the heat required to evaporate the moisture comes entirely from the skin rather than from the air within the hair coat. Sensible heat loss occurs by conduction, radiation and local convection currents. However, the flow of heat from these mechanisms is frequently described using an effective thermal conductivity of the hair coat in the heat transfer calculation as is highlighted in the review of literature. The resulting all-inclusive sensible heat loss component is accordingly labeled $q_{\cdot cond^{\cdot\cdot}}$.

The water lost from the skin was assumed to be transported directly to the surface of the hair coat and away from the body by the movement of air about the body. No condensation or evaporation within the hair coat is accounted for in the model unless the cow is soaked with water.

Sensible heat that reaches the outer surface of the hair coat was assumed to be lost to the environment via the convective cooling of the ventilation air and by radiant heat exchange with interior enclosure surfaces. It was recognized that either of these processes could add heat to the hair coat under extreme conditions. Positive heat flow implies that heat leaves the hair coat, negative flow is a heat gain in the model.

Metabolic heat production. Several equations for estimation of the metabolic heat production of lactating dairy cattle within the thermoneutral zone were presented in the review of literature. The criteria used in selecting an appropriate relationship were that the equation should consider the combined effects of maintenance, pregnancy and milk production and yet should provide realistic estimates of total heat production. Based on these criteria, either Equation 3.8.2 or 3.8.3 was acceptable.

Assumptions had to be made about the weight and productive status of the animals to arrive at an estimate of heat production. Hence it was assumed that all the cows weigh the same, are of the same reproductive standing and produce equivalent quantities of milk. Of course, this assumption never holds true on real farms, especially on a herd-wide basis. It does closely approximate groupings of cattle that are often sorted on the basis of these characteristics within large herds. If herd averages are used for input values in the equations, the user should be aware that model results may not be truly representative of the herd since any cows differing substantially from the average modeled cow will not be accurately modeled and because cows of differing size and productive standing are seldom uniformly distributed throughout the barn.

Body surface area. A single value of surface area to model all of the mechanisms of heat loss from the skin was desired. The effect of disregarding the thickness of the dairy cow's summer hair coat (1 to 3 mm) when calculating surface area for convection was examined. The error in neglecting any difference in surface area due to the presence of the hair coat is at most three percent for a 0.6 m diameter (ASAE, 1991) cow modeled as a cylinder or sphere. This margin of error is certainly small compared to the errors introduced when measuring or estimating hair coat depth, trunk diameter and base surface area. The surface area used for all heat transfer calculations was determined by subtracting the area of the cylinder ends from the area given by the Meeh relationship, Equation 3.8.5. If ANTRAN is used with animals having proportionately thicker hair coats, then a separate surface area for convection or modification of the sensible insulation provided (per Equation 3.8.11) must be determined.

Resistance circuit model. A resistance circuit approach was used to model heat flow from the outer surface of the cow. Figure 4.4.2 illustrates how the physical flows of heat and mass, the temperature and vapor pressure gradients that drive the flows and the circuit representation are interrelated. The physical flow portion of the figure illustrates in detail how the flows must equilibrate at each surface. A liquid layer of unspecified thickness is also shown on the outer surface of the skin. Prolific sweating or sprinkling with water are example situations that might require this representation. The liquid layer is drawn for illustrative purposes primarily as it was assumed that the sweating rate of cattle would not be sustained at a level that would form a thick layer of moisture and sprinkling of the animals was not assessed.

A surface energy balance is applied at the skin and outer hair coat surface to establish the equations relating the various heat flows. First, the heat leaving the skin surface must equal that entering. Similarly, the heat leaving the outer surface of the hair coat must also equal that entering with the requirement that the effective conduction of sensible heat through the hair coat, $q_{\text{"cond"}}$, be the same in both equations.

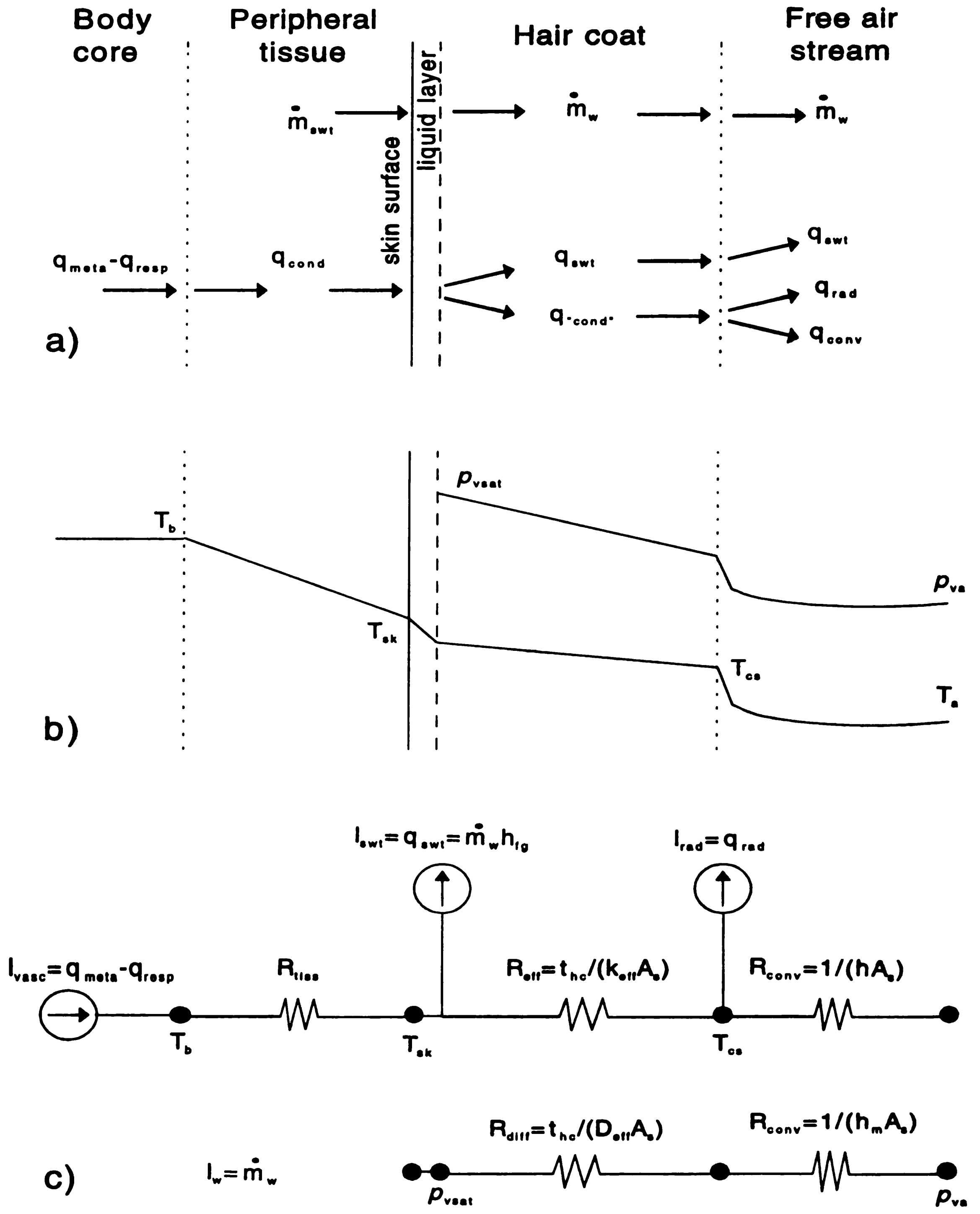


Figure 4.4.2 Model representations of the a) physical flow, b) driving potential and c) resistance circuitry of heat and mass transfer from an animal's outer body surface.

Heat approaching the skin is the residual metabolic heat that is being conducted across the peripheral surface tissue, hence the designation q_{cond}. Heat is removed from the evaporative surface as sweat or other moisture evaporates. Since evaporative heat loss q_{swt} is in latent form, it is not involved in the heat balance at the hair coat surface although conservation of mass must be satisfied throughout the system.

At the outer surface of the hair coat the thermal energy balance is comprised entirely of sensible heat flows. The rate of sensible heat flow through the hair coat must equal that of convection q_{conv} and radiation q_{rad} which occur simultaneously to remove heat from the coat.

$$q_{cond} = q_{"cond"} + q_{swt} \tag{4.4.1}$$

$$q_{"cond"} = q_{conv} + q_{rad} \tag{4.4.2}$$

If a liquid layer developed over the skin, the effects of the layer on the liquid's surface temperature and saturation vapor pressure, the effective conductivity of the coat and the resistance to diffusion of water vapor through the coat each would need to be addressed. And if the moisture was incrementally applied by external means, the dynamic character of the ensuing drying layer would have to be considered. Since moisture for vaporization from the skin comes entirely from internally supplied sweat, these concerns are not relevant in this study and the requirement for conservation of mass reduces to one of constant flow of moisture with the limitation that vaporization rates do not exceed the animal's maximal sweating rate.

Turning to the thermal and vapor density gradient diagrams, it should be evident that skin temperature must be maintained below body temperature for heat to flow by conduction from the body. Normally, a continuously decreasing temperature gradient will exist from the skin to the ambient air as well. However, at extremely high ambient air temperature or under substantial radiant heat load, the thermal gradient may be reversed within the hair coat so that the temperature of the skin is below the temperature of the coat itself. The vapor pressure at the skin surface of sheltered animals will be at or above that of the ambient air during summer conditions even if the animal is not sweating. This ever-present vapor pressure gradient within the hair coat is the driving force that feeds the vaporization process that cools the skin.

Finally, the analogous resistance model defines how the heat and mass flows from the animal are modeled within ANTRAN. Each flow (electric current analogy) is either modeled as a difference in potential (as in voltage) and an equivalent resistance or as a heat (current) source/sink.

Surface moisture flow. Starting with the simple water vapor circuit, moisture flows from points of high vapor pressure to lower vapor pressure. Thus, moisture must move from the skin to the ambient air for all realistic warm weather circumstances encountered. The movement of moisture is resisted by both the nature of the hair coat and the character of the air moving about the body. These resistances act in series which simplifies the model since the mass flow rate can thus be expressed as the total vapor pressure difference divided by the sum of the resistances.

$$\dot{m}_w \equiv I_w = (\rho_{vsat} - \rho_{va})/(R_{diff} + R_{conv}) \tag{4.4.3}$$

where: $\dot{m}_w$ = mass flow rate of water vapor (kg/s);

I_w = analogous current for water vapor flow (kg/s);

ρ_{vsat} = saturation vapor density at liquid interface (kg/m^3);

ρ_{va} = vapor density of free stream air (kg/m^3);

R_{diff} = resistance of hair coat to diffusion of water vapor (s/m^3); and

R_{conv} = resistance to convection of water vapor from hair coat (s/m^3).

