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Climate dependent heat stress mitigation modeling for dairy cattle housing

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Climate dependent heat stress mitigation modeling for dairy cattle housing

by

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TABLE OF CONTENTS

Page

LIST OF FIGURES

LIST OF TABLES

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NOMENCLATURE

A_{barn}	Average cross-sectional area of the barn (773.2 m^2)
A_{cow}	Area of the cow (5.37 m^2)
AOZ	Animal occupied zone
BGHI	Black globe humidity index
BT	Body temperature (°C)
BPM	Breaths per minute
cp_a	Specific heat of dry air (1006 J kg_a^{-1} K^{-1})
Cp_{body}	Specific heat of the cow's body (3400 J kg^{-1} K^{-1})
Cp_{DM}	Specific heat of dry matter (3250 J kg^{-1} K^{-1})
cp_w	Specific heat of water vapor (1850 J kg_w^{-1} K^{-1})
Cp_{water}	Specific heat of liquid water (4186 J kg_w^{-1} K^{-1})
cp_{mix}	Moist air mixture specific heat (1015 J kg_a^{-1} K^{-1})
CBAC	Compression-based air-conditioning
CCI	Comprehensive climate index (dimensionless)
CDH	Capacity to dissipate heat (dimensionless)
D_{cow}	Diameter of the cow (0.727 m)
d_{vapor}	Diffusivity of water vapor (0.0000285 m^2 s^{-1})
DMI	Dry matter intake (kg day^{-1})
E_f	Fan energy use (kW hr cow^{-1} yr^{-1})
E_{pr}	Energy price (\$ kW^{-1} hr^{-1})
EIT	Equivalent temperature index

$F_{cow,all,else}$	Shape factor of the cow to the remainder of the building less the roof
$F_{cow,roof}$	Shape factor for the cow to the roof (0.499)
g	Gravitational constant (9.81 m s^{-2})
GHI	Global horizontal irradiance (W m^{-2})
Gr	Grashof number (dimensionless)
h	Enthalpy of dry air (J kg_a^{-1})
H_{barn}	Average height of the barn (5.95 m)
$h_{cv,out}$	Exterior convective heat transfer coefficient (W m^{-2} K^{-1})
$h_{v,eq}$	Equivalent convective heat transfer coefficient (m s^{-1})
h_{fg}	Latent heat of vaporization (2410000 J kg_w^{-1})
$h_{fg,0}$	Latent heat of vaporization at 0^oC (2501000 J kg_w^{-1})
$h_{overall}$	Convective heat transfer coefficient (W m^{-2} K^{-1})
h_s	Adiabatic saturation of air (kJ kg_a^{-1})
h_w	Enthalpy of water vapor (kJ kg_w^{-1})
HT	Heat tolerance index
HTC	Heat tolerance test
HVC	High velocity case (2 m s^{-1})
HVCE	High velocity case (2 m s^{-1}) with evaporative cooling
HVCN	High velocity case (2 m s^{-1}) without evaporative cooling
L_{cow}	Length of the cow (1.98 m)
L_{coat}	Coat length (0.01 m)
Le_{coat}	Lewis number (dimensionless)
LCT	Lower critical temperature (°C)

LHP	Latent heat production (W)
LHP_{factor}	Latent heat production factor (1.4 W kg^{-1})
LHP_{max}	Maximum latent heat production (W)
LVC	Low velocity case (2°C Rule)
LVCE	Low velocity case (2°C Rule) with evaporative cooling
LVCN	Low velocity case (2°C Rule) without evaporative cooling
m_{actual}	Actual mass of animal (kg)
m_{cow}	Mass of modeled cow (600 kg)
m_{table}	Mass of comparison cow (500 kg)
M_{pr}	Milk price (\$ 100^{-1} lbs^{-1})
M_{tot}	Total milk weight (kg)
MET	Meteorological
MVC	Medium velocity case (1 m s^{-1})
MVCE	Medium velocity case (1 m s^{-1}) with evaporative cooling
MVCN	Medium velocity case (1 m s^{-1}) without evaporative cooling
N_{cows}	Number of cows
NRC	National Research Council
NSRDB	National solar radiation data base
Nu_{coat}	Nusselt number within the coat layer (dimensionless)
P	Atmospheric pressure (Pa)
$\Delta P_{a,mbar}$	Difference between the atmospheric and vapor pressure at the center of the barn (mbar)
P_{mbar}	Atmospheric pressure (mbar)

P_w	Partial pressure of water vapor (Pa)
$P_{w,coat,mabar}$	Average of the vapor pressure of the skin and the vapor pressure at the center of the barn (mbar)
$P_{w,lungs}$	Vapor pressure in the lungs (Pa)
$P_{w,mbar}$	Vapor pressure at the center of the barn (mbar)
$P_{w,skin}$	Vapor pressure at the skin level (Pa)
P_{ws}	Saturation water vapor pressure (Pa)
$P_{ws,air}$	Saturation vapor pressure of the air (Pa)
$P_{ws,body}$	Saturation vapor pressure of the body (Pa)
$P_{ws,skin}$	Full saturation vapor pressure at the skin level (Pa)
R^2	Coefficient of determination
R_a	Gas constant of dry air (287.1 J kg_a^{-1} K^{-1})
R_{bl}	Boundary layer resistance (m^2 K W^{-1})
$R_{material}$	Equivalent resistance of the wall (m^2 K W^{-1})
R_p	Pelage resistance (0.086 m^2 K W^{-1})
R_{roof}	Roof resistance (m^2 K W^{-1})
$R_{still,air}$	Still air resistance (0.12 m^2 K W^{-1})
$R_{t,actual}$	Tissue resistance (m^2 K W^{-1})
$R_{v,bl}$	Resistance to vapor flow through the boundary layer (s m^{-1})
R_{vc}	Resistance to vapor flow through the coat (s m^{-1})
$R_{vc,bl}$	Resistance to vapor flow through the boundary layer of the coat (s m^{-1})
$R_{v,eq}$	Equivalent resistance to vapor flow through the coat (s m^{-1})

R_w	Gas constant of water vapor (461.5 J kg_w^{-1} K^{-1})
R_{walls}	Wall resistance (m^2 K W^{-1})
rad	Radiation (W m^{-2})
rh	Relative humidity (%)
RMSE	Root mean square error
ROI	Return on Investment (\$ yr^{-1})
RR	Respiration rate (BPM)
RR_{max}	Maximum respiration rate (120 BPM)
RT	Rectal temperature (°C)
Sc_{coat}	Schmidt number (dimensionless)
Sh	Sherwood number (dimensionless)
SHP	Sensible heat production (W)
SHP_m	Sensible heat production produced per meter (W m^{-2})
SR	Sweating rate (g_w m^{-2} h^{-1})
SR_{max}	Maximum sweating rate (288 g_w m^{-2} h^{-1})
$T_{ave,cv}$	Weighted sum of the skin and surrounding air temperature (°C)
$T_{ave,rad}$	Weighted sum of the skin and surrounding air temperature (°C)
T_{bg}	Black globe temperature (°C)
T_{coat}	Coat temperature (°C)
$T_{coat,K}$	Coat temperature (K)
T_{core}	Core body temperature of cow (°C)
$T_{core,current}$	Current core temperature of cow entering the current hour of analysis (°C)

$T_{core,new}$	Updated core temperature after 1-hr (°C)
T_{db}	Dry-bulb temperature (°C)
$T_{db,center}$	Dry-bulb temperature at the center of the barn (°C)
$T_{db,evap}$	Dry-bulb temperature after passing through evaporative cooler (°C)
T_{dp}	Dew point temperature (°C)
T_{DM}	Dry matter temperature (°C)
T_{MRT}	Mean radiant temperature (K)
T_{roof}	Roof temperature (K)
T_{sa}	Sol-air temperature (°C)
$T_{surface}$	Surface temperature (°C)
T_{skin}	Skin temperature of cow (K)
T_{va}	Virtual air temperature (K)
T_{vb}	Virtual body temperature (K)
T_{wb}	Wet-bulb temperature (°C)
T_{walls}	Wall temperature (K)
T_{water}	Water temperature entering the cow (°C)
TCF	Time conversion from days to seconds $(24 * 3600 \text{ s day}^{-1})^{-1}$
TF	Time conversion from hour to seconds (3600 s hr^{-1})
THI	Temperature humidity index
THI_{adj}	Adjusted temperature humidity index
THP	Total heat production
TMY	Typical meteorological year
TNZ	Thermoneutral zone

Totcld	Total sky cover (0-1, tenths)
TV	Tidal volume (L $breath^{-1}$)
TV_{max}	Maximum tidal volume (4.24 L $breath^{-1}$)
UC_w	Weight unit conversion (2.2 lbs kg^{-1})
UCT	Upper critical temperature (°C)
UF	Unit conversions from m^3 s^{-1} to gal hr^{-1} (3.28^3 ft^3 m^{-3} * 7.5 gal ft^{-3} *3600 s hr^{-1})
U.S.	United States
VBA	Visual basic for applications
VHVC	Very high velocity case (3 m s^{-1})
VHVCE	Very high velocity case (3 m s^{-1}) with evaporative cooling
VHVCN	Very high velocity case (3 m s^{-1}) without evaporative cooling
W	Humidity ratio (kg_w kg_a^{-1})
W_{barn}	Width of barn (129.9 m)
W_{center}	Humidity ratio at the center of the barn (kg_w kg_a^{-1})
W_{cool}	Cooling water use (gallons cow^{-1} yr^{-1})
W_{lungs}	Humidity ratio of air from the lungs (kg_w kg_a^{-1})
W_{pr}	Water price (\$ $gallon^{-1}$)
W_s	Humidity ratio at saturation (kg_w kg_a^{-1})
W_{skin}	Humidity ratio at the skin surface (kg_w kg_a^{-1})
WI	Water Intake (kg day^{-1})
ws	Wind speed (m s^{-1})
yr	Year

α	Absorptivity of the roof (0.39 dimensionless)
β	Sweating factor (0.5534 dimensionless)
ΔL_{coat}	Coat length decrease due to airspeed over coat (m)
ε_{barn}	Emissivity of the barn surfaces (0.92 dimensionless)
ε_{cow}	Emissivity of the cow (0.98 dimensionless)
ε_{roof}	Emissivity of the roof (0.90 dimensionless)
$\dot{m}$	Mass flow rate ($kg_a\ s^{-1}$)
$\dot{m}_{wp,m}$	Mass flow rate of moisture production per meter ($kg_w\ s^{-1}\ m^{-1}$)
η_{evap}	Evaporative cooler efficiency (70%)
Θ_{roof}	Tilt angle of the roof (4.76°)
ν	Specific volume ($m^3\ kg^{-1}$)
ν_a	Specific volume of dry air ($m^3\ kg_a^{-1}$)
ν_{center}	Specific volume at the center of the barn ($m^3\ kg^{-1}$)
ν_{evap}	Specific volume of air after evaporative cooler ($m^3\ kg^{-1}$)
ν_w	Specific volume of moist air ($m^3\ kg_w^{-1}$)
π	Pi (3.14159)
ρ_{water}	Density of water (997 kg m^{-3})
σ	Stefan Boltzmann Constant ($5.67*10^{-8}$ W m^{-2} K^{-4})
$\dot{Q}_{cv}$	Energy loss due to convection (W)
$\dot{Q}_{FW}$	Energy loss due to feed and water intake (W)
$\dot{Q}_{evap}$	Energy loss due to evaporation (W)
$\dot{Q}_{heat,remain}$	Energy remaining to be dissipated (W)

$\dot{Q}_{latent,resp}$	Latent energy loss due to respiration (W)
$\dot{Q}_{rad}$	Energy loss due to long-wave radiation (W)
$\dot{Q}_{resp}$	Energy loss due to respiration (W)
$\dot{Q}_{RR,loss,max}$	Heat loss at the maximum respiration rate (W)
$\dot{Q}_{sensible,resp}$	Sensible energy loss due to respiration (W)
$\dot{Q}_{SR,loss,max}$	Heat loss at the maximum sweating rate (W)
$\dot{Q}_{stored}$	Stored energy (W)
τ	Inverse of the film temperature (K^{-1})
ϕ	Cloudiness factor (0-10, dimensionless)
ω	Relative humidity (decimal form)
$\dot{V}$	Ventilation rate ($m^3\ s^{-1}$)
$\vec{V}$	Velocity ($m\ s^{-1}$)

ABSTRACT

Production loss due to heat stress is a major concern in the livestock industry. Dairy cattle are especially susceptible to heat stress. Milk production loss due to heat stress accounted for about 1.2 billion dollars in 2010 (Lundeen, 2014). Heat stress occurs when combinations of environmental parameters (temperature, airspeed, relative humidity, etc) reach levels where the animal struggles to release internally produced heat. Implementing mitigation strategies to reduce heat stress has been a crucial need as dairy housing has transitioned from pasture to indoor housing systems.

Currently, limited recommendations exist directing producers towards implementing one cooling system over another, specific to the climate in their region. In order to maximize production, producers need the most optimal cooling system in their operation in order to reduce heat stress. To assist producers in heat stress mitigation decision making, a thermal interaction model was developed to quantify heat dissipation from a dairy cow's core to her surrounding environment using procedures and parameters from published thermal balance models. Environmental input parameters used for the model were taken from typical meteorological year (TMY3) data sets.

The objectives of this research were to: (i) analyze the thermal environment's ability to reduce heat stress in dairy cattle in selected regions using TMY3 data, (ii) model Holstein cattle subjected to evaporative cooling + airspeed, and direct water sprinkling + airspeed cooling systems by region, (iii) create a universal barn/cooling system model to apply to selected regions with given TMY3 data inputs, and (iv) develop contour maps with optimal cooling system recommendations throughout the United States.

A thermoregulation model was developed combining equations from previous models to analyze a cow in her ambient environment in a barn to determine if she is heat stressed using various cooling strategies. The model was tested in two stations in California (SN:723890) and Wisconsin (SN:726435). The model's predictions were within one standard deviation of field data. Once the model was validated, the model was ran for all 215 TMY3 Class 1 stations and contour maps of the U.S. were created for producers to determine which cooling strategy is the most economical in their region.