The resistances were modeled as shown in Equations 3.8.26 and 3.8.27 except that the resistance terms include the surface area A_s. This was done so that heat flow is described rather than heat flux. Also, the rate of moisture loss by evaporation was limited to the maximal sweating rate that could be sustained by the cows.

$$R_{diff} = t_{hc}/(D_{eff}\,A_s) \tag{4.4.4}$$

$$R_{conv} = 1/(h_m\,A_s) \tag{4.4.5}$$

where: t_{hc} = thickness of the hair coat (m);

D_{eff} = effective mass diffusivity of water vapor into air (m^2/s); and

h_m = convective mass transfer coefficient (m/s).

Surface heat flow. A few mechanisms of heat flow were modeled as heat (analogous current) sources or sinks. The residual metabolic heat load is modeled as a vascular source, I_{vasc}, vaporization from the skin as a latent sink, I_{lat}, and the net radiant heat load from the surroundings as a radiant sink, I_{rad}. The vascular and latent currents are modeled in Equations 4.4.6 and 4.4.7. The radiant current is determined from a matrix solution of the radiant energy exchange within the enclosure which is described later. Energy required to evaporate the surface moisture was assumed to be provided by the body via the skin rather than by the air within the hair coat. The literature shows this assumption holds as long as evaporation occurs from the skin.

$$I_{vasc} = q_{meta} - q_{resp} \tag{4.4.6}$$

where: q_{meta} = rate of metabolic heat production (W) and

q_{resp} = rate of respiratory heat loss (W).

$$I_{swt} = \dot{m}_w h_{fg} \tag{4.4.7}$$

where: I_{swt} = analogous current for evaporative heat flow (W);

$\dot{m}_w$ = mass transfer rate of water vapor (kg/s); and

h_{fg} = latent heat of vaporization of water (J/kg).

Heat flow through the peripheral tissue was modeled as in Equation 3.6.8. The thermal resistance (or insulation) of the peripheral tissue is held constant within the model at its minimum value. In reality, tissue insulation is rapidly lowered to a minimum level under hot conditions, but it also varies with the environment to balance heat loss with the residual metabolic heat load. The objective of this effort was not to model the exact nature of heat flow under mild conditions, but rather to model heat flow in stressful thermal environments. Therefore, excess heat loss was accounted for solely through decreased sweating rate. It seems logical that if too much heat is being lost from the skin, reducing the amount of moisture that can be vaporized from the skin is the most expedient means available to the animal to maintain a reasonable skin temperature and rate of heat loss from the body.

Sensible heat flow through the hair coat is evaluated as a conduction phenomenon using a thermal resistance that incorporates an effective thermal conductivity as shown in Equations 4.4.8 and 4.4.9.

$$q_{\text{"cond"}} = (T_{sk} - T_{cs})/R_{eff} \tag{4.4.8}$$

where: $q_{\text{"cond"}}$ = rate of sensible heat flow through the hair coat (W);

T_{sk}, T_{cs} = skin and hair coat surface temperature, respectively (°C); and

R_{eff} = thermal resistance of the hair coat (°C/W).

$$R_{eff} = t_{hc}/(k_{eff} A_s) \tag{4.4.9}$$

where: t_{hc} = thickness of the hair coat (m);

k_{eff} = effective thermal conductivity of hair coat (W/m °C); and

A_s = exposed body surface area (m^2).

The rate of convective heat loss from the coat is modeled as shown in Equation 4.4.10 where the external resistance to convection is presented in Equation 4.4.11.

$$q_{conv} = (T_{cs} - T_a)/R_{conv} \tag{4.4.10}$$

where: q_{conv} = convective heat loss from the hair coat (W);

T_a = free stream air temperature (°C); and

R_{conv} = external thermal resistance of the hair coat (°C/W).

$$R_{conv} = 1/(hA_s) \tag{4.4.11}$$

wherein: h = convective heat transfer coefficient (W/m^2 °C).

The convective heat transfer coefficient is calculated using the relations reported by Wiersma rewritten in Equations 4.4.12 through 4.4.14. These relationships and the previous equations are in harmony with the selected animal physical model as long as the hair coat remains relatively thin.

$$h = \bar{N}u(k/d_b) \tag{4.4.12}$$

$$\bar{N}u = 0.65\ Re_d^{0.53} \text{ for } 8 \times 10^3 < Re_d < 1.5 \times 10^5. \tag{4.4.13}$$

$$Re_d = U_\infty d_b/\nu \tag{4.4.14}$$

where: Nu, Re = Nusselt and Reynolds number, respectively;

k = thermal conductivity of air (W/m K);

d_b = mean trunk diameter (m);

U_∞ = free stream air velocity (m/s); and

ν = kinematic viscosity of air (m^2/s).

4.5 Structure of the Computer Program

ANTRAN was written in the Fortran programming language using the MS Fortran® software package. The program is designed to operate using input that could either be entered by the user at the keyboard or retrieved from an ASCII data file. The basic structure of the program is illustrated in Figure 4.5.1.

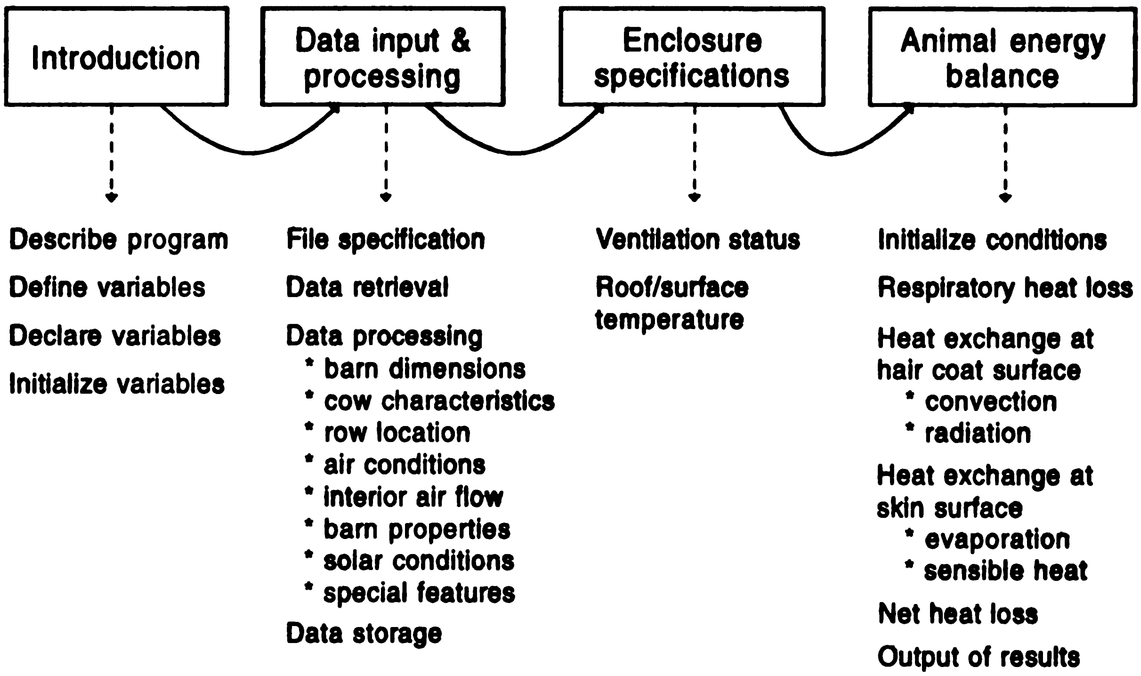


Figure 4.5.1 Structure of the computer model, ANTRAN.

After providing a brief introduction to the program user, ANTRAN defines and declares all the required program variables. Constant values and variables that are rarely modified are initialized so the program can gather input data. In the data input and processing portion of the program, sources of input and output data are specified, as are names and paths for data files. Then input data are retrieved and subsequently processed to provide values having units appropriate to the basic model variables.

Default values are provided for each variable based on Holstein cows in a partially open four-row barn located in Lansing, MI during solar noon on a summer day. The user may accept the defaults or enter new input values. Input data are screened for values that are unrealistic or that might produce an error in the program. Brief descriptions of the inputs that are needed in the program follow. Once all required data are entered, the variable values are sent to a storage file. This allows the user to define default input data which may be very helpful for multiple model runs.

a) Structural dimensions - the program user is first requested to enter the barn width, wall height, roof pitch and wall open area. From these data, the segments of the barn enclosure and the cow row are described in Cartesian coordinates and their surface dimensions are calculated.

b) Livestock characteristics - the species, weight, production level, gestation state and other details describing the animal are entered next. From these data, all the necessary physical characteristics of the cows are determined in preparation for calculating the flow of heat to, through and from the body.

c) Animal position - the location of the row of cows is determined with input from the user. By default, ANTRAN initially locates the row of cows inside the windward sidewall by one-third the width of the barn. This generally coincides with a cow alley along a feeding area.

d) Ambient air conditions - air temperature, relative humidity and atmospheric pressure must be provided. ANTRAN uses these data to calculate all the pertinent psychrometric values that may be utilized within the model, as well as to compute the temperature humidity index for reference.

e) Air flow parameters - wind speed and direction and the coefficient of discharge for the wall openings are entered next. ANTRAN calculates the air speed in the animal zone based on the building ventilation rate and continuity. Alternatively, velocities obtained from other models can be entered. The ventilation rate through the building is calculated according to Equation 3.11.2 using 0.5 to 0.6 for the opening effectiveness as recommended by Bottcher (1995). Ventilation rate is reduced for angled winds according to Equation 3.11.4.

f) Surface properties and level of insulation - emissivity values of the building components are specified for both long-wave and short-wave radiation. The barn's roof is initially uninsulated. A fully-insulated roof can be simulated wherein the underside temperature is assumed to equal inside air temperature.

g) Solar conditions - for some advanced model applications, the solar date and time, orientation of the barn, and other data are required. In this study the ambient radiant conditions are entered directly.

h) Special features - some of the model relationships can be used in situations where water is applied to the skin surface. Since soaking cows with sprinklers was not considered in this study, evaporation from the cows was limited to the cows' maximal sweating rate.

The next segment of the program specifies the nature of the enclosure. This small segment of the program accommodates use of the model under special circumstances. As described in the following chapter, special considerations are taken to allow the program to simulate laboratory conditions.

Lastly, a thermal energy balance is calculated for the animals within the enclosure through an iterative process. When performing the energy balance, the program first performs an energy balance on the acclimatized, but unacclimated animals as described previously in this chapter. Heat generation by and dissipation from the cows is calculated. The radiant energy balance of the row of cows is performed using a program called RADENC (Copyright W.A. Thelen, 1993). This program calculates the net heat flux of each pertinent surface within an enclosure given the complete set of view factors and either a constant temperature or heat flux value for each surface. View factors for the building enclosure surfaces and the row of animals are determined through the use of basic two-dimensional view factor equations. Because the row of cows blocks the view of some interior surfaces by others, extensive use of Hottel's equation (Gray and Müller, 1974 and Siegel and Howell, 1980), sometimes referred to as the method of strings, is employed.

If the cows are not in a state of thermal equilibrium, ANTRAN simulates acclimatory actions on their part and performs the energy balance again. If equilibrium can be attained through temporary acclimatory actions, the program terminates and creates a summary of output data. If equilibrium cannot be achieved via acclimatory actions other than through a reduction in metabolic heat generation, the program terminates at the maximal allowed acclimation level, reports that thermal equilibrium is not attainable for the given conditions and creates a summary of output data which includes the surplus heat load upon termination of the program run. A sample listing of program output is provided in Appendix B.