The results from this work show that CCI is a better thermal index than using THI and that the MVCE and LVCE cooling strategies are the most economical cooling strategy for dairy cattle housing when the milk price is fixed at \$0.363 kg^{-1}, water price is fixed at \$0.0015 $gallon^{-1}$ and energy cost is fixed at \$0.12 $kW\text{-}hr^{-1}$. Multiple maps were made for each of the eight cooling strategies that producers can use with a developed equation to determine which cooling strategy is the most economical for their region.

CHAPTER 1. INTRODUCTION

With the world's growing population, it is important that producers continue to increase their production of crops, meat, and/or milk to ensure that there will be enough nutritional food to feed everyone. Heat stress is detrimental to the livestock industry due to the decrease in production of meat and milk from the animal. Dairy cattle are especially susceptible to heat stress, and one of the major problems that the dairy industry faces is having cost-effective heat stress mitigation.

An animal becomes heat stressed when they are unable to dissipate internally generated heat to their surrounding environment. Animals housed in climates that have a high temperature profile or a high temperature profile combined with high relative humidity have higher heat stress risk. In dairy cattle, research suggests that heat stress occurs when the Temperature Humidity Index (THI) exceeds 68 (Collier, Zimbelman, Rhoads, Rhoads, & Baumgard, 2006). When a dairy cow undergoes heat stress, milk production and conception rates drastically decrease. The total annual economic loss in the United States (U.S.) due to heat stress in the dairy cattle industry is estimated at $897 to $1500 million (St-Pierre, Cobanov, & Schnitkey, 2003).

In order to reduce heat stress, cooling strategies have been implemented into indoor housing facilities. Small facilities may use natural ventilation with stir fans, but many modern facilities use forced mechanical ventilation, which may be combined with supplemental cooling systems like evaporative coolers or sprinkler systems. Although there are multiple methods of heat stress mitigation, it is unknown which cooling strategy is most effective and economical at reducing heat stress in any given climatic region.

With the varied climates throughout the U.S., weather data from specific locations throughout the U.S. was used for this project as a representation of the typical weather conditions for the majority of the dairies. For this project, typical meteorological year (TMY3) Class I data was used (Wilcox & Marion, 2008). The most updated and complete data sets are found under TMY3 Class I data; therefore, Class I data was used as the inputs in a dairy cow thermoregulation model to determine the most optimal cooling system to reduce dairy cattle heat stress. The developed model and procedures was first tested for a station in both California and Wisconsin, representing top producing states with vastly different climates. After the model and procedures were verified using a station in southern California (SN: 723890) and a station in northern Wisconsin (SN: 726435), all 215 TMY3 Class I stations throughout the U.S. were investigated to gather enough data with the ultimate objective to develop U.S.-wide contour maps directing dairy cooling strategies.

1.1 Objectives

The overall goal of this project was to determine which heat stress mitigation strategy has the greatest cooling potential and was the most economical for dairy cattle dependent on historical climatic conditions. The specific objectives of this research were to:

1) Analyze the thermal environment's ability to reduce heat stress in dairy cattle in selected regions with TMY3 Class I data, specifically highly populated dairy cattle areas such as California and Wisconsin,
2) Model Holstein cattle subjected to evaporative cooling + airspeed, and direct water sprinkling + airspeed cooling systems by U.S. region,

3) Create a universal barn/cooling system model to apply to selected regions with given TMY3 or equivalent data inputs, and,
4) Develop contour maps with optimal cooling system recommendations throughout the United States.

1.2 Thesis Organization

Chapter 1 provides a brief background and rationale for this project, Chapter 2 summarizes past research on dairy cattle heat stress and mitigation as well as past and current thermal indices and thermal models used to quantify heat stress, Chapter 3 outlines the development of the thermoregulatory model which includes procedures for calculating the barn environmental conditions using TMY3 data, the base barn model development, and the development of the cow model, Chapter 4 summarizes the evaluation of heat stress mitigation methods from the model for a station located in both California and Wisconsin, Chapter 5 investigates geospatial mitigation methods for all TMY3 Class 1 locations throughout the U.S., and finally, Chapter 6 provides project conclusions and suggestions for future work.

CHAPTER 2. LITERATURE REVIEW

In order to understand the importance of reducing dairy heat stress, literature was selected and reviewed focusing on (1) the economic impact of dairy heat stress, (2) the physiological impact of heat stress on dairy cattle, (3) models to predict dairy sensible and latent heat transfer, (4) experimentally measured dairy sensible and latent heat transfer for comparison with model predictions, (5) methods used to mitigate dairy heat stress, (6) indices used to assess heat stress, and (7) characteristics and use of Typical Meteorological Year (TMY) data.

2.1 Economic Impact of Dairy Heat Stress

Heat stress has a major economic impact in the dairy industry. Estimated annual losses in the United States total $897 million (St-Pierre et al., 2003). This loss is primarily due to the loss in milk production. With dairy operations spread throughout the US, the economic impact of heat stress varies with the location of the operations due to the climate variation throughout the country. California, Wisconsin, and Texas have the greatest combined economic losses from dairy cows and replacement heifers combined. California totaled about $125 million, Wisconsin lost about $61 million, and Texas had $132 million loss due to heat stress (USDA, 2016). Although Texas is ranked 6th in the nation as far as number of dairy cows, it has the highest economic loss due to heat stressed dairy cattle. This can be attributed to inventory size as well as the hot summer weather that Texas faces each year. California and Wisconsin experience high economic losses as well due to hot summers with higher humidity, further complicating a cow’s ability to dissipate body heat.

2.2 Physiological Impact of Heat Stress

During heat stress, many physiological changes occur that impacts a dairy cow. Table 2.1 shows multiple physiological responses during heat stress situations (Nickerson, 2014).

Table 2.1. Changes in animal physiology and other parameters due to heat stress (Nickerson, 2014)

Decrease in	Increase in
Dry matter intake	Weight loss
Rate of feed passage	Somatic cell counts
Blood flow to organs	Clinical mastitis
Rumen buffering capacity	Respiration rates
Milk yield and quality	Rectal temperature
Body condition score	Sweating
Heifer growth	Salivation
Immune function	Health care costs
Reproductive efficiency	Water intake

Understanding the thermoneutral zone (TNZ) is important to understand when a cow is in heat stress, prompting mitigation. Shown in Figure 2.1 (Kerr, 2015), the TNZ is a zone where dairy cows exhibit optimum performance and normal core body temperature (Kadzere, Murphy, Silanikove, & Maltz, 2002). Between the lower critical temperature (LCT) and upper critical temperature (UTC), dairy cows require no additional energy above maintenance to cool or heat their body (Avendano-Reyes, 2012). This zone depends on the age, breed, feed intake, diet composition, previous state of temperature acclimatization, production, housing, stall conditions, tissue insulation, coat insulation, and behavior of the animal (Yousef, 1985).

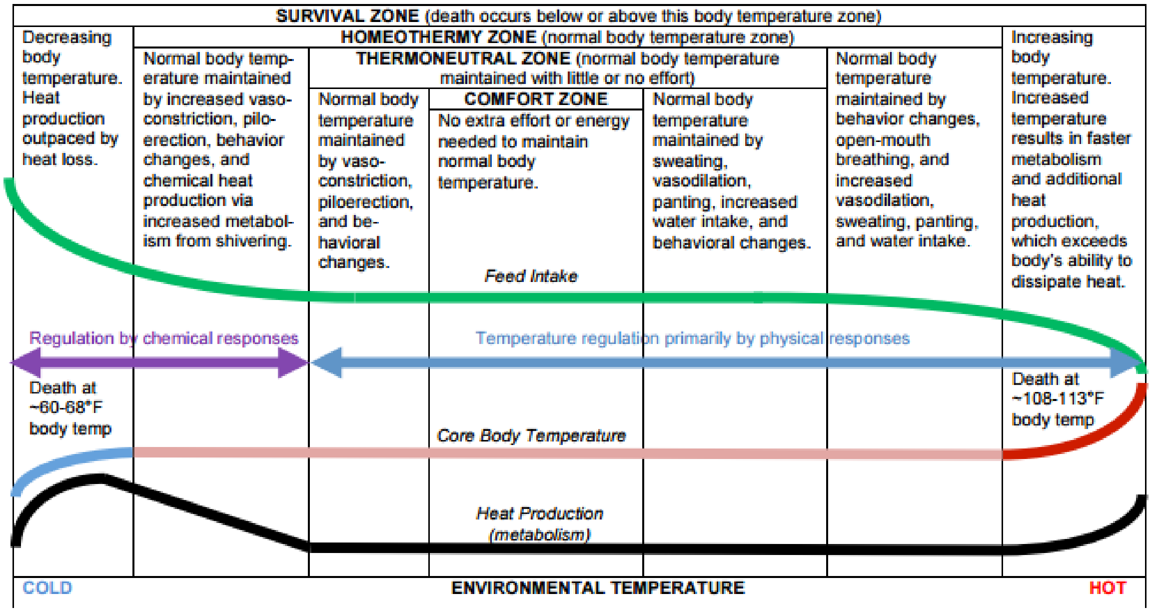


Figure 2.1. Thermoneutral zone for dairy cattle (Kerr, 2015)

When a dairy cow is heat stressed, she uses energy to try and cool herself instead of utilizing that energy for milk production or basic body maintenance. Figure 2.2 shows a schematic of energy use from feed energy in a dairy cow. It was taken from a similar diagram found in NRC 1981 (National Research Council (U.S.), Committee on Animal Nutrition, & Subcommittee on Environmental Stress, 1981). The last section that will receive excess energy is production. Heat stress greatly reduces milk production in dairy cattle, as well as decreases conception rates. These two areas negatively influence the overall profit for a producer.

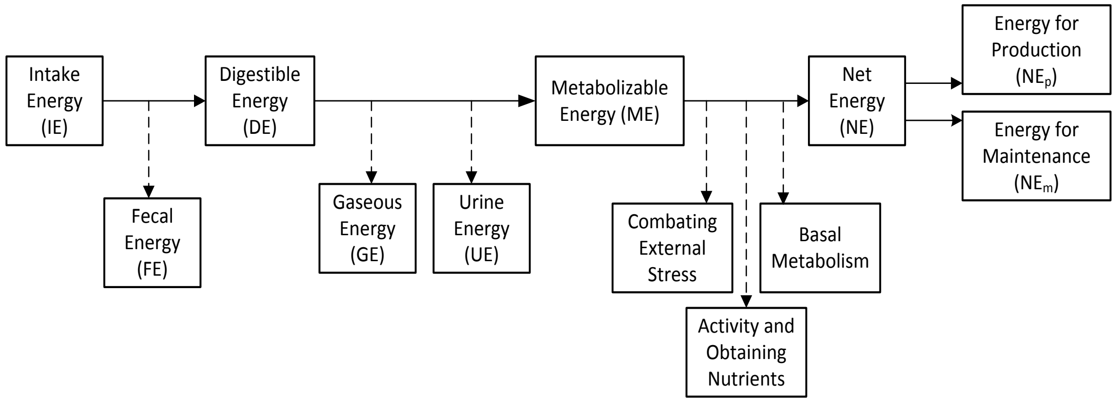


Figure 2.2. Schematic of the partitioning of feed energy within an animal

The biggest impact heat stress has on dairy cattle is on milk production. One study suggested that milk production is reduced 15%, accompanied by a 35% decrease in the efficiency of energy utilization for productive purposes when a lactating cow is transferred from an air temperature of 18 to 30°C (McDowell, Hooven, & Camoens, 1976). Also, cows are less able to cope with heat stress during early lactation (Kadzere et al., 2002). Milk production loss can typically be attributed to a decrease in dry matter intake. When a dairy cow’s body temperature increases due to heat stress, there is a decrease in feed intake. Decreased feed intake accounts for 50% of milk production loss while the other 50% loss is caused by the lack of lactose in the mammary gland due to the cow’s body diverting glucose away from the mammary gland (Bailey, Sheets, McClary, Smith, & Bridges, 2016). Table 2.2 shows the top 10 states with the highest milk production losses according to a study done by St-Pierre (2003) as well as dry matter intake, and Figure 2.3 shows a comparison amongst all of the lower 48 states for milk production losses. California is ranked #29, Wisconsin is ranked #34, and New York is ranked #42 when comparing milk production losses amongst all of the states.

Table 2.2. Estimated DMI reduction and annual production losses by dairy cows under minimum heat abatement intensity (St-Pierre et al., 2003)

State (Rank by number of milk cows)	Number of Milk Cows (1000 head)*	DMI Reduction (kg cow^{-1} yr^{-1})**	Milk Production Loss (kg cow^{-1} yr^{-1})
1. Louisiana (37)	14	1028	2072
2. Texas (6)	436	996	1803
3. Florida (18)	125	894	1803
4. Mississippi (41)	11	808	1629
5. Oklahoma (31)	39	737	1486
6. Alabama (43)	8	648	1305
7. Arkansas (44)	7	611	1233
8. Georgia (25)	83	600	1209
9. South Carolina (36)	15	484	975
10. Missouri (24)	88	464	936

*This data was taken from the USDA Summary (USDA, 2016)

**The DMI reduction is linearly correlated with the milk production losses

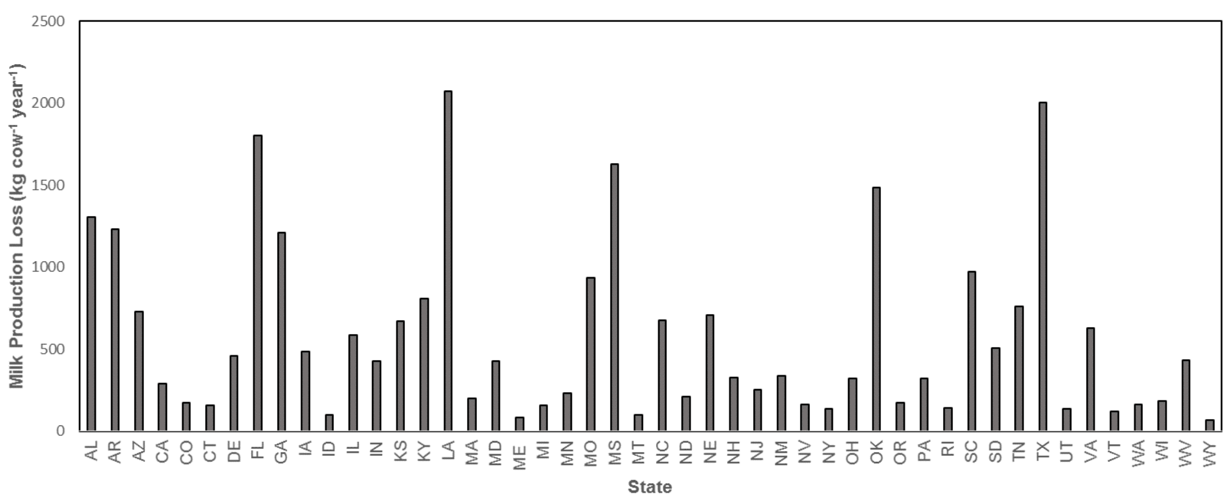


Figure 2.3. Milk Production loss due to heat stress in each state excluding Alaska and Hawaii (St-Pierre et al., 2003)

Additionally, when dairy cattle are heat stressed, pregnancy rates drop significantly. Cattle who are heat stressed are less likely to show signs of estrous. The embryo is extremely susceptible to heat stress. Embryonic death is caused by the cow's internal body temperature reaching over 39°C (102.2°F). This death primarily occurs within the first six days of embryonic development because the embryo hasn't developed the heat tolerant protein that protects it from the heat stressed environment in the uterus (Bailey et al., 2016). When the embryo dies, she is considered to be open since there is no embryo present in her uterus. Research shows that the negative effects of heat stress have been identified from 42 days before insemination to 40 days after insemination (Jordan, 2003). Follicular dynamics are altered by thermal stress, and oocyte quality is reduced for an extended interval after being exposed to heat stress conditions (Collier, Dahl, & VanBaale, 2006). Table 2.3 shows the top 10 states that have the highest increase in average days open due to heat stress as well as the number of heat stress hours per year (based on combinations of temperature and humidity) and the number of cattle deaths due to heat stress on a per 1000 cow basis.

Table 2.3. Estimated heat stress hours and deaths due to heat stress for dairy cows under minimum heat abatement intensity (St-Pierre et al., 2003)

State (Rank by number of milk cows)	Heat Stress (hr yr^{-1})	Deaths due to Heat Stress (per 1000 cows)	Increase in Average Days Open
1. Florida (18)	4261	17.2	59.2
2. Louisiana (37)	3551	19.3	57.7
3. Texas (6)	3185	15.9	53.9
4. Mississippi (41)	2993	13.6	47.0
5. Oklahoma (31)	2434	11.1	40.8
6. Alabama (43)	2679	10.4	40.5
7. Georgia (25)	2765	9.7	38.9
8. Arkansas (44)	2418	9.5	37.0
9. South Carolina (36)	2547	7.9	33.2
10. Missouri (24)	1875	6.7	29.0

Figure 2.4 shows a comparison amongst all of the lower 48 states for increase in average days open. California is ranked #28, Wisconsin is ranked #36, and New York is ranked #43 when comparing an increase in average days open amongst all states.

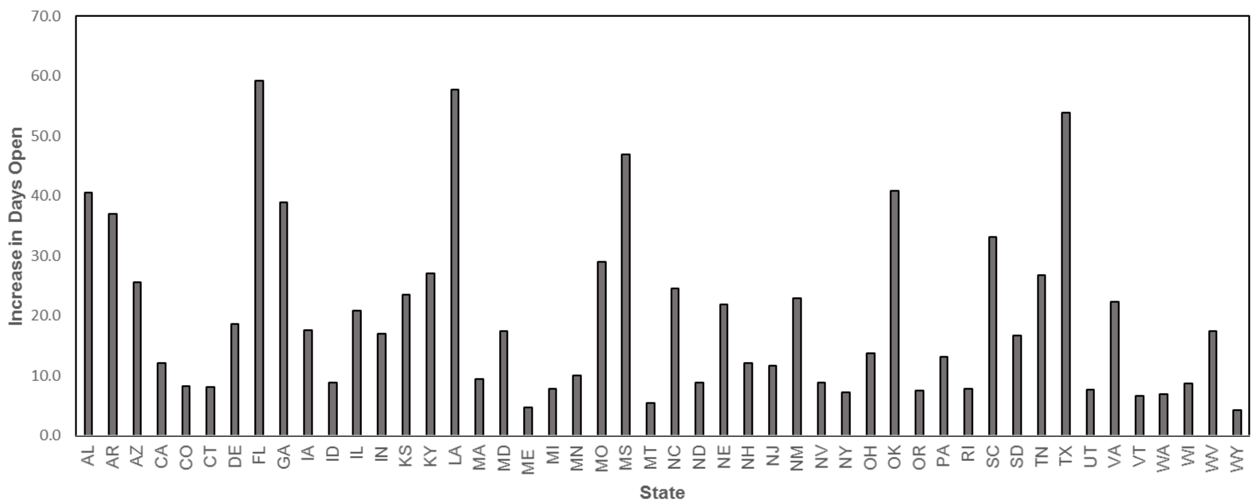


Figure 2.4. Increase in days open due to heat stress in each state excluding Alaska and Hawaii (St-Pierre et al., 2003)

Due to the significant drop in milk production and lowered conception rates as a result of heat stress, it is crucial that proper mitigation strategies are implemented in facilities so that producers can continue to profit in heat stress situations.

2.3 Models to Predict Dairy Sensible and Latent Heat Transfer

Dairy cattle dissipate heat to their environment through several means including conduction, convection, radiation, and evaporation. Conduction, convection, and radiation heat transfer are forms of sensible heat transfer because a temperature difference must occur between the cow and the environment in order for these three transfers to occur. Beneficial latent heat transfer (evaporation) results when liquid water changes phase to a vapor. For latent heat transfer to occur, a difference in vapor pressure between the cow's skin/respiratory tract and surrounding air must be present (Collier & Collier, 2012). Figure 2.5 shows a diagram of the variable means of heat transfer to and from a cow to her environment.

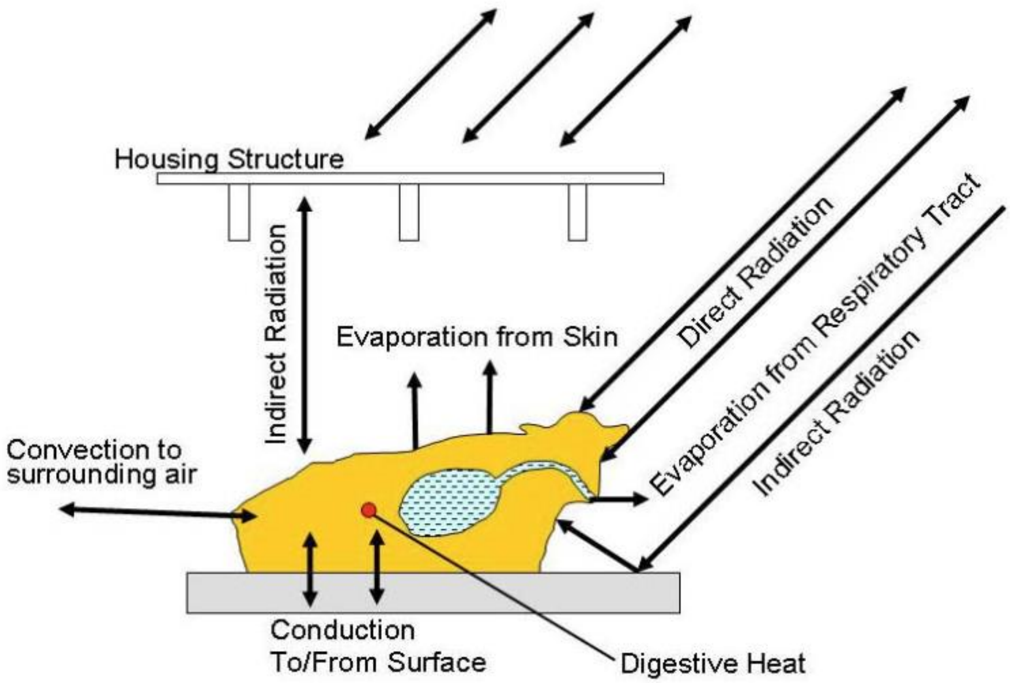


Figure 2.5. Energy exchanges between a dairy cow and its surroundings (Tyson, 2007).

During ambient temperatures under approximately 21°C, the primary methods that dairy cattle use to dissipate heat are conduction, convection, and radiation. At ambient temperatures above 21°C, the primary cooling method transitions to evaporation by increased sweating and respiration rates (Collier & Collier, 2012). When the air temperature is between 10 and 20°C, cutaneous evaporation accounts for about 20 to 30% of the total heat loss, and accounts for about 85% of the total heat loss when the air temperature is above 30°C with the rest lost by respiratory evaporation (Maia, daSilva, & Battiston Loureiro, 2005).

Several mathematical and mechanistic models have been developed to simulate the thermal interactions of dairy cattle and the environment. These models predict the outcome of the complex interactions of the multiple environmental and physiological variables on sensible and latent heat transfer. McArthur (1987) developed a comprehensive model describing the thermal interaction between an animal and its microclimate. Gebremedhin (1987) developed a model that predicted the sensible heat transfer across the boundary layer of animal hair coat as a function of hair coat properties and the thermal environment which aids in calculating conduction rates from the animal to the environment. McGovern & Bruce (2000) developed a model using Visual Basic for Applications (VBA) for modeling the thermal balance for cattle in hot conditions. This model includes heat loss by convection, radiation, and evaporation and allows the animal to vasodilate, sweat, pant, and store heat. Additionally, a model was developed that coupled heat and mass transfer to predict the evaporative and convective heat losses from a cylinder that simulates a full-sized cow (Gebremedhin & Wu, 2001). As cattle housing has moved towards mechanical ventilation systems, additional models were developed. One model was developed that predicts heat loss from cows in a ventilated space where the airflow in the space is modified by the cows themselves (Gebremedhin & Wu, 2003). Some models have focused on specific areas of energy exchange such as latent heat production by predicting sweating and respiration rates (Thompson, Fadel, & Sainz, 2011). Several heat transfer processes in animal coats were studied including radiative, conductive, and convective heat transfer (Cena & Monteith, 1975a, 1975b).

Models were further modified due to more in-depth research of cattle heat transfer and modern genetics. A model was developed that takes into consideration the breed of the animal but still follows similar procedures as McGovern & Bruce (2000) (Thompson, Barioni, et al., 2011). One of the most recently published models was developed as a dynamic model since variables such as body-core temperature and respiration rate lag behind changes in the environment (Thompson, Barioni, Rumsey, Fadel, & Sainz, 2014).

Finally, Nelson & Janni (2016) presented a model that follows procedures used by McGovern & Bruce (2000), Berman (2005), and McArthur (1987) and compares the results with actual measured data. Their model also introduced a new equation for determining tissue insulation tied to respiration rate.

There are multiple thermal balance models for livestock that have been developed, and models will continue to be developed as research advances. All of the models previously mentioned follow the same basic heat transfer principles, but as the years progressed, modified equations were developed to account for modern genetics and higher milk production which can impact, for example, the magnitude of heat generation and sweating rates.

2.4 Experimentally Measured Dairy Sensible and Latent Heat Transfer for Comparison with Model Predictions

There have been many studies conducted looking at how effective each method of cooling (conduction, convection, radiation, and evaporation) impacts heat stress in dairy cattle. Most newer housing construction methods takes into consideration all of these methods of cooling in order to reduce heat stress.

Many models developed neglect cooling due to conduction due to its complexity and there is too much variability with bedding type and contact surface area. There have been studies; however, analyzing cooling pads and mats and it’s capability of reducing dairy heat stress. One study looked at the effectiveness of conductive cooling by using continuously cooled waterbeds and found that the beds helped the cattle reduce their rectal temperature and respiration rates, increasing dry matter intake (DMI) and milk production (Perano, Usack, Angenent, & Gebremedhin, 2015).

Convective cooling greatly reduces heat stress in dairy cattle as well. One study compared cattle housed indoors with supplemental stir fans versus cattle with control-limited time inside the barn. The indoor housed cows with stir fans had lower respiration rates and higher DMI as compared to the control group that was allowed to go outside of the barn (Lin et al., 1998). Implementing ventilation to provide airspeeds throughout a facility up to about 3 m s^{-1} has a positive effect on cooling the animal with limited cooling benefit above 3 m s^{-1} (Gaughan et al., 2009).

One of the most economical ways of reducing dairy heat stress is to reduce the solar load, which is one reason why dairy cattle are housed indoors. It was estimated that total heat load is reduced by 30-50% with well-designed shade compared to solar exposure (Bond & Kelly, 1995). Also, another study showed that cows in a shaded environment had lower rectal temperatures, reduced respiratory rates, and yielded 10% more milk (Roman-Ponce, Thatcher, Buffington, Wilcox, & Van Horn, 1977).

As the temperature surrounding a cow increases towards its body temperature, sensible methods of transferring heat subside, reverting instead to latent cooling primarily through sweating and respiration. Increasing sweating rate is a natural physiological response cattle implement in response to a heat stress situation, but for this method to be effective, the surrounding environment must have the ability to evaporate sweat at the rate required. The environment must be able to handle the amount of sweat produced from a cow in order for sweating to be effective at cooling the cow. The cooling potential of sweating comes from the evaporation of water from her surface as well as using sprinklers to produce artificial sweat, which follows the same principle as sweating does in that the environment must be able to absorb excess water vapor to allow the evaporation process to take place.