5 - MATERIALS AND METHODS

The heat transfer model was evaluated and utilized in two stages:

i) The computer program was used to model the response of dairy cattle housed within artificial environments simulating the conditions of a previous research study conducted in environmentally controlled facilities and the model results were qualitatively compared to measurements taken during that study; and

ii) The computer model was used to project the impact of some features of naturally ventilated dairy facilities on the thermal status of the housed animals during various weather conditions.

5.1 Comparing Model Results with Environmental Chamber Measurements

Two references cited in the literature review, Kibler and Brody (1954) and Thompson, et al. (1954), report results from controlled environmental chamber studies with dairy cattle that assessed the effect of varied air speeds on cows' thermoregulatory responses. The interior chambers were sufficiently insulated and isolated so that chamber surface temperatures were maintained very close to interior air temperature.

ANTRAN was modified to simulate such an environmental chamber by approximating the chamber size, covering the sidewall openings, prescribing that surface temperatures equal air temperature and artificially imposing on the cows a stream of air that was equivalent to that used in the chamber studies. The option of covering the sidewall openings was provided to avoid any difficulties that might arise in the calculation of shape factors with zero surface area.

In the environmental chamber study, several breeds of cows were monitored at different temperature and air velocity combinations. Air speed was controlled within the chamber and within the model at three velocity levels characterized as follows:

Low = 0.2 m/s (0.4 to 0.5 mph);

Medium = 2.0 m/s (3 to 6 mph); and

High = 4.0 m/s (8 to 9 mph).

Comparisons of model output and data obtained from the environmental chamber study were made for the low and medium air flow conditions. The air speeds used in the high velocity (4.0 m/s) chamber trials are beyond the appropriate use of this model in its current form (Reynolds number exceeds 150,000 upper limit), so those conditions were not simulated. Conditions at low air velocities were modeled, but it should be recognized that Reynolds numbers for such conditions are at the lower fringe of ANTRAN's usable range (Re ~ 8000).

ANTRAN was used to simulate only those environmental conditions with air temperature above 15 °C (59 °F) because the model components were primarily derived and intended for use in warm conditions. Air temperature was raised from 10 °F to nearly 100 °F in the chamber study and the relative humidity was generally maintained at 60 to 70%. Each of the following temperature and humidity combinations were simulated at the low and medium air velocity levels: 15, 20, 25, and 30 °C @ 65% RH; 35 °C @ 60% RH; and 40 °C @ 50% RH. The relative humidity was lowered slightly as the temperature rose since the physical relationship of relative humidity with temperature and the wording of the reports suggest that the variation in moisture content of the air would result in lower humidities at higher temperature.

Lastly, to simplify predictions and comparisons, only Holstein cows were considered. Three Holstein cows were used in the chamber study. Average characteristics were used as inputs to the model and the results were compared to the reported averages for that breed. The specific inputs that were held constant for the simulation runs are provided in Table 5.1.1.

Table 5.1.1 Cattle and building inputs for validation simulations.

Cow characteristics	Building characteristics
Weight: 600 kg	Width (length of flow): 8 m
Production: 24 kg/d	Wall height: 2.75 m
Gestation: 0 days pregnant	Roof slope: 1: 12
Trunk diameter: 0.65 m	Surface emissivity: 0.95
Coat thickness: 2.0 mm	
Max. sweating rate: 300 $g/m^2/hr$	
Coat emissivity: 0.95	
Normal body temperature: 38.5 °C	
Maximal short-term rise in body temperature: 3.0 °C	
Normal respiratory ventilation rate: 100 L/min.	

The Holstein cows in the chamber study were larger and higher producing than the average 1950's era cow. Because of their size and production, their cylindrical representation was assumed to be typical of today's animals and the trunk diameter reflects that assumption. Also, the warm weather portion of the chamber study was performed in early spring. It was assumed that the cows would retain some remnants of their winter coat as might be observed in warm housing; i.e., a thick summer-condition coat was assumed.

No published value exists for the maximal short-term rise in body temperature that can occur without affecting production. Several producers and at least one experienced researcher in this area (Beede, 1995) were confident that some rise in body temperature and respiration rate can be tolerated by cows for a few hours a day without affecting production as long as the cows had opportunities later in the day to dissipate that excess internal heat. From these contacts, an extreme value of 3 °C was selected for the maximal rise in body temperature allowed during the simulations. The simulations were conducted under the assumption that elevated body temperature and respiration rate over a short duration would not affect daily feed intake and milk production, without considering potential effects on reproduction or other health measures. ANTRAN does allow this value to be reduced during input and preliminary output is always produced during the initial simulation iterations corresponding to the assumption that feed intake falls as soon as body temperature begins to rise.

Simulations were then performed given the parameters just described. Simulated skin and hair coat surface temperatures were compared to values measured (with thermocouples) in the chamber study. Due to the limited amount of experimental data, statistical measurements were not performed and few statistical measures were provided to evaluate the reliability of the chamber data. The data from the series of environmental chamber studies, of which these trials are a part, are still the best of very limited data of this nature available.

5.2 Projecting the Effect of Weather and Barn Features on Heat Loss

Next, the model was used to evaluate how heat transfer from the cows was impacted by different weather conditions and changes in pertinent building design features. The cow representation used for this portion of the study was that of Holstein cows in mid-lactation on a modern dairy operation. In addition to being with calf, these animals possess a somewhat larger frame and level of production than were used for the previous validation simulations. The revised characteristics of the cows are given in Table 5.2.1.

Table 5.2.1 Cattle characteristics used as model input for projections.

Weight: 635 kg	Production: 30 kg/d
Gestation: 120 days pregnant	Trunk diameter: 0.70 m
Coat thickness: 1.4 mm	Max. sweating rate: 300 g/m^2/hr
Coat emissivity: 0.95	Normal body temperature: 38.5 °C
Maximal short-term rise in body temperature: 3.0 °C	
Normal respiratory ventilation rate: 100 L/min.	

Effect of varied weather conditions. Each trial was evaluated for mid-day conditions from atypical days characterized as i) 'hot & breezy' and ii) 'calm & muggy'. Weather data for Lansing, MI were obtained from the Michigan Meteorological Resources Program at Michigan State University. These data are archived by the National Climatic Data Center in Asheville, NC and include measured and modeled solar radiation values. Weather conditions are summarized in Table 5.2.2.

Table 5.2.2 Summary of weather conditions on comparison days.

Parameter	Hot & breezy	Calm & muggy
Date	6/25/88	8/27/90
Time (local)	1:00 pm	1:00 pm
Dry-bulb temperature (°C)	36.1	29.4
Relative humidity (%)	35	75
Wind speed (m/s)	13.4	3.6
Wind direction (degrees from north)	270 (W)	230 (SW)
Direct beam shortwave solar (W/m^2)	848	117
Diffuse solar on horizontal (W/m^2)	142	341

The weather data are mid-day values recorded near solar noon. This time was selected to best represent highly shaded conditions inside the barn so that only diffuse radiation would enter through sidewall openings. The hot & breezy day weather conditions were relatively dry with clear skies. By comparison, the calm & muggy day conditions were not as hot, much more humid, and included the effects of hazy skies. These weather conditions are rather extreme for the Lansing area and were selected primarily for illustrative use.

Effect of varied building parameters. Simulations of heat loss from a row of dairy cows were made as one building design feature was varied during each simulation. The features evaluated were barn width, percentage sidewall open, sidewall height and roof insulation level. Descriptions of the control barn features and of the variations considered are shown in Table 5.2.3. Ample insulation was taken to imply that the interior roof surfaces would be at ambient air temperature. Without insulation, the roof underside temperature was assumed to be at least 10 °C warmer than the air temperature. This approximation was derived by assessing roof underside temperature measurements taken at six locations within each of two free stall barns in mid-Michigan over a one-month summer period (8/95).

Table 5.2.3 Summary of barn features and alterations evaluated in the model.

Barn feature	Control	Alteration
Width	27 m	18 m
Wall height	3 m	2 & 4 m
Percent open	60%	30 & 80%
Roof slope	4:12	---
Ridge opening	none	---
Surface emissivity	0.95	---
Insulation	none	ample
Orientation	N-S	---

6 - RESULTS AND DISCUSSION

6.1 Comparison of Model Results with Environmental Chamber Measurements

Skin and hair surface temperature. In this section, ANTRAN simulation results are compared to reported data that were obtained from studies conducted in environmental chambers at the University of Missouri. Skin and hair surface temperatures obtained using the model and measured in the chamber experiments are shown alongside one another in Figures 6.1.1 and 6.1.2.

Figure 6.1.1 displays the cow surface temperatures where the interior air speed was 0.2 m/s. Modeled and average measured skin temperatures for low air speed conditions varied by no more than one degree Celsius at each of the three chamber study temperatures. The modeled skin temperature is higher than that measured at the lowest environmental temperature, but is lower at the warmer chamber temperatures. Modeled hair coat surface temperatures, on the other hand, were always less than measured values. Additionally, the difference between modeled and measured hair temperatures was almost 2 °C at the intermediate (26.8 °C) chamber temperature.

A similar comparison is shown for results at a 2.0 m/s air velocity in Figure 6.1.2. It should be noted that measurement data from the chamber study were not available for a chamber temperature beyond 30 °C, but data at the two lower chamber temperature levels were reported. The differences between modeled and measured

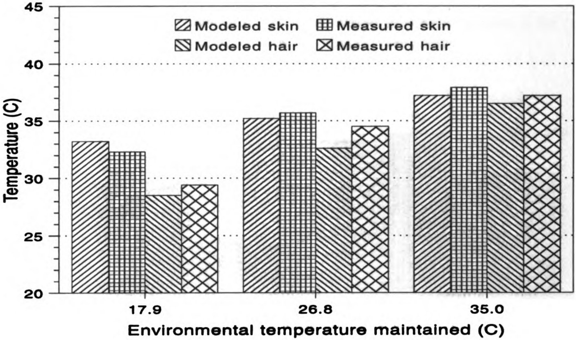


Figure 6.1.1 Comparison of modeled cow surface temperatures with reported controlled environment measurements where air velocity is 0.2 m/s and environmental temperatures are varied (RH ~ 65%).

skin temperatures are roughly 0.5 °C. In this case, differences between hair coat surface temperatures are smaller than those between skin temperatures at both chamber temperatures.

The model slightly overestimated the amount of cooling that occurs at the skin surface for most of the air temperatures and velocities considered while hair coat surface temperatures are modeled with greater accuracy at the 2.0 m/s air speed. In general, there is good agreement between measured and modeled temperatures at the higher air velocity.

With limited measurement data available in the literature, it is difficult to assess why differences between the modeled and measured surface temperatures

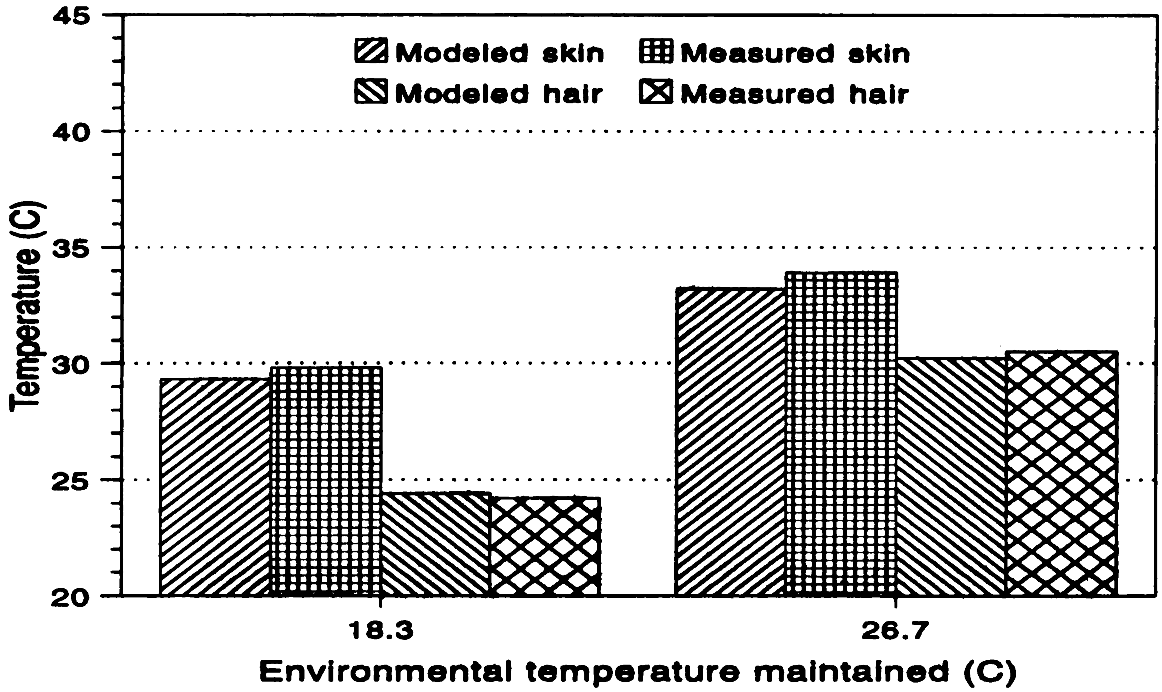


Figure 6.1.2 Comparison of modeled cow surface temperatures with reported controlled environment measurements where air velocity is 2.0 m/s and environmental temperatures are varied (RH ~ 65%).

occurred. At the lower air speed, the comparatively larger deviations may have arisen because of inaccuracies in the application of the model's governing heat transfer equations for low Reynolds number flows or, more likely, due to differences between the modeled animal representation and actual conditions. In the environmental chamber studies, air flow was fairly evenly distributed amongst the cows, but the cows were not arranged in conveniently modeled lengthwise rows. Thus, radiant heat exchange was taking place between cows and not solely between a row of cows and the chamber enclosure as was assumed by the model. At each chamber temperature, the measured hair coat surface temperature was warmer than the temperature of the chamber surfaces. Under such conditions, there would be a net transfer of radiant heat from the cows to the exposed chamber surfaces, but there would be no net exchange between cows having a similar hair coat surface temperature. Since the cows in the chamber study were partially shielded from the chamber enclosure whereas the modeled row of cows was not, the chamber study cows would lose less heat by radiant means than was assumed by ANTRAN at low air velocities. This is a possible explanation for the higher hair coat surface temperatures in the experiment compared to modeled temperatures.

Further temperature results. Figures 6.1.3 and 6.1.4 illustrate the animal temperature data produced by ANTRAN for a range of modeled environmental temperatures at 0.2 and 2.0 m/s air velocity levels, respectively. In each figure, temperatures are shown after the cow has acclimated to the thermal surroundings established by the modeled environmental chamber. Thus, a steady-state condition is described. If the environmental temperature is raised or lowered, the temperatures given by the figures are either those at the newly achieved steady-state equilibrium point (meaning the cow is dissipating as much heat as is produced) or the maximal body temperature condition (implying there is still an excess of metabolic heat).

In these figures, the dashed line labeled "Equilibrium skin" represents the skin temperature that would need to exist to achieve equilibrium without exceeding the allowed body temperature or decreasing metabolic heat production. This line further illustrates the environmental conditions at which equilibrium can no longer be achieved without decreasing feed intake. The magnitude of the difference between modeled values of skin temperature and equilibrium skin temperature indicates the level of excess heat load the cows may be experiencing.

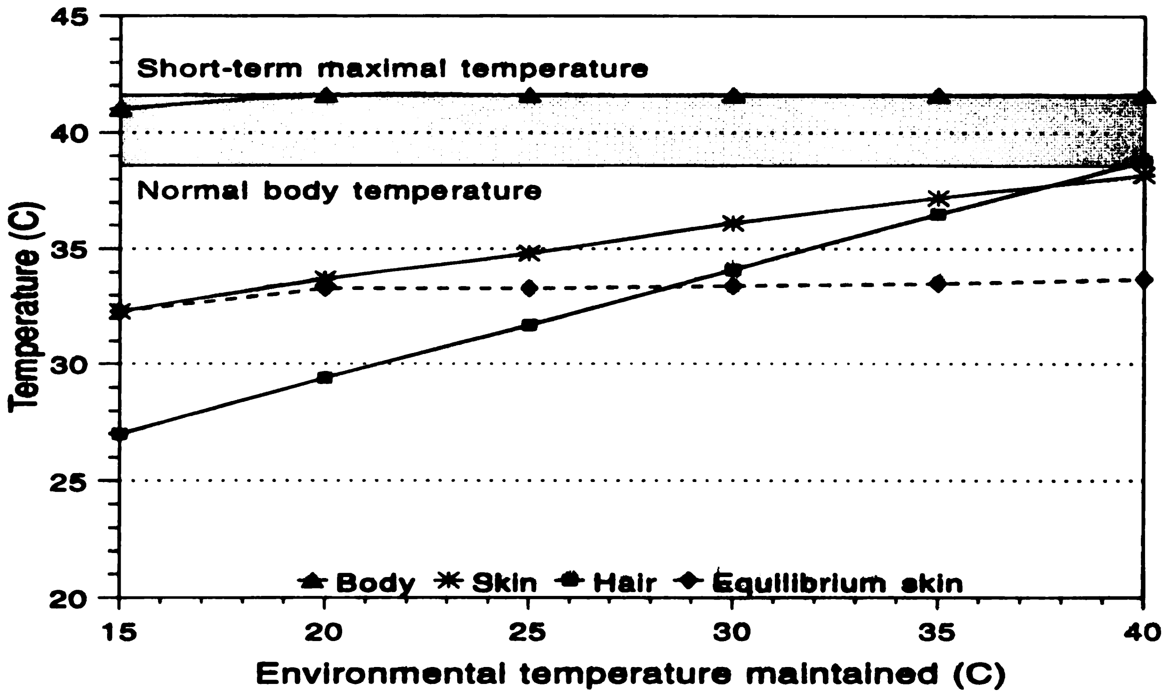


Figure 6.1.3 Modeled cow body core and surface temperatures within a simulated environment with a 0.2 m/s air velocity vs. temporary steady-state environmental temperature maintained (RH ~ 65%).

In both Figure 6.1.3 and 6.1.4, it is evident that hair and skin temperatures approach body temperature under warmer ambient environmental conditions. The clear goal of the cow in a hot environment is to maintain as large a temperature difference between the body core and the skin surface as is reasonably possible. If some temporary rise in body temperature occurs during acclimation, an improved thermal gradient results which allows either for greater heat dissipation at environmental temperatures below body temperature or for less heat gain when near or above body temperature.

The cooling potential of vaporization from the skin of the cow and the buffering capacity of the hair coat for sensible heat gain at high ambient temperature are

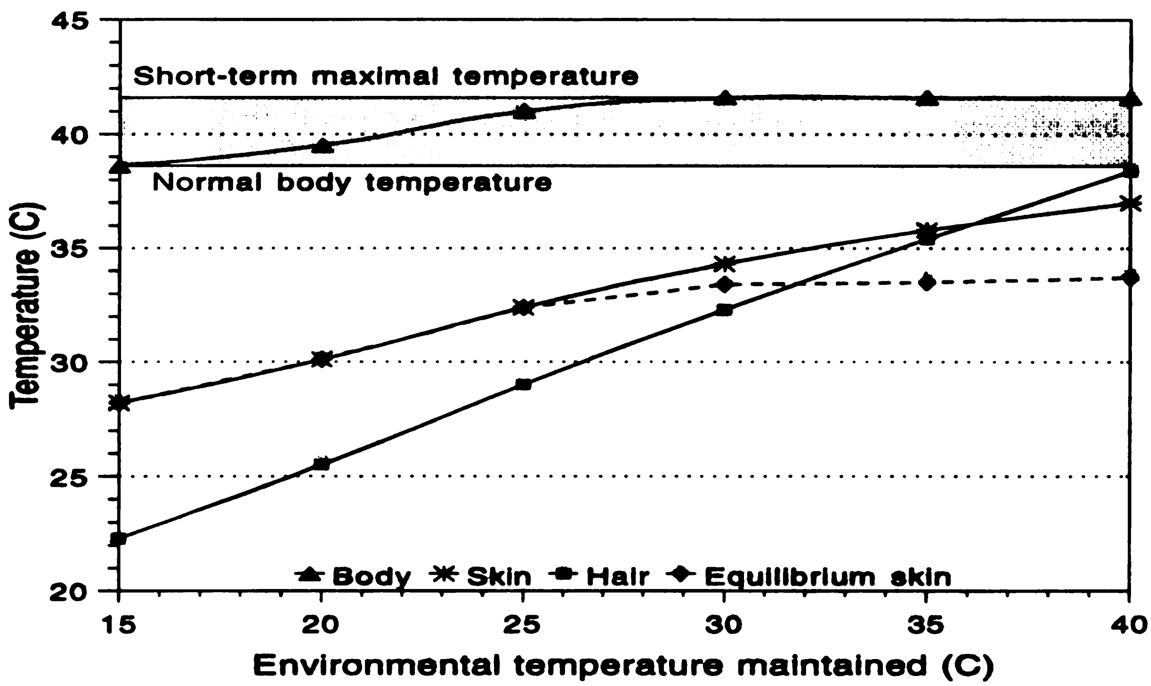


Figure 6.1.4 Modeled cow body core and surface temperatures within a simulated environment with a 2 m/s air velocity vs. temporary steady-state environmental temperature maintained (RH ~ 65%).

evident as well. The vaporization of sweat keeps the skin surface cooler than the surrounding body mass for all conditions and even keeps the skin cooler than the haircoat at the most oppressive temperatures. The hair coat serves both a positive and negative role when it becomes warmer than the skin. It protects the skin from the direct action of radiant and convective heating on the one hand, but it also limits moisture loss as is detailed later.