There have been many studies looking at sweating rates and the effects of sprinkler systems. One researcher's goal was to update the data on sweating rates and compare the sweating rates amongst different breeds of dairy cattle as well as determine the level of influence that the environmental conditions and hair-coat color have on sweating rates (Gebremedhin et al., 2008). Later, a study was done on the physiological responses of dairy cows during extended solar exposure in hot and dry conditions versus hot and humid conditions. Sweating rates were measured from cows that were exposed to each condition, and it was concluded that sweating rates are higher in hot and dry conditions due to the higher moisture gradient between the skin surface and the ambient air (Gebremedhin, Lee, Hillman, & Collier, 2010). There was also a study done comparing sprinklers and misters implemented with fans with a special cooling evaporation system that decreased air temperature (Frazzi, Calamari, & Calegari, 2002). This study concluded that the cows located in the pens with fans and sprinklers and misters had lower rectal temperature, respiration rates, and lower reduction in milk yield compared to the control group, which had no cooling system. Table 2.4 summarizes the results of the studies previously mentioned. All results were collected from Holstein cattle of the *Bos taurus* species under shaded conditions.

Table 2.4. Field data comparison between multiple researchers

Study*	$T_{db,ave}$ (°C)	THI	Airspeed ($m\ s^{-1}$)	Evaporation ($g\ m^{-2}\ hr^{-1}$)	RR (BPM)	T_{core} (°C)	T_{rectal} (°C)	T_{skin} (°C)	T_{dorsal} (°C)
1	33	79	<0.12	350	-	-	39.3	-	35
2	45	89	<0.12	213	-	-	-	-	-
3	35.0	82.7	1	296	-	-	39.6	-	39.5
4	29.1	79.6	1	205.7	71.7	38.8	-	33.9	-
4	35.1	79.6	1	173.6	95.8	39.4	-	36.5	-

*1. Maia et al., 2005

2. McDowell, McDaniel, Barrada, & Lee, 1961

3. Kifle G. Gebremedhin et al., 2008

4. Kifle G. Gebremedhin et al., 2010

2.5 Methods Used to Mitigate Dairy Heat Stress

Numerous heat stress mitigation methods for dairy have been developed and researched. In addition to the methods previously mentioned such as cooling water-beds and solar shade, mechanical ventilation is used and stir fans are implemented throughout dairy facilities. Typically, dairy facilities have tunnel or cross-flow ventilation systems. Recently, combining mechanical ventilation with another means of cooling, such as evaporative coolers (indirect cooling) or sprinkler systems (direct cooling), has become an industry norm, especially for large scale operations. Many studies have looked at the effects of implementing evaporative coolers, sprinkler systems, or misters in large-scale mechanically ventilated dairies. One study concluded that systems that wet the animals leading to evaporation of water directly off of the animal's skin was more effective than an evaporation system that only cooled the air's dry-bulb temperature (Frazzi et al., 2002). Researchers have also looked at water-droplet size and its impact on increasing heat loss for an animal. Large droplets from a low-pressure sprinkler system that completely wets the cow by soaking through the hair coat to the skin is more effective than a misting system, but a combination of fans and misters was just as effective as fans and sprinklers at maintaining DMI and milk yield (Collier, Dahl, et al., 2006). Some research has also been done on comparing compression-based air-conditioning (CBAC) or zone cooling to other cooling methods (Brouk, Smith, & Harner, 2003). Implementing cooling strategies such as CBAC can become very costly, especially on large-scale operations. Although researchers have observed increased DMI and milk production coupled with decreased rectal temperature and respiration rates in CBAC environments (Brouk et al., 2003), this method is not economically feasible to implement in a dairy facility.

The overall building design and layout as well as implementing specific management practices can have an impact on dairy cattle heat stress. According to Collier & Collier (2012), abiding by the following guidelines can reduce heat stress in dairy facilities:

1. Improve water availability
2. Provide shade in the housing areas and holding pen
3. Reduce walking distance
4. Reduce time in the holding pen
5. Improve holding pen ventilation
6. Add holding pen cooling and exit lane cooling
7. Improve ventilation in cow housing areas (freestalls)
8. Cool close-up cows (three weeks prior to calving)
9. Cool fresh cows and early lactation cows
10. Cool mid and late lactation cows

Heat stress is detrimental to milk production, which is why there are so many strategies and methods attempted in the dairy industry to help cattle alleviate heat stress. Not only is it important to have good management practices, but incorporating a cooling system such as fans, sprinklers, evaporative coolers or a combination in a facility can greatly help reduce heat stress in dairy cattle.

2.6 Indices Used to Assess Heat Stress

Being able to properly assess the animal's environment is crucial in order to determine if the animal is experiencing heat stress. Several indices have been developed throughout the years, but the current generally accepted method used for assessing livestock heat stress is the Temperature Humidity Index (THI). This index includes inputs such as temperature and a measurement of humidity (relative humidity, wet bulb temperature, or dew point temperature) to assess the animal's thermal exchange with the environment. Current cooling standards utilize a THI threshold of 72 before the initiation of cooling, however recent research indicates that adverse effects of heat stress on dairy cattle can be shown as early as a THI of 68 (Collier, Zimbelman, et al., 2006). The results of this research showed that when dairy cattle are subjected to THI values of 68 or greater, a milk loss of 2.2 kg day^{-1} was measured. However, this index does not include critical inputs to assess heat stress such as solar load or airspeed (Hahn, Mader, & Eigenberg, 2003).

Before THI was developed, other indices were used to assess heat stress. One of the first indices developed for cattle was the Iberia heat tolerance test (HTC), which is given by Equation 2.1 (Rhoad, 1944):

$$HTC = 100 - [10 * (BT - 101.0)] \tag{2.1}$$

where,

BT = body temperature (°F).

The HTC assesses heat tolerance by measuring the amount by which rectal temperature exceeds the normal rectal temperature of 101.0°F (Collier & Collier, 2012) although the core body temperature is typically considered to be 38.5 °C (101.5°F). Through many other studies, it was concluded that HTC did not account for adaptation of heat tolerance in cattle, which suggested that a new heat index tool needed to be developed (Collier & Collier, 2012).

A decade later, an index called the heat tolerance index (HT) was developed based on respiration rate (RR) and body temperature (BT), which is shown by Equation 2.2 (Benezra, 1954):

$$HT = \left(\frac{BT}{38.33}\right) + \left(\frac{RR}{23}\right) \tag{2.2}$$

where,

BT = body temperature (°C)

RR = breaths per minute.

Respiration rate was included in this equation because it is important in controlling body temperature. Under ideal conditions, the body temperature of cattle, assumed with HT, is 38.33°C and the respiration rate is 23 breaths per minute, implying that an HT = 2.0 is an ideal non-thermally stressed animal (Benezra, 1954).

Finally, in 1962 the first applications of THI to livestock performance was reported although it had been used for human comfort for several years previous to its implementation in the livestock industry. A study looking at the effects of temperature and humidity on Holstein cows showed a strong relationship between temperature (°F) and humidity on milk production and cow comfort (Johnston, Ragsdale, Berry, & Shanklin, 1962). The governing relationship is shown in Equation 2.3 (Johnston et al., 1962), but was later refined (Equation 2.4) based on numerous experimental studies (Johnson, 1965):

$$THI = 0.4 * (T_{db} + T_{wb}) + 15 \quad (2.3)$$

where,

T_{db} = dry bulb temperature (°F)

T_{wb} = wet bulb temperature (°F)

$$THI = (T_{db} * 0.55) + \left(T_{dp} * 0.2\right) + 17.5 \quad (2.4)$$

where,

T_{dp} = dew point temperature (°F).

THI for dairy was revised again after further research had been conducted. This THI equation (Equation 2.5) can be utilized to define thresholds where the potential for dairy heat stress exists (Yousef, 1985):

$$THI = T_{db} + (0.36 * T_{dp}) + 41.2. \quad (2.5)$$

There have been multiple THI modifications; however, the one used in this current project follows Equation 2.6 (Hahn et al., 2003):

$$THI = 0.72 * (T_{db} + T_{wb}) + 40.6. \quad (2.6)$$

A similar equation as the THI was developed (Equation 2.7) to account for solar load, known as the black globe humidity index (BGHI), to further explain reductions in milk production (Buffington et al., 1981):

$$BGHI = T_{bg} + (0.36 * T_{dp}) + 41.5 \quad (2.7)$$

where,

T_{bg} = black globe temperature (°C).

The effects of ambient temperature, solar radiation, and convective cooling via elevated airspeed are all integrated into a temperature value for the black globe (Li, Gebremedhin, Lee, & Collier, 2009). The black globe temperature is obtained by using a matte black copper ball that has a diameter between 12.5 and 15.0 cm with a temperature sensor centered internally (Lee, 1953). During one study however, when calculating the black globe humidity index, only four out of the eight studies actually recorded the appropriate values in order to produce the BGHI, therefore there was little evidence that BGHI was superior to THI for estimating the threshold temperatures for milk yield loss (Collier, Zimbelman, et al., 2006).

Another index developed that takes into consideration airspeed along with dry-bulb temperature and relative humidity is the equivalent temperature index (EIT), which was derived from the analysis of milk production and heat-loss rates (Baeta, Meador, Shanklin, & Johnson, 1987). This index is shown in Equation 2.8:

$$\begin{aligned} EIT = 27.88 - 0.456 * T_{db} + 0.010754 * T_{db}^2 - 0.4905 * rh + 0.00088 * rh^2 \\ + 1.15 * ws - 0.12644 * ws^2 + 0.019876 * T_{db} * rh \\ - 0.046313 * T_{db} * ws \end{aligned} \tag{2.8}$$

where,

rh = relative humidity (%)

ws = wind speed (m s^{-1}).

EIT can be used for ambient temperatures between 16 and 41°C, relative humidity (rh) from 40 to 90%, and airspeed from 0.5 to 6.5 m s^{-1} (Baeta et al., 1987). Although the EIT equation is bounded by specific limits, all conditions that could lead to heat stress for dairy fall within these boundaries. The difficulty in using EIT is that no accepted thresholds exist from which to assess the severity of heat stress and therefore is of limited use.

Most recently, a comprehensive climate index (CCI), was developed that corrects ambient temperature based on relative humidity, wind speed, direct solar radiation, and ground surface radiation (Mader, Johnson, & Gaughan, 2010). The CCI index was developed for ambient temperature between -30 and 45°C. Equation 2.9 is the relative humidity correction factor, and Equation 2.10 is the wind speed correction factor:

$$rh_{CCI} = e^{(0.00182*rh+(1.8*10^{-5})*T_{db}*rh)} * ((5.4*10^{-5})*T_{db}^2 + 0.00192*T_{db} - 0.0246)*(rh-30) \tag{2.9}$$

and

$$ws_{CCI} = \left(\frac{-6.56}{e^{\left(\frac{1}{(2.26*ws+0.23)^{0.45*(2.9+(1.14*10^{-6})*ws^{2.5}-\log_{0.3}(2.26*ws+0.33)^{-2})}}\right)}}\right) - 0.00566 * ws^2 + 3.33. \tag{2.10}$$

Equation 2.11 is the direct solar radiation correction factor, and Equations 2.12 and 2.13 both are surface temperature correction factors; however, Equation 2.12 can be used if the surface temperature is unknown:

$$rad_{CCI,DR} = 0.0057*rad - (2*10^{-4})*rad*T_{db} + (5*10^{-5})*T_{db}^2*\sqrt{rad} \tag{2.11}$$

$$rad_{CCI,S} = 0.1*(T_{db} + 0.019*rad) - 2 \tag{2.12}$$

$$rad_{CCI,S} = 0.1*\left(T_{surface}\right) - 2 \tag{2.13}$$

where,

$T_{surface}$ = surface temperature (°C)

rad = radiation (W m^{-2}).

Finally, the CCI index can be calculated (Equation 2.14):

$$CCI = T_{db} + rh_{CCI} + ws_{CCI} + rad_{CCI}. \tag{2.14}$$

The model developed for the project; however, currently does not incorporate radiation due to the complexity of the equations.

Along with THI and CCI, respiration rates (RR) can also be an indicator of heat stress. Using respiration rates as a heat stress indicator is practical for producers and physically indicates the severity of cow heat stress. A producer may not know what the humidity in the barn is at that time or be able to calculate a CCI value, which is why using a cow's respiration rate is a fast and easy way to assess the level of heat stress. Table 2.5 shows a comparison of CCI, THI, and RR values associated with various categories of heat stress.

Table 2.5. Comparison of CCI, THI, and RR values

Environment	CCI*	THI**	RR (BPM)**
No Stress	< 25	< 68	< 60
Mild	25 – 30	68 – 71	61 – 75
Moderate	30 – 35	72 – 79	76 – 85
Severe	35 – 40	80 – 89	86 – 119
Extreme	40 – 45	90 – 99	120 – 140

*****Values taken from Mader et al. (2010)

******Values taken from Renaudeau et al. (2012)

Finally, a heat stress indicator proposed and developed for this research project to quantify heat stress is the capacity to dissipate heat (CDH). A cow's CDH, deviating from CDH=1.0 (Equation 2.15), occurs when heat is internally stored causing an increase in her core body temperature, a consequence of her inability to dissipate heat to her environment:

$$CDH = 1 - \frac{\dot{Q}_{storage}}{THP} \tag{2.15}$$

where,

$\dot{Q}_{storage}$ = amount of heat stored (W)

THP = total heat production (W).

A CDH of 1.00 indicates that she is technically within the thermoneutral zone and not heat stressed. If the CDH goes below one, the assumption of heat stress is made indicating the need to store generated heat to maintain thermal balance with the surrounding environment.

2.7 Characteristics and Use of Typical Meteorological Year Data

There are multiple weather stations located throughout the U.S. that collect weather data for that given area. There are many different weather data sets to choose from, but for this study, TMY data was used. A typical meteorological year (TMY) data set contains hourly meteorological values for a specific location over an extended period of time (Wilcox & Marion, 2008). Weather stations throughout the U.S. have TMY data, but for this project, TMY3 Class 1 data was used because it is the most recent and accurate data available. TMY3 data sets are derived from the 1961-1990 and 1991-2005 National Solar Radiation Data Base (NSRDB) archives and their intended use is for simulations of solar energy conversion and building systems to facilitate performance comparisons of different system types, configurations, and locations throughout the U.S. ("Typical Meterological Year 3," 2005). In addition, TMY3 data sets contain all of the required input data for this project including, but not limited to, hourly dry-bulb temperature, relative humidity, atmospheric pressure, wind speed and direction imposed on a building, and solar radiation. This data was used as inputs in the barn model in order to select the most optimal cooling system to reduce dairy heat stress for each selected region.

Little to no research has been done on determining dairy cooling systems based on TMY3 data. One study was conducted using TMY3 data to determine what retro-commissioning may do to save energy and associated costs of dairy housing fan operation (Brinker, Reinholtz, Williams, & Bergum, 2013). Another study was done in swine housing that consisted of developing a building thermal analysis and air quality model to predict indoor climate and long-term air quality (Sun & Hoff, 2009).

In order to select the weather stations with TMY3 data that were needed to verify the model's predictions, the states that contain the most dairy cattle were identified and then regions within those states with high dairy cattle density were selected. The top two states with the greatest dairy cattle population are California and Wisconsin ("U.S.," 2016). Weather stations that are located within these high dairy cattle density areas include Fresno Yosemite International Airport, CA (SM: 723890) and Eau Claire County Airport, WI (SN: 726435). Figure 2.6 (USDA, 2012) shows a map of the initial selected states and stations. These stations were used to test the model before it was applied to the rest of the stations throughout the country. Since the climates in CA and WI are very different, this comparison was a good assessment of various cooling systems and the developed models prediction and comparison of the cooling strategies evaluated for this project.

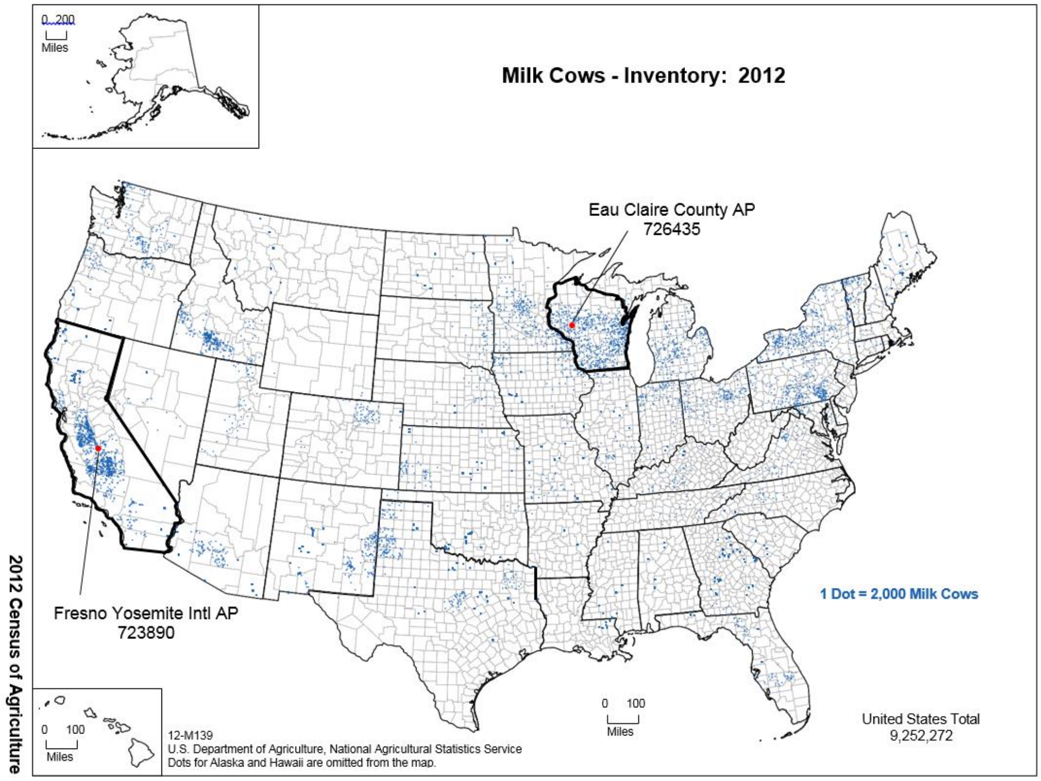


Figure 2.6. Map of selected states and TMY3 stations

In order to produce a contour plot of heat stress mitigation recommendations throughout the U.S., more TMY3 weather stations are required regardless of dairy density. Figure 2.7 shows a map of the country containing all TMY3 stations. The green stars designate Class I stations. Only the contiguous 48 states were used for this study.

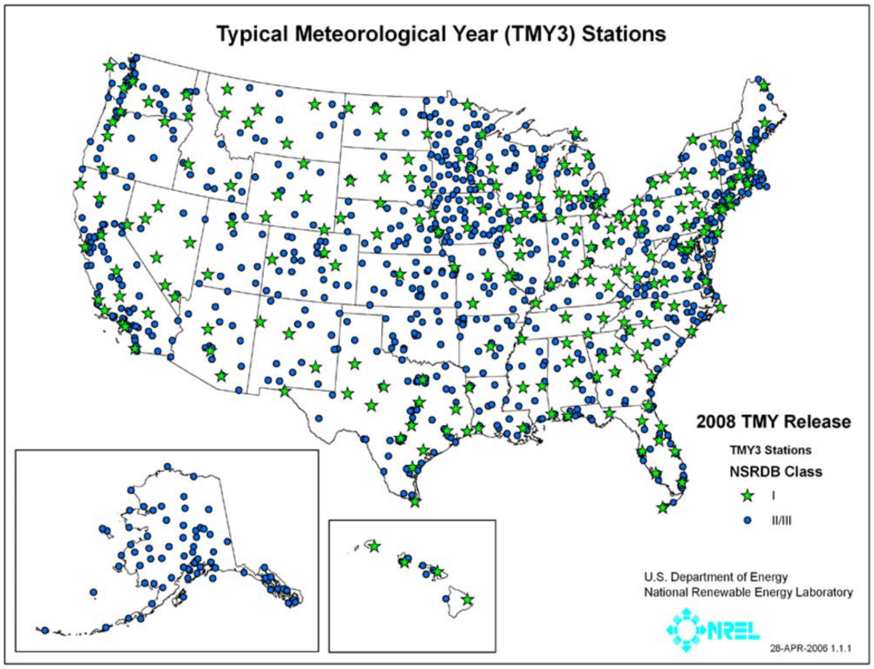


Figure 2.7. All TMY3 stations

Since only Class 1 stations are being used, the other stations were disregarded in order to develop enough spatial density to produce meaningful contour plots. Figure 2.8 shows a map of the existing 215 TMY3 Class I stations used for this project.

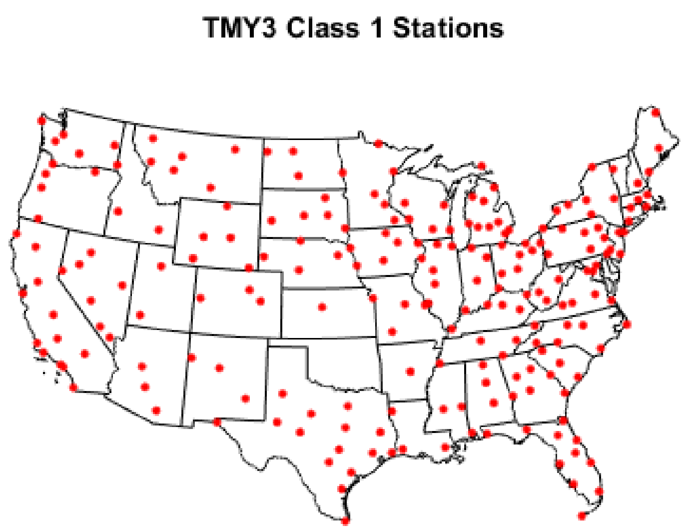


Figure 2.8. Class 1 stations

CHAPTER 3. MODEL DEVELOPMENT

The following chapter describes the methods and procedures taken in order to develop a thermoregulation model to determine if a cow is in heat stress. This chapter includes the barn model that was created with selected cooling systems as well as the cow model and results comparing the model's predictions with field data.

3.1 Typical Meteorological Year Data (TMY) to Psychrometric Properties

In order to properly analyze the thermal environment around the cow, psychrometric (i.e., dry-air and water vapor mixture) properties of the surrounding air need to be determined as well as the mean radiant temperature and airspeed of her surroundings. This information is required in order to model the expected sensible and latent heat released to her environment. In order to calculate the psychrometric properties of the ambient air and the mean radiant temperature of the animal's surroundings, variables outlined in Table 3.1 where used from each TMY3 Class I data set.

Table 3.1. Variables used from TMY3 Class I data set

Variable	Abbreviation or Symbol	Unit or Range	Description
Date	--	MM/DD/YYYY	
Time	--	Hour	
Global horizontal irradiance	GHI	W m^{-2}	Total amount of direct and diffuse solar radiation received on a horizontal surface during the 60-minute period ending at the timestamp
Dry bulb temperature	T_{db}	°C	
Relative Humidity	Φ	%	
Pressure	P	mbar	

Table 3.1 (cont.). Variables used from TMY3 Class I data set

Total sky cover	Totcld	Tenth	Amount of sky dome covered by clouds or obscuring phenomena
Wind speed	ws	m s^{-1}	

The dry-bulb temperature, relative humidity, and atmospheric pressure data from each TMY3 Class I data set were used to determine the remaining psychrometric properties using standard ideal gas thermodynamic relations (Appendix A).

3.2 The Base Barn Model

3.2.1 Barn parameters

A common barn configuration and herd size was chosen for analysis at each TMY3 location regardless of the barn style typical of each climatic region. This was done as a standard baseline condition to compare various mitigation strategies. According to Dairy Management Inc., farms with more than 100 cows produce 86% of the U.S. milk supply (USDA, 2012). The top two dairy producing states throughout the U.S. are California (#1) and Wisconsin (#2). The average sized dairy herd in California is 1438 cows, whereas, in Wisconsin the average herd size is only 129 cows. Since many dairies throughout the U.S. have a larger herd size, a barn was designed for this project to house 1000 head of Holstein dairy cattle.

After the herd size was selected, a freestall barn was designed to house the 1000 head herd in order to determine required barn dimensions and the barn's ventilation system. The design of the freestall barn was a 6-row barn divided into three pens. The outer two pens have 300 freestalls, and the middle pen has 400 freestalls with all pens containing one crossover at the center of each pen (Appendix B).

The resulting freestall barn was 129.9 m x 81.1 m x 4.27 m (426 ft x 266 ft x 14 ft) with a 1√12 roof pitch (Figure 3.1). These dimensions were used in turn to design the ventilation system (Section 3.2.2).

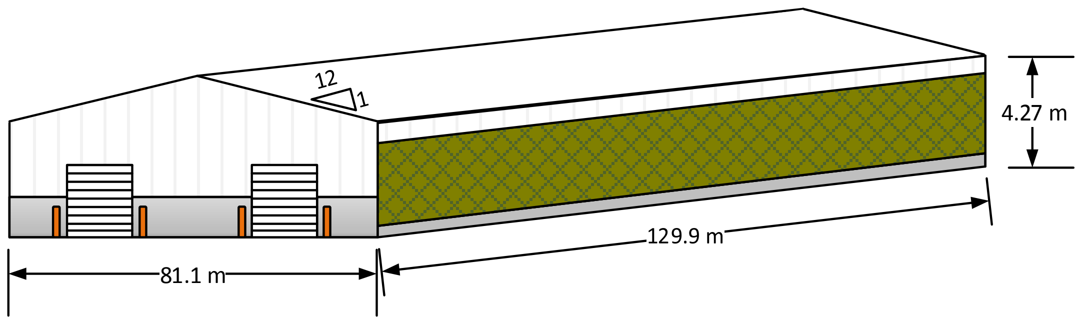


Figure 3.1. Outside view of model barn

The building dimensions were then used to determine the overall insulation level of the building, commonly referred to as the UA-value. The UA-value describes the ability for heat flow into or out of the building through the building's outer shell. Assumptions made for the design of the dairy facility included:

- Wood post frame construction
  - Board sizes: 2x4's, 2x6's and 2x12's
  - Acrolite Insulation (applied on all fixed surfaces built with common wood studs)
  - 24 gauge steel exterior roofing material
- 2.44 m (8 ft) concrete end walls
- 3.66 m (12 ft) curtain sidewall
- Mechanically cross-ventilated building
- Inside air film was still air

Commonly, agricultural facilities use white, corrugated steel roofing. Some producers may use a darker colored roof for appearance, but a darker roof color increases the solar absorptivity of the material, which increases the solar load on the building and in turn increases the long-wave thermal load on the housed cows, especially if little or no insulation is incorporated on the underside of the roof. For this barn model, a white roof was used, with an assumed absorptivity of 0.39 and a long-wave emissivity of 0.90 (Suehrcke, Peterson, & Selby, 2008). The emissivity for the remainder of the barn was 0.92, which is typical for common building materials at the temperatures encountered in animal housing (Holman, 1986).

Appendix C outlines the make-up of the building components with the associated thickness and resistivity to heat flow of those materials which ultimately resulted in an overall UA-value of the model barn of 10415 W $°C^{-1}$. This value was also used as an input to determine the ventilation requirements for the building (Section 3.2.2).

3.2.2 Ventilation requirements

The ventilation system for this facility was designed using basic mass and energy balance equations. Input parameters include the UA-value of the building, determined in section 3.2.1, the atmospheric pressure of the surrounding air, the cow's body weight, the cow's thermoneutral body temperature, the sensible and latent heat production of a thermoneutral cow, the desired indoor ambient conditions, and the outdoor weather conditions. For this model, a 600-kg cow was assumed with a core body temperature of 38.5°C. The desired indoor conditions that the ventilation rates were designed for was 20°C and 74% relative humidity resulting in a THI value of 67. This condition was selected in order to keep the moist air conditions in the building at levels just before the perceived onset of heat stress (THI = 68), if possible. As outdoor ambient conditions approach the desired levels (i.e., 20°C), without any additional cooling, inside conditions will rise above the desired conditions due to cow heat and solar loading inside the building. The level of temperature rise accepted above ambient conditions is one method for determining the required maximum hot weather ventilation rate. Through mass and energy balances, four different ventilation curves were developed to coincide with the various cooling strategies studied in this project, with the maximum hot weather rate dictated by either the level of temperature rise accepted or the average airspeed desired in the building. The first maximum ventilation rate was dictated by the desire to keep the inside air temperature no more than 2°C above entering conditions. This rule, called the “2°C rule”, resulted in the lowest average barn airspeed (low velocity case; LVC). The remaining three ventilation curves were maximized based on the average barn airspeed desired, a common design constraint used in tunnel or cross-flow ventilated dairy barns (See Appendix D for ventilation curve development). The three added ventilation curves were based on a desired average airspeed of 1 m s^{-1} (medium velocity case; MVC), 2 m s^{-1} (high velocity case; HVC), and 3 m s^{-1} (very high velocity case; VHVC). For these four ventilation design cases, the maximum airflow and actual average airspeed obtained in the model barn is shown in Table 3.2.