The increase in air velocity from 0.2 to 2 m/s shifted the modeled surface temperature curves to the right which results in a cooler surface temperature at the higher air speed for any given environmental temperature. The model results suggest that, when greater air movement exists in the animal area, a satisfactory temperature gradient between the body core and skin surface can be more readily maintained, reducing the need for body temperature to rise as environmental temperature increases.

Heat loss. Figures 6.1.5 through 6.1.8 show the modeled cows' sensible and latent heat losses in comparison to the thermoneutral-state metabolic heat production rate for the two air speeds and for either unacclimated or acclimated conditions. The reader is reminded that real cows would naturally take actions to acclimate to their environment if such actions were required to maintain a thermal energy balance. Heat losses were modeled and are shown for the unacclimated condition only to gain an appreciation for the positive effects of acclimation actions, especially increased respiratory activity, on the cow's ability to approach a thermal energy balance.

In these figures, latent heat losses clearly become a larger share of the total heat lost by the animals in hot weather. The relative magnitude of the latent heat loss varied little with ambient temperature, however. Because less heat is lost by sensible means under warmer ambient conditions and latent heat losses remain relatively constant, the modeled cows experienced a thermal energy imbalance at the higher ambient temperatures. This state of imbalance would require that measures be taken to restore the thermal energy balance.

Comparison of Figure 6.1.5 with Figure 6.1.6 and Figure 6.1.7 with Figure 6.1.8 shows that the modeled acclimation responses of the cows achieve an energy balance for a substantially greater range of environmental temperatures. In the low air velocity environment, the cows would not have been able to maintain a constant body temperature even at 15 °C were respiratory activity not increased and body temperature not allowed to rise some amount.

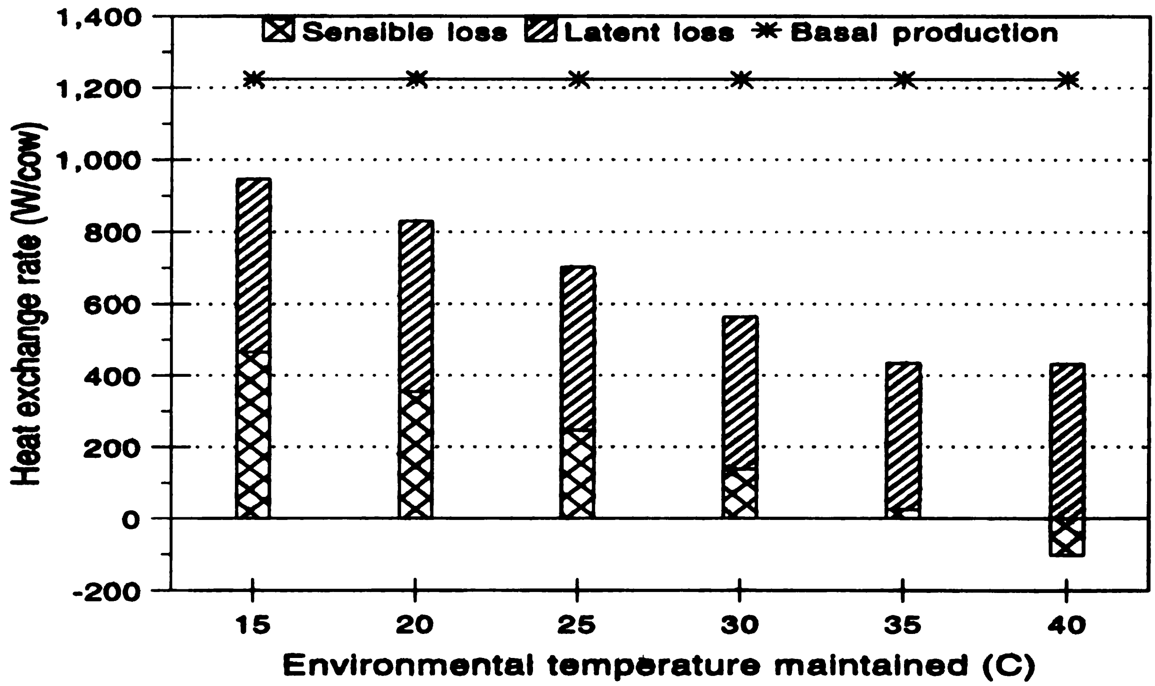


Figure 6.1.5 Modeled heat loss of a row of cows within a simulated environment with a 0.2 m/s air velocity vs. steady-state environmental temperature maintained (RH ~ 65%) if acclimation is inhibited.

In the environmental chamber study, the cows were reported to have expanded thermoneutral zones when supplied with good air movement during hot conditions. This effect was simulated by the model for the step increase from 0.2 to 2 m/s when Figures 6.1.5 and 6.1.7 or Figures 6.1.6 and 6.1.8 are compared. The resulting increase in convective heat loss, including the convection of water vapor from the cow's body is effective in maintaining an energy balance to beyond 25 °C with better air movement.

The Holstein cows in the environmental chamber study were reported as having maximal vaporization rates at 35 °C of 500 g/hr and 260 g/hr from the skin and lungs, respectively, at low air velocity. By comparison, ANTRAN modeled maximal vapor

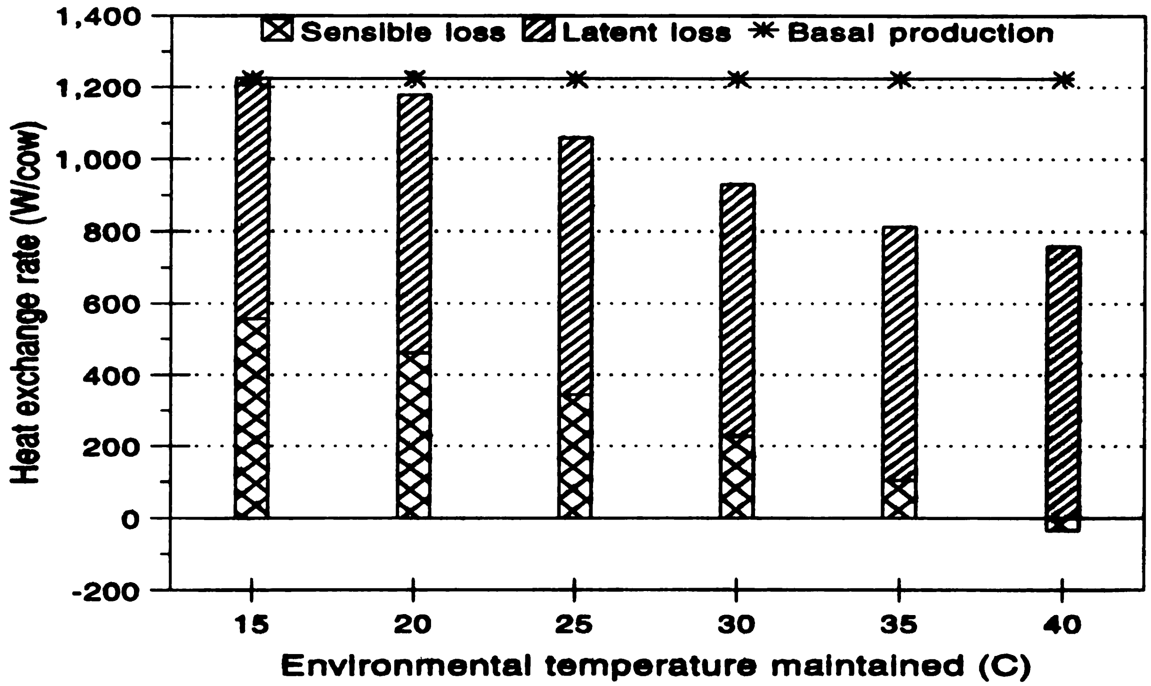


Figure 6.1.6 Modeled heat loss of a row of acclimated cows within a simulated environment with a 0.2 m/s air velocity vs. temporary steady-state environmental temperature maintained (RH ~ 65%).

loss rates of 620 g/hr and 440 g/hr. At 2 m/s air velocity, the rates from the environmental chamber study were about 350 g/hr and 150 g/hr compared to the modeled rates of 950 g/hr and 420 g/hr.

If the vaporization measurement techniques used in the environmental chamber study are assumed to be accurate, there is evidently some difference in the partitioning of heat loss or methodology taken by ANTRAN and that of real cows when managing thermal stress. No decline in feed intake, heat production or milk production is modeled in ANTRAN. Rather, the program predicts an impending decline whenever excess heat production exists. In the Missouri chamber study, the cows' feed intake did decline. Thus, much of the difference in moisture vaporization results may be

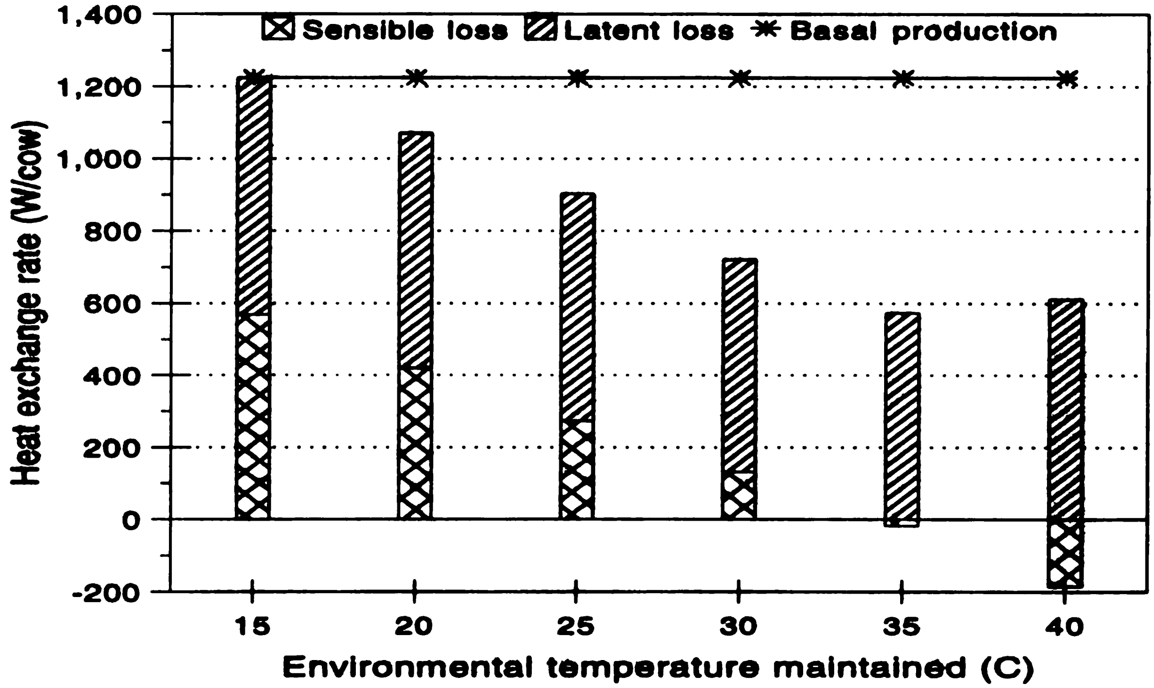


Figure 6.1.7 Modeled heat loss of a row of cows within a simulated environment with a 2 m/s air velocity vs. steady-state environmental temperature maintained (RH ~ 65%) if acclimation is inhibited.

attributed to the fact that data were gathered over several days under hot conditions, while ANTRAN models the cow's response to conditions over a shorter period of time and without an allowance for reduced feed intake. Also, the cows were reported to have lost weight during the course of the chamber study and it was presumed that body reserves were utilized to produce milk at a lower metabolic energy expenditure than if only consumed feed was used to produce the milk. More recent data (Monsanto, unpublished 1987 data) indicates that high producing cows exposed to a 30 °C environment may easily have vaporization rates from the skin and lungs that are higher than those produced by the model.