Table 3.2. Maximum barn airflow and average barn cross-sectional velocity modeled

Ventilation Case	Maximum Airflow (m^3 s^{-1})	Average Velocity (m s^{-1})
LVC (2°C Rule)	387	≈0.5
MVC	773	≈1
HVC	1547	≈2
VHVC	2321	≈3

Supplemental cooling was also used with these airflow cases. Evaporative cooling, which will be talked about more in depth in the next section, was incorporated with each ventilation case. The resulting abbreviations (Table 3.3) will be used through the remainder of this thesis indicating the ventilation case and the use of evaporative cooling.

Table 3.3. Cooling strategy abbreviations

Ventilation Case	Evaporative cooling?	Abbreviation
LVC	Yes	LVCE
LVC	No	LVCN
MVC	Yes	MVCE
MVC	No	MVCN
HVC	Yes	HVCE
HVC	No	HVCN
VHVC	Yes	VHVCE
VHVC	No	VHVCN

3.2.3 Psychrometric properties during evaporative cooler operation

Evaporative coolers are a type of cooling system used to decrease the dry-bulb temperature of the air through water evaporation. In doing so, however, the water vapor content of the air increases. Equation 3.1 was used to determine the new entering barn air temperature ($T_{db,evap}$) just after the outside air (at T_{db}) passes through the evaporative cooling pad and starts entering the animal occupied zone (AOZ):

$$T_{db,evap} = T_{db} - \eta_{evap} * (T_{db} - T_{wb}) \tag{3.1}$$

where,

η_{evap} = evaporative cooler efficiency (0-1, decimal).

During evaporative cooling, the wet-bulb temperature (T_{wb}) remains constant allowing the new dry-bulb temperature entering the AOZ to be determined. In all evaporative cooler applications for this project, an efficiency of 70% (η_{evap}=0.70) was assumed, representing a realistic industry average efficiency.

The new dry-bulb temperature entering the barn ($T_{db,evap}$) along with the known wet-bulb temperature was used to determine all of the remaining psychrometric properties of the moist air entering the barn (see Appendix A). If an evaporative cooler was not used, the outdoor dry-bulb temperature (T_{db}) and humidity ratio (W; kg_w/kg_a) entered the barn directly without alteration and was modeled as η_{evap} = 0.0.

For this research project, evaporative cooling was used when the outdoor temperature exceeded 23°C. This activation temperature was selected to ensure that the evaporative cooler would not be used in clearly non-heat stress situations such as late fall, winter, and early spring months. It was also used to ensure that the ventilation rate is maximized before any supplemental cooling begins in anticipation of potential heat stress conditions.

3.2.4 Predicting inside barn temperature, humidity ratio, and airspeed at the cow level

In tunnel or cross-flow ventilated barns, heat and moisture build up occurs from fresh-air inlet to fan exhaust due to incremental sensible and latent heat (i.e., water vapor) added by the cows up to the point of interest. Figure 3.2 shows a schematic of a cow in the modeled barn giving off heat and moisture. The outdoor air is being drawn in through the evaporative cooler, increasing water vapor content and decreasing dry-bulb temperature (if the evaporative cooler is on). The resulting moist air (with or without evaporative cooling) moves through the barn where sensible heat and water vapor given off by the cows is incrementally added before exiting the building through the fans.

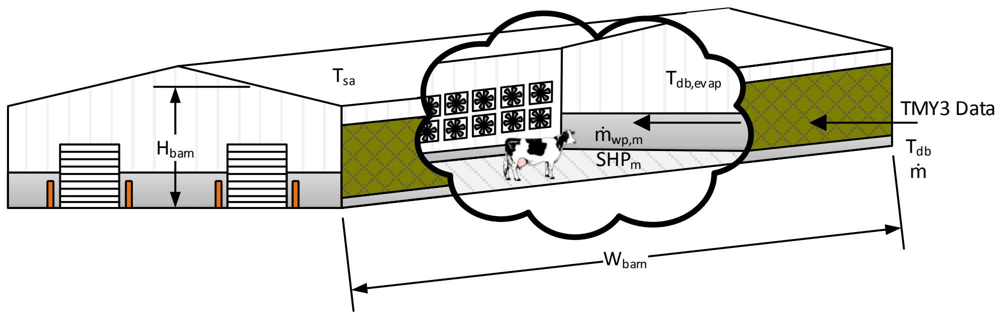


Figure 3.2. Model of the barn showing temperature and heat build up

In order to determine the ambient conditions surrounding the modeled cow, assumed at the center of the barn, ventilation rates were first determined based on the desired indoor conditions and the current outdoor conditions (previously described). A sensible energy balance (Equation 3.2) was then applied to determine the temperature rise through the barn:

$$T_{db,evap}(x+\Delta x) = T_{db,evap}(x) + \Delta x * W_{barn} * \frac{(T_{sa} - T_{db,evap})}{\dot{m} * cp_{mix} * R_{roof}} + \Delta x * H_{barn} * 2 * \frac{(T_{db} - T_{db,evap})}{\dot{m} * cp_{mix} * R_{walls}} + \Delta x * \frac{SHP_m}{\dot{m} * cp_{mix}} \quad (3.2)$$

where,

cp_{mix} = moist air specific heat (assumed at 1015 J kga^{-1} K^{-1})

H_{barn} = average height of barn (5.95 m)

$\dot{m}$ = mass flow of air (kg_a s^{-1})

R_{walls} = equivalent wall resistance (1.21 m^2 K W^{-1}; Appendix C)

R_{roof} = equivalent roof resistance (1.22 m^2 K W^{-1}; Appendix C)

SHP_m = sensible heat production by the cows per meter in the direction of air flow (W m^{-1})

T_{sa} = sol-air temperature (°C)

W_{barn} = width of barn (129.9 m).

A mass balance for water vapor (Equation 3.3) was then applied to predict the humidity ratio rise through the barn:

$$W(x + \Delta x) = W(x) + \frac{\dot{m}_{wp,m}}{\dot{m}}\Delta x \tag{3.3}$$

where,

$\dot{m}_{wp,m}$ = mass flow of water vapor produced by the cows per meter in the direction of air flow ($kg_w\ s^{-1}\ m^{-1}$).

The heat and moisture rise throughout the barn assumes that the barn is full, the cows are uniformly distributed, and is taking into consideration all of the sensible heat and water vapor given off by all cows, and then incrementing that sensible heat and water vapor production every Δx=0.10 meters up until the center of the barn, where the modeled cow is located. After the barn's center dry-bulb temperature and humidity ratio (W) were determined, all other psychrometric properties were calculated and used as input parameters for modeling the sensible and latent heat exchange from the cow to her surroundings.

3.2.5 Inputting meteorological psychrometrics into barn for sprinkling

Large droplet sprinklers and small droplet misters are common added cooling strategies used on dairy operations. For this research project, water is directly applied to the cow's back. Section 3.3 describes the heat and mass transfer effects of sprinkling. The assumption made for the sprinkling system in this research project was that if the environment was capable of "accepting" additional water vapor beyond sweating needs, then additional water via sprinkling was applied to cool her down. Typically in dairy operations, a sprinkler system would run on a timer system for programmed on and off times. For this research project, only the amount of water actually needed to cool her down is applied, and only at a time when it is beneficial (i.e., the water will be utilized to cool her and not wasted).

3.3 The Cow Model

3.3.1 Sensible heat and water vapor transfer of a cow

Being able to predict sensible and latent heat exchange between a cow and her surroundings is a crucial step in assessing the effectiveness of heat stress mitigation and requires an understanding of how a cow dissipates heat to her surroundings as shown in Figure 3.3.

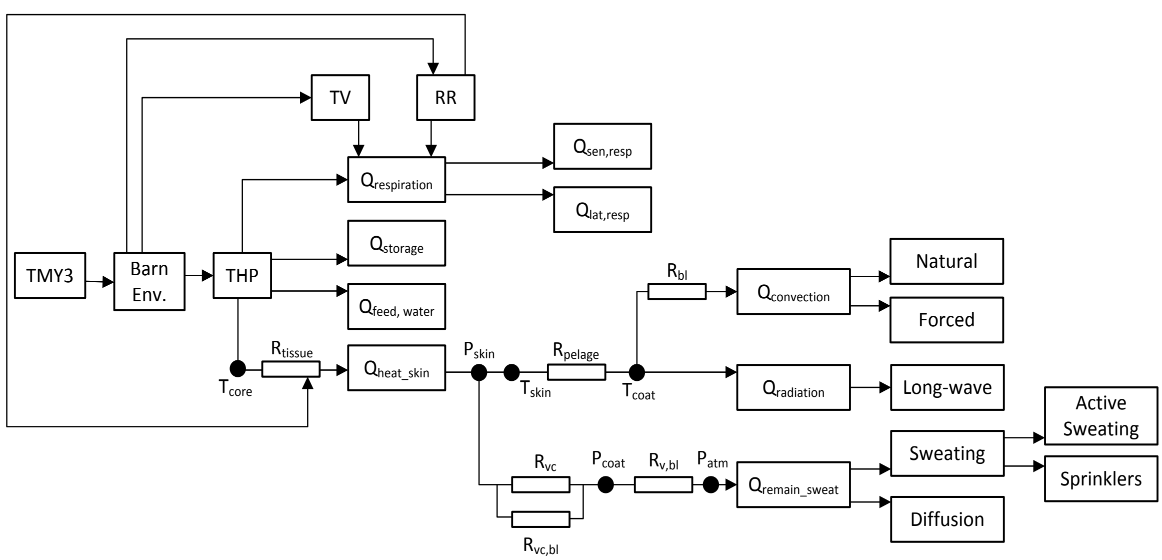
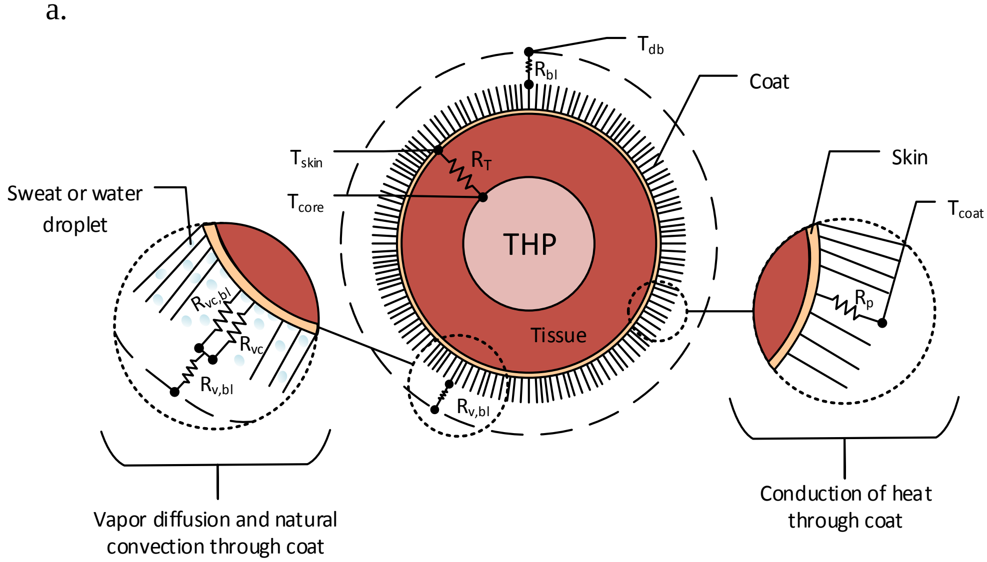


Figure 3.3. Flow chart of heat dissipation from cow

Specific details of the processes outlined in Figure 3.3 will be discussed in a later section, but in general, the parameters from the TMY3 Class 1 data set (Table 3.1), with or without evaporative cooling, is brought into the barn model and the moist air conditions surrounding her at the barn center are determined using Equations 3.2 and 3.3. The surrounding moist air properties are then used to determine her tidal volume (TV) and respiration rate (RR), with RR then used to determine the cow's tissue insulation level ($R_{t,actual}$). To dissipate the total heat produced (THP), four different heat loss paths are possible. She can lose heat latently and sensibly by heating and humidifying respired moist air, she uses a portion of the THP to heat up the ingested food and water to her core body temperature, and she can dissipate a portion of THP through her tissue to her skin where evaporation of sweat occurs or convection and radiation to her surroundings occur. The balance of THP, if any, is then stored if she is unable to dissipate all of the heat produced.

Figures 3.4a,b show diagrams of the sensible heat and water vapor transfer from the core of the cow. Figure 3.4a is a close up view of the coat layer when a cow sweats or water is added to her back from a sprinkler system, and Figure 3.4b shows a resistance network of heat and water vapor flow from her core to her surroundings. This network applies to the balance of THP after respiration and the heat used to raise feed and water to the cow's core temperature has been accounted for. It also describes the portion of THP transferred by conduction through her tissue layer ($R_{tissue} = R_{t,actual}$) to the skin. From the skin, heat will either be conducted through her coat layer (R_{pelage}) ultimately leaving through convection (R_{bl}) to the surrounding moist air, through radiation (R_{rad}) to the surrounding surfaces, or used to evaporate sweat or added sprinkling water. The mean radiant temperature (T_{MRT}), dictated by surrounding surface temperatures, influences the heat lost by radiation, and the dry-bulb temperature and airspeed influence the convective heat lost to the surroundings. A portion of the heat that reaches the skin can be used to evaporate sweat or added sprinkling water. There are two different paths that influence water evaporation at the skin. Due to the water vapor pressure differential between the skin layer ($P_{w,skin}$) and the top of the coat layer ($P_{w,coat}$), water vapor can be diffused through the coat itself (R_{vc}) or in parallel through the boundary layer of the coat ($R_{vc,,bl}$) as a result of natural mass convection (McGovern & Bruce, 2000). Finally, this parallel water vapor path through the coat layer is in series with the mass convection ($R_{v,bl}$) that occurs to the surrounding moist air, influenced by the moist air water vapor pressure differential ($P_{w,atm}$) and the free-stream airspeed which influences the outer boundary layer into the moist air surrounding the cow (McGovern & Bruce, 2000).

a.



b.

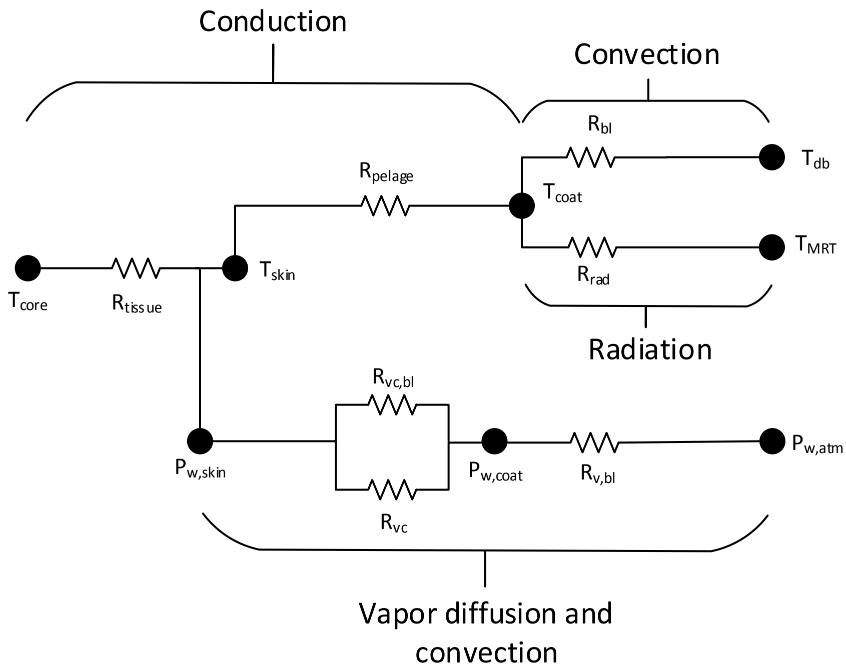


Figure 3.4. (a) Schematic diagram of modeled heat and vapor flow from core to surrounding environment and (b) Resistance network for heat and vapor flow from core of cow to surrounding environment

3.3.2 The cow model components

After extensive literature review, the model developed for this project originated from McGovern & Bruce (2000) supplemented with constants and equations from several other sources, which are outlined in Appendix E.

Heat dissipation from a cow to her surrounding begins with the amount of heat the cow internally generates (THP) and the heat flow paths available to reach the surrounding environment (Equation 3.4). As described earlier, the total heat produced has multiple paths:

$$THP = \dot{Q}_{resp} + \dot{Q}_{FW} + \dot{Q}_{cv} + \dot{Q}_{rad} + \dot{Q}_{evap} + \dot{Q}_{stored} \tag{3.4}$$

where,

$\dot{Q}_{resp}$ = THP portion lost to surrounding moist air through respiration (W)

$\dot{Q}_{FW}$ = THP portion used to heat the ingested feed and water to the cow's core temperature (W)

$\dot{Q}_{cv}$ = THP portion lost to surrounding moist air through convection (W)

$\dot{Q}_{rad}$ = THP portion lost to surrounding surfaces through long-wave radiation (W)

$\dot{Q}_{evap}$ = THP portion lost to surrounding moist air through water evaporation via sweating or sprinkling (W)

$\dot{Q}_{stored}$ = THP portion that must be stored to balance energy (W).

The total heat exchanged from a cow is the sum of the latent and sensible heat dissipated (or gained). Dairy cattle heat and moisture production values, based on calorimetry studies, can be found in Appendix 5-1 of *Environment Control for Animals and Plants* (Albright, 1990). The total heat produced by a cow is dependent on many variables, but typically THP for homeotherms is published as a function of air temperature surrounding the cow. Published THP data for dairy cows was used for this project as a baseline THP for all climates. The published data (Albright, 1990) was used and a regression equation was developed from which THP was calculated (Equation 3.5). A factor of 1.3 was applied to Equation 3.5 to account for modern higher performing cows, based on more recent heat production data (described below). For this model, it was assumed that there was an optimal THP that occurs at T_{db} = 15°C:

$$THP = \left(-0.0168 * T_{db,center} + 2.3666\right) * \left(\frac{m_{cow}}{m_{table}}\right)^{0.734} * m_{table} \tag{3.5}$$

where,

m_{cow} = mass of modeled cow (600 kg)

m_{table} = mass of comparison cow (500 kg)

$T_{db,center}$ = air temperature surrounding the cow modeled (°C).

The ratio of the mass of the modeled cow to the mass of the comparison cow is used in order to adjust for different weights of animals (Albright, 1990).

A similar procedure was followed to estimate latent heat production (Equation 3.6). A factor of 1.3 was also applied to this equation:

$$LHP = \left(0.0263 * T_{db,center} + 0.4629\right) * \left(\frac{m_{cow}}{m_{table}}\right)^{0.734} * m_{table}. \tag{3.6}$$

Finally, the sensible heat production was determined by subtraction (Equation 3.7):

$$SHP = THP - LHP. \tag{3.7}$$

This sensible heat production value was used to determine the heat build-up and ventilation requirements as previously described in Section 3.2.2 and used in Equation 3.2. A comparison with field estimated THP was used to determine the adequacy of the THP level modeled with equation 3.5. The Department of Animal Science from The Ohio State University determined, through calorimetry, that a lactating 600 kg cow producing 35 kg of milk a day, produced 34.6 Mcal day^{-1}, or 1677 W of total heat (Weiss, 2016). Using equations 3.5-3.7 at 15°C, the modeled THP was equivalent to 1571 W (LHP = 637 W, with the balance SHP = 934 W). The agreement (within 6.3% of actual THP) warranted the use of equations 3.5-3.7 for varying dry bulb temperatures surrounding the modeled cow.

The energy lost during respiration (Equation 3.8), for the modeled cow at the center of the barn, was handled as a classic psychrometric heating and humidifying process including virtual body and air temperatures (McArthur, 1987):

$$\dot{Q}_{resp} = \dot{Q}_{sensible,resp} + \dot{Q}_{latent,resp} \tag{3.8}$$

where,

$\dot{Q}_{sensible,resp}$ = the dry air heat lost during respiration (W)

$\dot{Q}_{latent,resp}$ = the moist air heat lost during respiration (W).

The equations used for the sensible and latent heat lost during breathing (Equations 3.9 and 3.10) were modified from the equations used in Thompson, Barioni, et al., (2011):

$$\dot{Q}_{sensible,resp} = \left(\frac{1000}{100^3}\right) * \left(\frac{RR}{60}\right) * TV * \left(\frac{1}{\nu_{center}}\right) * cp_{mix} * (T_{vb} - T_{va}) \tag{3.9}$$

where,

cp_{mix} = mixture specific heat of moist air (J kg_a^{-1} K^{-1})

RR = respiration rate (BPM) – See Appendix E

TV = tidal volume (L $breath^{-1}$) – See Appendix E

T_{va} = virtual air temperature (K)

T_{vb} = virtual body temperature (K)

ν_{center} = specific volume of air at the center of the barn (m^3 kg_a^{-1})

$$\dot{Q}_{latent,resp} = \left(\frac{1000}{100^3}\right) * \left(\frac{RR}{60}\right) * TV * \left(\frac{1}{\nu_{center}}\right) * h_{fg} * (W_{lungs} - W_{center}) \tag{3.10}$$

where,

h_{fg} = latent heat of vaporization (assumed at 2410000 J kg_w^{-1})

W_{center} = humidity ratio of air at the center of the barn (kg_w kg_a^{-1})

W_{lungs} = humidity ratio of air from the lungs (kg_w kg_a^{-1}).

The mixture specific heat of moist air (Equation 3.11) used the humidity ratio at the center of the barn:

$$cp_{mix} = cp_a + W_{center} * cp_w \tag{3.11}$$

where,

cp_a = specific heat of dry air (1006 J kg_a^{-1} K^{-1})

cp_w = specific heat of water vapor (1850 J kg_w^{-1} K^{-1}).

The virtual body and air temperatures (Equations 3.12 and 3.13), defined as the temperature at which dry air has the same density (McArthur, 1987), includes temperature and vapor pressure differences:

$$T_{vb} = (T_{core} + 273.15) * \left(1 + 0.38 * \frac{P_{ws,body}}{P}\right) \tag{3.12}$$

$$T_{va} = (T_{db,center} + 273.15) * \left(1 + 0.38 * \frac{P_{ws,air}}{P}\right) \tag{3.13}$$

where,

T_{core} = core body temperature of cow (°C)

$P_{ws,body}$ = saturation vapor pressure of the body (Pa)

$P_{ws,air}$ = saturation vapor pressure of the air (Pa)

P = atmospheric pressure (Pa).

The required saturation vapor pressure and the humidity ratio of the lungs is given in Equations 3.14 – 3.16 (Albright, 1990):

$$P_{ws,body} = \frac{e^{77.345+0.0057*(T_{core}+273.15)-\frac{7235}{T_{core}+273.15}}}{(T_{core} + 273.15)^{8.2}} \tag{3.14}$$

$$P_{ws,air} = \frac{e^{77.345+0.0057*\left(T_{db,center}+273.15\right)-\frac{7235}{T_{db,center}+273.15}}}{\left(T_{db,center} + 273.15\right)^{8.2}} \tag{3.15}$$

$$W_{lungs} = 0.6221 * \left(\frac{P_{w,lungs}}{P - P_{w,lungs}}\right) \tag{3.16}$$

where,

$P_{w,lungs}$ = vapor pressure in the lungs ($0.85*P_{ws,body}$, Pa).

The amount of heat required to raise ingested food and water ($\dot{Q}_{FW}$) to core temperature was calculated by using a basic internal energy calculation (Equation 3.17):

$$\dot{Q}_{FW} = (TCF)[DMI * Cp_{DM}(T_{core} - T_{DM}) + WI * Cp_{water}(T_{core} - T_{water})] \tag{3.17}$$

where,

Cp_{DM} = specific heat of dry matter (3250 J kg^{-1} $°C^{-1}$; JIANG & Jofriet, 1988)

Cp_{water} = specific heat of water (4186 J kg_w^{-1} $°C^{-1}$)

DMI = dry matter intake (kg day^{-1})

TCF = time conversion from days to seconds $(24 * 3600\ s\ day^{-1})^{-1}$

T_{core} = body core temperature (°C)

T_{DM} = dry matter temperature entering cow (°C)

T_{water} = water temperature entering cow (°C)

WI = water intake (kg day^{-1}).

As shown in Equation 3.17, the cow uses internally produced energy to heat ingested food and water to her core temperature. Due to the complication of correlating feed and water intake with milk production, which can be greatly influenced during heat stress, feed (DMI) and water (WI) intake rates (Equations 3.18 and 3.19) were modeled (Appendix G) from the National Research Council (U.S.), Committee on Animal Nutrition, & Subcommittee on Environmental Stress (1981):

$$DMI = -0.0003 * T_{db,center}^3 + 0.0056 * T_{db,center}^2 - 0.0103 * T_{db,center} + 22.919 \tag{3.18}$$

and

$$WI = 0.0347 * T_{db,center}^2 + 2.1338 * T_{db,center} + 78.37. \tag{3.19}$$

If the predicted DMI of the cow was below 2% of her body weight, the assumption was made that her DMI would minimize at 2% of her body weight (Baumgard, 2016). The specific heat of dry matter (C_{DM}) used was estimated at 3250 J kg^{-1} K^{-1} (Jiang & Jofriet, 1988). The temperature of the ingested feed was assumed to have the same temperature as the outdoor dry bulb temperature, which assumes feed is stored outside the barn before being mixed. The water temperature was assumed to be the average of the dry bulb temperature and the ground water temperature (Figure 3.5). This assumption was made since the water sits in the water tanks in the barn and is therefore exposed to the ambient conditions in the barn after being pumped from ground water temperature.

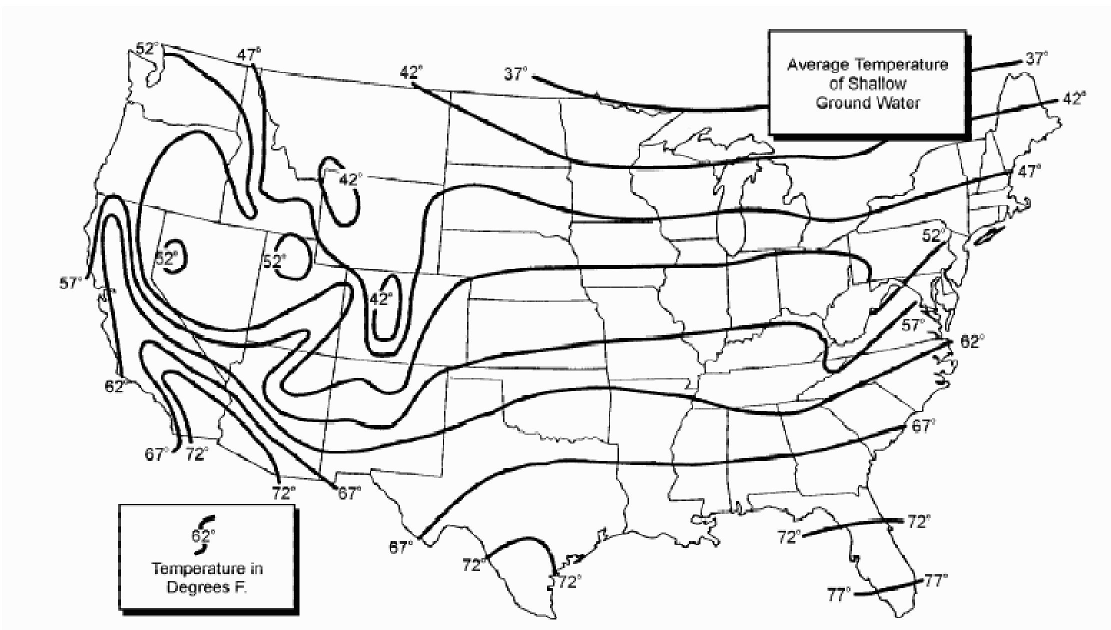


Figure 3.5. Map of average ground water temperatures throughout the United States

Finally, the remaining heat can be dissipated through body surface heat transfer (i.e., convection, radiation, and evaporation of sweat or sprinkler water). Convection heat transfer was modeled as:

$$\dot{Q}_{cv} = A_b * \frac{(T_{skin} - T_{db,center})}{(R_{bl} + R_p)} \tag{3.20}$$

where,

A_{cow} = area of the body of the cow (5.37 m^2; Berman, 2003)

R_{bl} = resistance to heat flow through the boundary layer (m^2 K W^{-1})

R_p = resistance to heat flow through the pelage or coat layer (0.086 m^2 K W^{-1}; McGovern & Bruce, 2000)

T_{skin} = the skin temperature (°C).