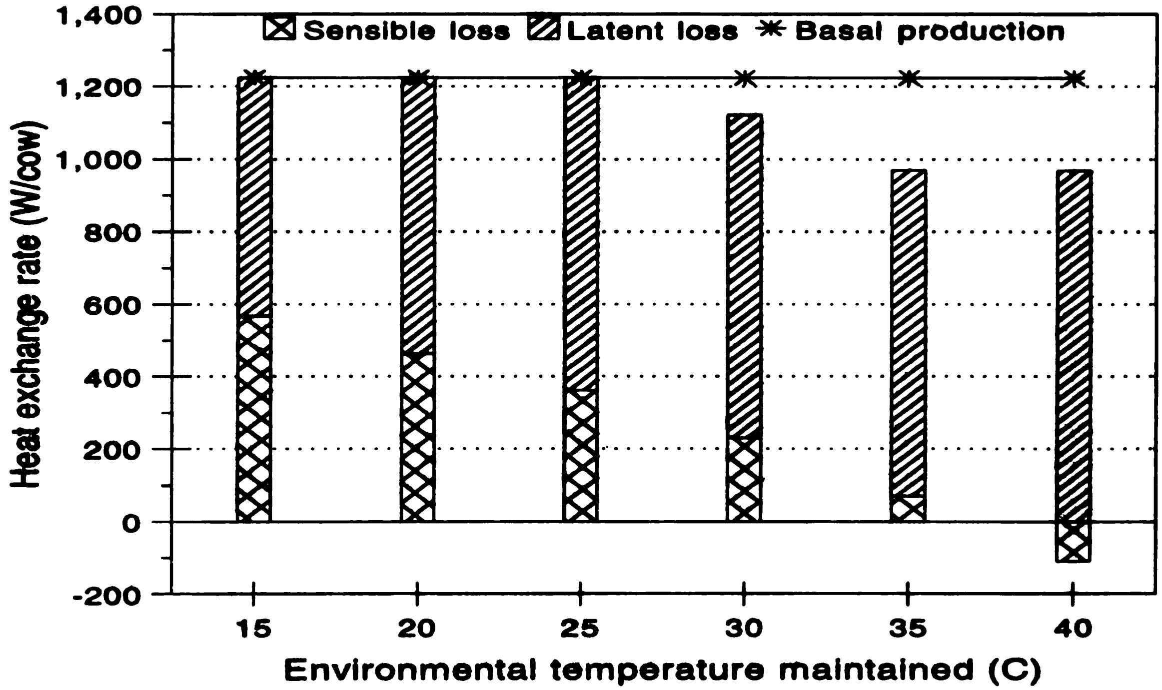


Figure 6.1.8 Modeled heat loss in a row of acclimated cows within a simulated environment with a 2 m/s air velocity vs. temporary steady-state environmental temperature maintained (RH ~ 65%).

The inherent differences between the assumptions incorporated into the model and the situation where cows are exposed to constant environmental temperatures for an extended period of time likely affect the results. Recall that ANTRAN allows for acclimation to temporary conditions. Temporary, in this case, means that adverse conditions may persist for a matter of hours before cooler conditions arrive. Experience in the field indicates that temporary acclimation often allows cows to endure such temporary conditions without decreasing feed intake or experiencing unmanageable body temperatures. In the environmental chamber studies, however, the cows were generally kept in a constant temperature environment for several days. The negative consequences to the cows when body temperatures and respiration rates are elevated for extended periods of time would soon preclude the continued dependency on these mechanisms as the only means of acclimation. The cows would, of necessity, resort to reductions in feed intake and metabolic heat production to prevent a potentially life threatening situation from developing.

6.2 Use of the Model in Predicting Heat Losses

ANTRAN was used to assess the impact of weather and naturally ventilated barn features on the ability of a row of cows to dissipate heat. The results are presented in stacked bar chart format in the following figures. Modeled absolute rates of heat loss are partitioned according to the primary heat transfer mechanisms involved: respiration, evaporation from the skin surface and radiant and convective heat losses from the hair coat surface. Also shown is any resulting excess heat load which exists when the rate of heat loss is less than the normal rate of metabolic heat production. This quantity is expressed as "heat loss needed" in the figures. The figures are used to draw implications about the effects of weather condition or barn design on the various heat transfer mechanisms and the overall thermal energy balance.

Effect of weather. As described in the methods section, model simulations were run and comparisons made for acclimatized Holstein cows, each producing 30 kg/day of milk, under two rather extreme summer, mid-day conditions in Lansing, MI. Figure 6.2.1 highlights the model results obtained for the reference or control barn for these two conditions. Weather conditions have considerable impact on the rate of heat loss available to the cows and thus on the excess heat load they experience.

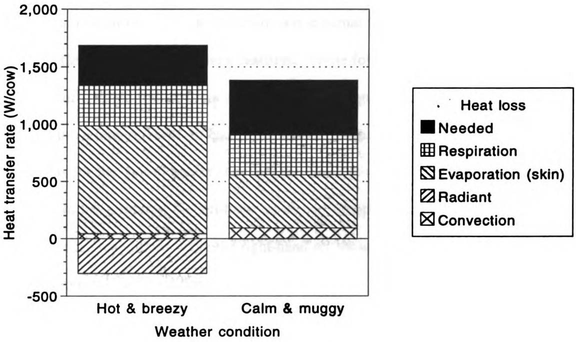


Figure 6.2.1 Magnitude of modeled heat loss via primary heat transfer mechanisms and the net heat loss available to cows in the control barn for two extreme mid-day weather conditions in Lansing, MI.

On the 'hot & breezy' day, substantial evaporative cooling potential existed to facilitate heat loss. Both the relatively high wind speed and the low humidity contributed to the large evaporative losses. The hot roof and diffuse radiation entering the sidewalls were significant heat loads on the cows, however. When the heat gained by radiant means was subtracted from the heat losses, each modeled cow still needed to dissipate 350 W of excess heat (represented as the solid-fill, "heat loss needed" area in the figures).

By comparison, the evaporative losses from the cows' body surface area on the 'calm & muggy' day were only about half those modeled on the 'hot & breezy' day.

As a result, the net heat load on each cow was about 500 W. This resulted even though the radiant heat load, as modeled, was minimal for the 'calm & muggy' day.

Effect of variation in barn features. In the following figures, the heat loss potential of a single row of cows in barns of varied construction is compared to that of cows in the reference barn. In each instance, the comparison is presented for mid-day conditions for the 'hot & breezy' day and the 'calm & muggy' day previously described. In these figures, the left-hand set of bars compares heat loss results for the 'hot & breezy' conditions while the right-hand set shows the same comparison for the 'calm & muggy' day conditions.

Barn width. Figure 6.2.2 compares heat loss results for two barns having widths of 18 m and 27 m, for the two mid-day weather conditions. In each barn, the row of cows was positioned one-third of the barn width from the windward sidewall and other barn features were unchanged. The gross impact of barn width on the energy balance of a single row of cows is minimal under these circumstances. There is a slight advantage to the narrow barn in terms of heat loss capacity still needed (10-20 W/cow). Most of the advantage in each case can be attributed to a small increase in evaporation from the skin surface. The velocity of air approaching a row of cows in the interior of a narrow barn would be expected to be slightly greater than that in a wide barn, other things being equal, because the air is not spread out over as wide a cross-sectional building area. Thus evaporation from the skin surface reflects that very small advantage.

During the 'calm & muggy' day, much of the narrow barn's advantage in evaporative capacity was offset by radiant heating effects. Due to the hazy sky and

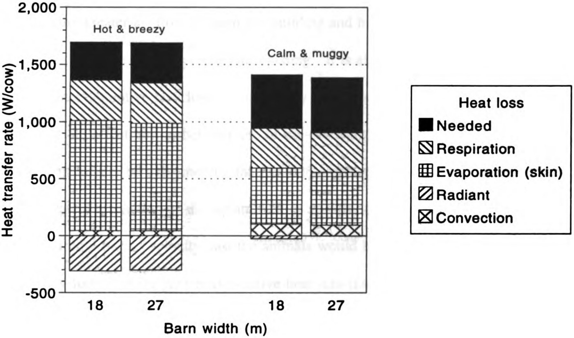


Figure 6.2.2 Effect of barn width on the modeled heat loss potential of a row of cows within the barn during mid-day of two extreme summertime weather conditions in Lansing, MI.

the more moderate dry-bulb temperature, the small mid-day radiant heat load on the row of cows, as modeled, was impacted more by exposure to diffuse solar radiation entering the sidewall than by the effect of the warm roof. The cows in a narrow barn have greater exposure to the open sidewall and thus are under a very slight disadvantage for radiant heat exchange under these conditions. In realistic situations, a greater advantage would be expected with the narrow barn since a wider barn would likely have more cows (rows of cows) per unit of building length and more heat to dissipate.

Sidewall open area. The effect of open area provided on heat loss and net heat load on the row of cows is shown in Figure 6.2.3. The most obvious change occurs in the modeled evaporation from the skin surface, with heat loss from that mechanism increasing for both weather conditions with increasing open area provided. This is a result of the greater air flow through the building and higher air velocities that result within the barn due to greater sidewall opening. It is also evident, however, that the increase in evaporative heat loss is not directly proportional to the open area provided, but rather the effect diminishes as more of the sidewall is opened up. The diminishing returns can largely be explained by the fact that convection of heat and mass are approximately a function of the square root of velocity (refer to Equation 3.6.7). Thus, doubling the air velocity past the animals would be expected to generate, at most, only forty percent greater convective heat loss (i.e. evaporation).

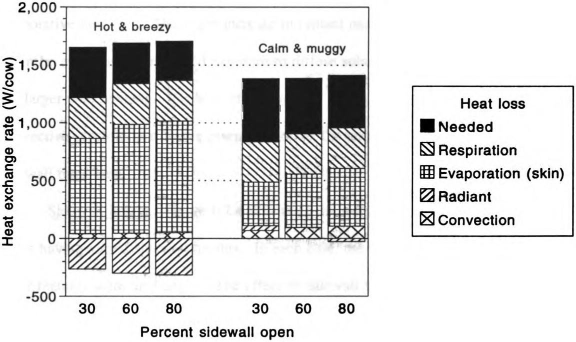


Figure 6.2.3 Effect of sidewall open area provided on the modeled heat loss potential of a row of cows within the barn during mid-day of two extreme summertime weather conditions in Lansing, MI.

Another major consideration is that the resistance to evaporation from the skin surface is affected by the hair coat in addition to the basic air conditions. As described in the review of literature, the resistance of the hair coat to diffusion of moisture is not significantly affected by the low to moderate air velocities that exist within naturally ventilated buildings. Therefore, the hair coat resistance remains largely unaffected by increased ventilation rate and actually becomes the increasingly larger resistor on a percentage basis in the overall mass transfer circuit.

Simultaneous increases in the convective heat loss and radiant heat load also resulted with greater sidewall open area, although to a much smaller extent. The explanation for increased convective heat loss is analogous to that just described for evaporative heat loss. The slight increase in radiant heat gain by the row of cows can again be attributed to increased exposure to diffuse solar radiation, in this case due to the larger wall opening. For both of the weather conditions modeled, the increased convective sensible heat losses essentially offset the added radiant heat load with more sidewall open area.

<u>Sidewall height.</u> Figure 6.2.4 illustrates the results of the modeling runs with barns having varied sidewall heights. In each case, the sidewalls were open 60% and other features were unchanged. The effect of sidewall height on the heat exchange potential of a single row of cows in the barn is very small, being in favor of the barns with taller sidewalls.

The benefit of greater wall height is less than that observed for the increase in sidewall open area percentage provided. With taller sidewalls, there is an increase in the building's ventilation rate. For example, the ventilation rate through the barn with

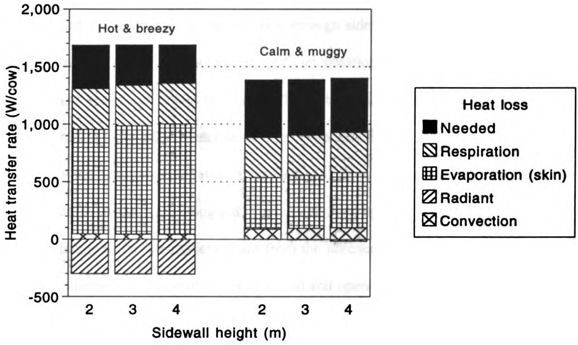


Figure 6.2.4 Effect of sidewall height on the modeled heat loss potential of a row of cows within the barn during mid-day of two extreme summertime weather conditions in Lansing, MI.

4 m sidewalls would essentially be twice that of the one with 2 m sidewalls if they both have the same percentage opening. However, since the cross-sectional area of the barn's interior also increases, the air velocity past the cows in the taller barn is not double that of the shorter barn. There is very minor impact from greater diffuse radiation entering the sidewalls. This is likely due to the fact that the row of cows is located in the interior of a fairly wide building (each barn is 27 m wide), so the view factor of the cow and open sidewall is not significantly increased with more wall opening. The results would likely change considerably if the row of cows was located closer to the sidewalls during the model simulations. The remaining explanations for the observed results are similar to those given for the discussion of open area effects.

Diffuse radiation through wall openings. ANTRAN was also used to simulate the effect of the incoming diffuse radiation through sidewall openings. In this case, the reference barn was compared to an artificial situation where the sidewalls are covered and yet an equivalent interior air flow rate is maintained (possibly via mechanical means). In these model results, the evaporation from the skin surface is essentially the same, as illustrated in Figure 6.2.5. This is expected since the velocity and properties of the air flowing past the row of cows in each case are the same. The small differences that are evident result from the absence or presence of diffuse radiation entering the sidewall for the covered and open sidewalls, respectively.

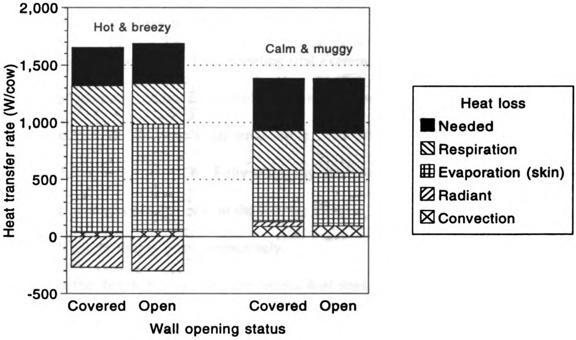


Figure 6.2.5 The effect of radiation entering the open barn sidewall on modeled heat loss potential of a row of cows during mid-day of two extreme summertime weather conditions in Lansing, MI.

For both weather conditions, having a closed sidewall provides the modeled cows a slight advantage in terms of radiant heat transfer. On the hot and breezy day, there is somewhat less radiant heat load on the animals, whereas on a calm and muggy day, a small amount of heat is lost to the building's shell that otherwise would be negated by the diffuse radiation entering the sidewall. Radiant and convective heat transfer from the cows depend on the hair coat surface temperature. With greater radiant heat load, the hair coat surface warms, and some of the benefits achieved by closing the sidewall are offset by the increased convection from cows with higher hair coat temperatures in the naturally ventilated barn. The net effect of eliminating the entry of diffuse radiation through the sidewalls when ventilation is maintained in the modeled barn is very small at best.

Effect of insulation. This modeling trial examined the effect of fully insulating the roof of the barn. In the fully insulated barn, the interior roof surface temperature was assumed to be the same as air temperature. The effect of insulation on the row of cows is shown in Figure 6.2.6. Fully insulating the barn roof reduced the need for additional heat loss of each cow in the row by 150 and 75 W on the 'hot & breezy' day and 'calm & muggy' day, respectively.

On the 'hot & breezy' day, the radiant heat load was modeled to be reduced by more than 200 W/cow. The net benefit derived from the insulation is less substantial, however, since what was a small sensible convective heat loss became an equivalent heat gain and a drop in evaporation from the skin occurred with a lower hair coat surface temperature. This particular case provides a good illustration of the strong interactions that exist between the heat transfer mechanisms and the potential for these

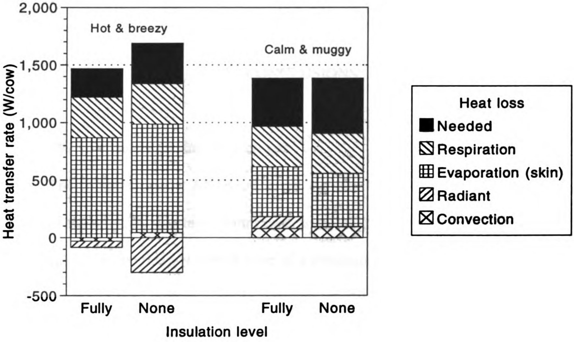


Figure 6.2.6 Modeled heat loss potential of a row of cows within fully insulated and uninsulated barns during mid-day of two extreme summertime weather conditions in Lansing, MI.

interactions to counteract each other. On the 'calm & muggy' day, a fully insulated roof provided an additional avenue for heat loss via radiant heat exchange with the roof due to its lower surface temperature. Again, the net result is a somewhat smaller benefit to the net heat load on the modeled animals.

7 - CONCLUSIONS

The conclusions of this research are:

1) The computer model ANTRAN, given the specific heat transfer characteristics of the animals, structural features and ambient summer weather conditions, calculates the thermal energy state of a continuous row of cows in a naturally ventilated barn with reasonable accuracy (modeled surface temperatures within 0.5 °C of those measured in an environmentally controlled chamber).

2) Partitioning of the potential for heat loss amongst the prevailing heat transfer mechanisms is an essential component of the model. It enables the user to infer from model results how the heat losses due to the different heat transfer mechanisms respond to changes in model input.

3) The impacts of varied weather conditions and certain primary barn design features on the heat exchange of dairy cattle in naturally ventilated barns were readily evaluated using the model. Specifically, when modeling a single row of cows within a barn:

    a) The deficit in heat loss when compared to heat production was greater for a 'calm & muggy' day compared to a 'hot & breezy' day due to the significantly lower evaporative heat loss available to the cows.

b) Increasing sidewall open area provided noticeably increased evaporative heat loss from the cows' outer body surface resulting in a lower net heat load.

c) Convective heat transfer is increased with improved levels of ventilation, but the increases are proportionately smaller than the increases in gross ventilation rate.

d) The effects of barn width, sidewall height and the entrance of diffuse radiation through open sidewalls on the net thermal energy balance of the row of cows were minimal.

e) The model projected that heavily insulating a barn greatly reduced the radiant heat load of the roof on the cows for the 'hot & breezy' conditions. However, the benefits to the cows of insulating the roof were minimal for the 'calm & muggy' day when the surplus of metabolic heat was greatest.

f) Interactions between the convective and radiant heat transfer mechanisms were observed, with a negative change resulting from one mechanism often counteracting a positive response from the other.

8 - LIMITATIONS ON TRANSFERRING MODEL RESULTS TO PRACTICE

Considerable care and discretion need to be taken when attempting to apply the results of this research directly to field situations. This section emphasizes some of the more important considerations and alerts the reader to some differences that might be expected in real field applications. The current version of the computer model ANTRAN was based on assumptions that are quite restrictive compared to the conditions that exist in real free stall barns. Especially important are the assumptions of steady-state conditions, two-dimensional mass and heat flows, absence of direct solar radiant heating of the animals, known animal characteristics, and only a single row of cattle.

a) Steady-state conditions, although commonly assumed in modeling work, do not accurately represent the processes that are modeled here which all are, in reality, quite dynamic in nature. Acclimation activities were especially difficult to describe in a steady-state model. Appropriate application of this model requires the selection of a time period that is short enough to be descriptive of real conditions and yet long enough to stabilize the nature of the processes involved. Evaluation on an hourly basis is probably acceptable.

b) Two-dimensional model heat and mass flows and the implications of using a two-dimensional model are presented in some detail in Chapter 4. Restrictions on allowable wind angle, building shape and cow positioning immediately preclude the direct application of the model to most practical situations.

c) Direct solar heating is not accounted for in ANTRAN. Direct solar heating of the cow would occur in most barns when the sun is low on the horizon. This limits the use of ANTRAN to mid-day and evening hours. In some practical situations, the effects of the afternoon sun may be of great interest.

d) Animal characteristics, both physical and physiological, are not well known and often vary considerably even among similar animals. Parameters that were used in this research to determine quantities such as hair coat resistance, respiratory ventilation rate, and the allowable temporary rise in body temperature were selected based on a very limited number of studies.

e) A single row of cows and its consequences were alluded to earlier in Chapter 4. The primary limitation of this assumption is that it doesn't allow evaluation of real multi-row barn situations. Situations that are thus not accounted for include comparisons of barns of varied stall arrangement or similar barns with differing cow stocking densities.

9 - RECOMMENDATIONS FOR FURTHER STUDY

Less restrictive assumptions would expand application of the model ANTRAN. Included would be using a three-dimensional model of the cows within the barn or of the barn itself, incorporating routines to handle direct solar heating of the animals, evaluating the cows' cumulative thermal status over greater portions of the day, providing a means of assessing multiple rows of animals, possibly utilizing Monte Carlo techniques for analyzing the radiant heat exchange and integrating feed intake and milk production models.

Several of the current component models could be improved. The equations used to determine total ventilation rate, convective heat transfer and respiratory ventilation rate, among others, are still quite crude.

Input data to the model that more accurately represent the modeled animal, particularly at currently observed levels of production, would improve model accuracy. Areas for additional research include maximal sweating and respiratory ventilation rates, hair coat properties and feed consumption characteristics when under thermal stress.

APPENDICES

APPENDIX A

TEMPERATURE HUMIDITY INDEX FOR DAIRY COWS

THI for Dairy Cows. Modified from Dr. Frank Wiersma (1990)
Ag. Engineering - The University of Arizona, Tucson Arizona

NO STRESS

MILD STRESS

MEDIUM STRESS

SEVERE STRESS

DEG C	RELATIVE HUMIDITY 0	5	10	15	20	25	30	35	40	45	50	56	60	65	70	75	80	85	90	95	100
22.2																				72	72
22.8																		72	72	73	73
23.3																72	72	73	73	74	74
23.9														72	72	73	73	74	74	75	75
24.4												72	72	73	73	74	74	75	75	76	76
25											72	72	73	73	74	74	75	75	76	76	77
25.6										72	73	73	74	74	75	75	76	76	77	77	78
26.1									72	73	73	74	74	75	76	76	77	77	78	78	79
26.7							72	72	73	73	74	74	75	76	76	77	78	78	79	79	80
27.2						72	72	73	73	74	74	75	76	77	77	78	78	79	80	80	81
27.8						72	73	73	74	75	75	76	77	77	78	79	79	80	81	81	82
28.3					72	73	73	74	75	75	76	77	78	78	79	80	80	81	82	82	83
28.9				72	73	73	74	75	75	76	77	78	78	79	80	80	81	82	83	83	84
29.4			72	72	73	74	75	75	76	77	78	78	79	80	81	81	82	83	84	84	85
30			72	73	74	74	75	76	77	78	78	79	80	81	81	82	83	84	84	85	86
30.6		72	73	73	74	75	76	77	77	78	79	80	81	81	82	83	84	85	85	86	87
31.1	72	72	73	74	75	76	76	77	78	79	80	81	81	82	83	84	85	86	86	87	88
31:7	72	73	74	75	75	76	77	78	79	80	80	81	82	83	84	85	86	86	87	88	89
32.2	72	73	74	75	76	77	78	79	79	80	81	82	83	84	85	86	86	87	88	89	90
32.8	73	74	75	76	76	77	78	79	80	81	82	83	84	85	86	86	87	88	89	90	91
33.3	73	74	75	76	77	78	79	80	81	82	83	84	85	85	86	87	88	89	90	91	92
33.9	74	75	76	77	78	79	80	80	81	82	83	84	85	86	87	88	89	90	91	92	93
34.4	74	75	76	77	78	79	80	81	82	83	84	85	86	87	88	89	90	91	92	93	94
35	75	76	77	78	79	80	81	82	83	84	85	86	87	88	89	90	91	92	93	94	95
35.6	75	76	77	78	79	80	81	82	83	85	86	87	88	89	90	91	92	93	94	95	96
36.1	76	77	78	79	80	81	82	83	84	85	86	87	88	89	91	92	93	94	95	96	97
36.7	76	77	78	79	80	82	83	84	84	86	87	88	89	90	91	93	94	95	96	97	98
37.2	76	78	79	80	81	82	83	84	85	87	88	88	90	91	92	93	94	96	97	98	99
37.8	77	78	79	80	82	83	84	85	86	87	88	90	91	92	93	94	95	97	98	99	
38.3	77	79	80	81	82	83	84	86	87	88	89	90	92	93	94	95	96	97	98		
38.9	78	79	80	81	83	84	85	86	87	89	90	91	92	94	95	96	97	98			
39.4	78	79	81	82	83	84	86	87	88	89	91	92	93	94	96	97					
40	79	80	81	82	84	85	86	88	89	90	91	93	94	95	96						
40.6	79	80	82	83	84	86	87	88	89	91	92	93	95	96	97						
41.1	80	81	82	84	85	86	88	89	90	91	93	94	95	97	98						
41.7	80	81	83	84	85	87	88	89	91	92	94	95	96	98							
42.2	81	82	83	85	86	87	89	90	92	93	94	96	97								
42.8	81	82	84	85	87	88	89	91	92	94	95	96	98								
43.3	81	83	84	86	87	89	90	91	93	94	96	97									
43.9	82	83	85	86	88	89	91	92	94	95	96	98									
44.4	82	84	85	87	88	90	91	93	94	96	97										
45	83	84	86	87	89	90	92	93	95	96	98										
45.6	83	85	86	88	89	91	92	94	96	97											
46.1	84	85	87	88	90	91	93	95	96	98											
46.7	84	86	87	89	90	92	94	95	97												
47.2	85	86	88	89	91	93	94	96	98												
47.8	85	87	88	90	92	93	95	97													
48.3	85	87	89	90	92	94	96	97													
48.9	86	88	89	91	93	94	96	98													
49.4	86	88	90	92	93	95	97														
50	87	89	90	92	94	96	97														
50.6	87	89	91	93	94	96	98														
51.1	88	90	91	93	95	97															
51.7	88	90	92	94	96	97															
52.2	89	90	92	94	96	98															
52.8	89	91	93	95	97																
53.3	90	91	93	95	97																
53.9	90	92	94	96	98																
54.4	90	92	94	96	98																
55	91	93	95	97																	
55.6	91	93	95	97																	
56.1	92																				

THI = (Dry-Bulb Temp. ° C)+(0.36 dew point Temp., ° C)+41.2

APPENDIX B

SAMPLE OUTPUT FROM THE COMPUTER MODEL ANTRAN

Title:	More open 80% sidewall on hot day		
Input file:	c:hotopen.dat	Output file:	c:hotopen.out

** BARN DATA **

Barn width:	27.0 m	Wall height:	3.00 m
Roof pitch:	4.0:12	Wall opening:	80.0 %
		Wall opening:	2.40 sq. m/m

INDEX	NODE DESCRIPTION	X-COORD	Y-COORD	SEGMENT	LENGTH
1	Base, inlet wall	.00	.00	Floor	27.00
2	Base, outlet wall	27.00	.00	Base-out	.30
3	Base of outlet	27.00	.30	Outlet	2.40
4	Top of outlet	27.00	2.70	Supt-out	.30
5	Top, outlet wall	27.00	3.00	Roof-out	14.23
6	Peak of roof	13.50	7.50	Roof-in	14.23
7	Top, inlet wall	.00	3.00	Supt-in	.30
8	Top of inlet	.00	2.70	Inlet	2.40
9	Base of inlet	.00	.30	Base-in	.30

** LIVESTOCK DATA **

Livestock:	dairy cattle		
Body mass:	635. kg	Production:	30.0 kg/day
Gestation:	120 days	Heat Prod.:	1386. W .
Body diameter:	.70 m	Surface area:	5.57 sq. m
Model length:	2.18 m	Exposed area:	4.80 sq. m
Basal Vent.:	100.0 L/min	Max. sweating:	.30 kg/sq. m/hr
Coat thick.:	1.4 mm	Coat Emis.:	.95

** ANIMAL ROW DATA **

ROW #	X-COORD	Y-COORD	LEFT PT	RIGHT PT	TOP PT	LOWER PT
1	9.00	1.00	8.65	9.35	1.35	.65

Floor area: 58.9 sq. m/animal

** AMBIENT AIR DATA **

Ambient Temp.:	36.1 C	Rel. humidity:	35.0 %
Atm. Pressure:	101.3 kPa		
Dew point T.:	18.1 C	Humid. ratio:	.0131 kg/kg
Vapor Pres.:	2.09 kPa	Air density:	1.13 kg/cu. m
Enthalpy:	70.0 kJ/kg	T.H.I.:	83.8

** AIR FLOW DATA **

Wind speed:	6.00 m/s	Wind angle:	0.Deg. from Sq.
Opening Cd:	.50	Vent. rate:	16. cu. m/s
Vel. on An.:	1.30 m/s	Vel. on roof:	1.37 m/s
Vent. Air T.:	.0 C	Vent. Air RH:	.0 %
Day of year:	176 th day	Local time:	13.0 mil. hr
Longitude:	84.6 deg.	Latitude:	42.8 deg.
Std meridian:	75.0 deg.		
Elevation:	70.2 deg.	Azimuth:	13.8 deg.
Direct SW:	848.0 W/sq. m	Diffuse SW:	142.0 W/sq. m
Atmos. LW:	10.0 W/sq. m	Barn angle:	.0 Deg. W of S
Ground albedo:	.25		

** SPECIAL FEATURES DATA **

Added water: 0. % of capacity

** TEMPERATURE RESULTS **

Body Temp.:	38.6 C	Eql. skin T.:	27.0 C
Skin Temp.:	33.8 C	Coat Surf. T.:	36.3 C

** HEAT TRANSFER RESULTS **

Rsp. latent Q:	88. W	Total Resp. Q:	92. W
Convective. Q:	9. W	Radiant Q:	-348. W
Coat. Cond. Q:	-338. W	Latent skin Q:	873. W
Total Q loss:	627. W	Pct. of Meta.:	45.2 %
Meta. surplus:	759. W	Pct. of Meta.:	45.2 %
Sensible Q:	-334. W	Share total:	-53.3 %
Latent Q:	961. W	Share total:	153.3 %
Share of evaporative loss:		Share of sensible loss:	
Respiration:	9.2 %	Respiration:	-1.0 %
Body surface:	90.8 %	Convection:	-2.7 %
		Radiant:	104.2 %

** MASS TRANSFER RESULTS **

Resp. Vent.:	84. L/min	Resp. Evap.:	.13 kg/hr
Surf. Evap.:	1.30 kg/hr	Surf. Evap.:	.27 kg/sq. m/hr
Reynolds #:	.54E+05	Nusselt #:	210.
Eff Thrm Cond:	.04 W/m/C	Conv HT Coef:	8.09 W/sq. m/C

** TEMPERATURE RESULTS **

Body Temp.:	41.6 C	Eql. skin T.:	32.3 C
Skin Temp.:	35.4 C	Coat Surf. T.:	37.4 C

** HEAT TRANSFER RESULTS **

Rsp. latent Q:	339. W	Total Resp. Q:	352. W
Convective. Q:	49. W	Radiant Q:	-317. W
Coat. Cond. Q:	-268. W	Latent skin Q:	966. W
Total Q loss:	1050. W	Pct. of Meta.:	75.7 %
Meta. surplus:	336. W	Pct. of Meta.:	75.7 %
Sensible Q:	-255. W	Share total:	-24.3 %
Latent Q:	1305. W	Share total:	124.3 %
Share of evaporative loss:		Share of sensible loss:	
Respiration:	26.0 %	Respiration:	-5.3 %
Body surface:	74.0 %	Convection:	-19.1 %
		Radiant:	124.3 %

** MASS TRANSFER RESULTS **

Resp. Vent.:	322. L/min	Resp. Evap.:	.51 kg/hr
Surf. Evap.:	1.49 kg/hr	Surf. Evap.:	.31 kg/sq. m/hr
Reynolds #:	.54E+05	Nusselt #:	210.
Eff Thrm Cond:	.04 W/m/C	Conv HT Coef:	8.09 W/sq. m/C

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