Due to convection being dependent on the difference between the skin and air temperatures, an equation for the skin temperature was back-calculated as shown in Equation 3.21:

$$T_{skin} = T_{core} - \left(\frac{R_{t,actual}}{A_b}\right) * \dot{Q}_{heat,remain} \tag{3.21}$$

where,

$\dot{Q}_{heat,remain}$ = the amount of THP remaining after being used for respiration and feed and water intake (W)

$R_{t,actual}$ = tissue resistance (m^2 K W^{-1}) – See Appendix E

T_{core} = core body temperature of cow (°C).

The resistance to heat flow through the boundary layer (Equation 3.22) incorporates natural and forced convection. Forced convection is convection due to the air moving over the cow’s body from the ventilation rates determined and resulting cross-sectional velocity previously determined (Table 3.2):

$$R_{bl} = \frac{1}{h_{overall}} \tag{3.22}$$

where,

$h_{overall}$ = the convective heat transfer coefficient (W m^{-2} K^{-1}).

Appendix H outlines the natural and forced convective heat transfer equations used to determine the convective heat transfer coefficient. The overall convective heat transfer coefficient was determined by using the larger of natural versus forced convective heat transfer coefficients.

Radiation can be very impactful on a cow's heat load. Equation 3.23 shows the governing equation to determine the radiative load on the cows:

$$\dot{Q}_{rad} = A_b * \sigma * \varepsilon_{cow} * (\left(T_{ave,rad} + 273.15\right)^4 - T_{MRT}^4) \tag{3.23}$$

where,

$T_{ave,rad}$ = weighted sum of the skin and air temperatures (°C)

T_{MRT} = mean radiant temperature (K)

ε_{cow} = emissivity of the cow (0.98 dimensionless; Thompson et al., 2014)

σ = Stefan Boltzmann constant ($5.67*10^{-8}$ W m^{-2} K^{-4}).

The weighted sum of the skin (0.80) and air (0.20) temperatures was used to account for pelage effects on the actual skin temperature exposed for thermal radiation. These values were determined based on trial and error comparing published and modeled proportions of sensible and latent heat production.

The mean radiant temperature (Equation 3.24) was determined assuming two surfaces interacted radiatively with the cow, namely the roof by itself and the rest of the internal surfaces collectively grouped as one surface:

$$T_{MRT} = \left((T_{roof}^4 * F_{cow,roof}) + (T_{walls}^4 * F_{cow,all\ else})\right)^{0.25} - 273.15 \tag{3.24}$$

where,

$F_{cow,roof}$ = shape factor for the cow to the roof (0.499 – Monte Carlo simulation, Hoff, 1987)

$F_{cow,all\ else}$ = shape factor of the cow to the remainder of the building less the roof

T_{roof} = temperature of the roof (K)

T_{walls} = temperature of the walls (K).

The roof and wall temperatures were found using the sol-air temperature method (Equation 3.25, Albright, 1990) and basic conductive heat transfer equations (Equation 3.28):

$$T_{roof} = \left(\frac{(T_{sa} - T_{db,center})}{(\frac{1}{h_{cv,out}} + R_{roof} + R_{stillair})} * R_{stillair} \right) + T_{db,center} \tag{3.25}$$

where,

$h_{cv,out}$ = exterior convective heat transfer coefficient (W m^{-2} K^{-1})

R_{roof} = resistivity of the roof (m^2 K W^{-1}) – See Appendix C

$R_{stillair}$ = resistivity of the walls (0.12 m^2 K W^{-1}; ASHRAE 1985)

T_{sa} = Sol-air temperature (°C).

The sol-air temperature (Equation 3.26) takes into consideration the air temperature, solar heating, outside roof convection, and thermal radiation from the roof to the "sky" and combines these processes into an equivalent outside air temperature:

$$T_{sa} = T_{db} + \frac{\alpha * GHI - 6 * \varepsilon_{roof} * \cos\left(\theta_{roof} * \frac{\pi}{180}\right) * (10 - \phi)}{h_{cv,out}} \tag{3.26}$$

where,

GHI = global horizontal irradiance (W m^{-2})

α = roof absorptivity (0.39 dimensionless; (Suehrcke et al., 2008))

ε_{roof} = roof emissivity (0.92 dimensionless; Suehrcke et al., 2008)

θ_{roof} = tilt angle of the roof (4.76°)

ϕ = cloudiness factor (0 (no clouds) -10 (full cloud cover), dimensionless).

The exterior convective heat transfer coefficient is dependent on the outdoor wind speed. A second order polynomial correlation was made using Table 3.4 to produce Equation 3.27:

Table 3.4. Exterior convective heat transfer coefficient correlation

Wind Speed, ws (m s^{-1})	**$h_{cv,out}$ (W m^{-2} K^{-1})**
0	8.33
2.1	17.79
3.4	22.90
5.2	29.03
6.7	33.32

$$h_{cv,out} = -0.1681 * ws^2 + 4.8556 * ws + 8.33 \qquad (3.27)$$

where,

ws = wind speed imposed on exterior of modeled barn from TMY3 data (m s^{-1}).

The governing equation for determining the interior temperature of a barn wall surface due to the process of conductive heat transfer through a wall is shown in Equation 3.28:

$$T_{material} = T_{db,center} + \frac{R_{stillair} * (T_{db} - T_{db,center})}{R_{material}} \qquad (3.28)$$

where,

$R_{material}$ = equivalent resistance of the wall (m^2 K W^{-1}) – See Appendix C.

In order to find the average wall temperature for the barn used in Equation 3.24, a weighted average of the component area was used for each barn wall component (i.e. concrete, curtain, end wall, or fan wall). For example, if one 10 m^2 wall had an interior surface temperature of 20°C, with another 20 m^2 wall having an interior surface temperature of 25°C, the weighted average wall temperature is 23.3°C. This same procedure was conducted for all non-roof surfaces to yield an overall T_{walls} for determining T_{MRT} (Equation 3.24).

Evaporation due to sweat or applied water can be one of the best cooling mechanisms for dairy cattle. Equation 3.29 outlines evaporation heat loss for dairy:

$$\dot{Q}_{evap} = A_b * \beta * h_{fg} * \frac{(W_{skin} - W_{center})}{R_{v,eq} * v_{center}} \tag{3.29}$$

where,

h_{fg} = latent heat of vaporization (assumed at 2410000 J kg_w^{-1})

$R_{v,eq}$ = equivalent resistance to vapor flow through the coat (s m^{-1})

W_{skin} = humidity ratio at skin surface (kg_w kg_a^{-1})

β = sweating factor (0.5534: actual percentage, in decimal form, of the body area that is sweating at any given time; described below).

A cow can sweat, but if the surrounding environment is unable to evaporate this sweat, little if any cooling benefit is realized. The following relations and rationale were used to determine if the environment is capable of evaporating enough sweat in order to balance the cow's energy, which has been updated for vapor and mass transfer through the coat and outside coat boundary layer (McGovern & Bruce, 2000).

The humidity ratio at the skin surface (Equation 3.30) uses the vapor pressure at the skin level (Equations 3.31 and 3.32):

$$W_{skin} = 0.6221 * \frac{P_{w,skin}}{(P - P_{w,skin})} \tag{3.30}$$

where,

$P_{w,skin}$ = vapor pressure at the skin level (Pa)

$$P_{w,skin} = 0.95 * P_{ws,skin} \tag{3.31}$$

where,

$P_{ws,skin}$ = saturated vapor pressure at the skin level (Pa)

$$P_{ws,skin} = \frac{e^{77.345+0.0057*T_{skin}-\left(\frac{7235}{T_{skin}}\right)}}{T_{skin}^{8.2}} \tag{3.32}$$

where,

T_{skin} = skin temperature (K).

Equation 3.29 includes a "sweating factor" β that was required to align the maximum published latent heat loss with experimental data on the maximum respiration rate, tidal volume, and sweating rate. For example, if the entire surface area of a 600 kg cow is allowed to sweat at the maximum published rate, and is also respiring at the maximum respiration rate combined with the maximum published tidal volume, then the predicted latent heat loss far exceeds maximum published latent heat loss data. A procedure was developed to back calculate the actual surface area of a cow's body that will actively sweat by using the following equations:

$$\beta = \frac{(LHP_{max} - \dot{Q}_{RR,loss,max})}{\dot{Q}_{SR,loss,max}} \tag{3.33}$$

where,

LHP_{max} = maximum latent heat production from published data (W)

$\dot{Q}_{RR,loss,max}$ = heat loss at the maximum respiration rate (W)

$\dot{Q}_{SR,loss,max}$ = heat loss at the maximum sweating rate (W)

$$LHP_{max} = LHP_{factor}\left(\frac{m_{cow}}{m_{table}}\right)^{0.734} * m_{table} \tag{3.34}$$

where,

LHP_{factor} = estimated maximum latent heat production (1.4 W kg^{-1})

$$\dot{Q}_{RR,loss,max} = RR_{max} * TV_{max} * \rho_{air} * \left(W_{lungs} - W_{center}\right) * h_{fg} \tag{3.35}$$

where,

RR_{max} = maximum respiration rate (120 BPM; Renaudeau et al., 2012)

TV_{max} = maximum tidal volume (4.24 L; Berman, 2005)

$$\dot{Q}_{SR,loss,max} = SR_{max} * A_{cow} * h_{fg} \tag{3.36}$$

where,

SR_{max} = maximum sweating rate (288 g_w m^{-2} hr^{-1}; McArthur, 1987).

Figures 3.4a,b outlined the breakdown of the resistance to vapor flow through the coat layer (Equation 3.37):

$$R_{v,eq} = R_{v,bl} + \frac{R_{vc} * R_{vc,bl}}{(R_{vc} + R_{vc,bl})} \tag{3.37}$$

where,

$R_{v,bl}$ = resistance to vapor flow convection through the outer boundary layer (s m^{-1})

R_{vc} = resistance to vapor flow diffusion through the coat (s m^{-1})

$R_{vc,bl}$ = resistance to vapor flow convection through the coat (s m^{-1}).

These resistances are the inverse of the corresponding convective mass transfer coefficients. Equations 3.38 through 3.40 outline these coefficients. Equation 3.38 refers to the convection vapor transport through the outer boundary layer:

$$h_{v,bl} = h_{overall} * \frac{\nu_{center}}{cp_{air} * Le_{coat}^{\frac{2}{3}}} \tag{3.38}$$

where,

Le_{coat} = the Lewis number (dimensionless)

v_{center} = moist air specific volume at barn center ($m^3\ kg_a^{-1}$).

The Lewis number is a dimensionless number that is used in any situation involving the simultaneous heat and mass transfer by convection. Equation 3.39 shows the ratio used to solve for the Lewis number:

$$Le = \frac{Sc_{coat}}{Pr} \tag{3.39}$$

where,

Sc_{coat} = Schmidt number, which is the ratio of the kinematic viscosity over the diffusivity of water vapor (dimensionless).

Equation 3.40 gives the diffusion of vapor through the coat layer:

$$h_{vc} = \frac{d_{vapor}}{(L_{coat} - \Delta L_{coat})} \tag{3.40}$$

where,

d_{vapor} = diffusivity of water vapor (0.0000285 $m^2\ s^{-1}$; Thompson et al., 2014)

L_{coat} = coat length (0.01 m; McGovern & Bruce, 2000)

ΔL_{coat} = coat length decrease due to airspeed over coat (m).

If the airspeed is very high over a cow's hide, her pelage will naturally flatten, which decreases the physical distance between the skin and the ambient conditions. Equation 3.41 was used to account for this phenomena (based from McGovern & Bruce, 2000):

$$\Delta L_{coat} = 0.0025 * \vec{V} \tag{3.41}$$

where,

$\vec{V}$ = Velocity ($m\ s^{-1}$).

Equation 3.42 represents the vapor convection coefficient through the coat layer where it is assumed that natural convection governs (McGovern & Bruce, 2000):

$$h_{vc,bl} = Sh * \frac{d_{vapor}}{D_{cow}} \tag{3.42}$$

where,

Sh = Sherwood number (dimensionless).

The Sherwood number refers to the vapor concentration gradient at the surface and is represented by Equation 3.43:

$$Sh = Nu_{coat} * Le^{0.25} \tag{3.43}$$

where,

Nu_{coat} = Nusselt number within the coat layer (dimensionless).

The Nusselt number within the coat (Equation 3.44) is the ratio of convection to pure conduction heat transfer using the diameter of the cow as the characteristic dimension:

$$Nu_{coat} = 0.48 * Gr^{0.25} \tag{3.44}$$

where,

Gr = Grashof number (dimensionless).

The Grashof number (Equation 3.45) is a measure of the ratio of buoyancy forces to viscous forces:

$$Gr = g * D_{cow}^3 * P_{mbar} * \left|T_{coat} - T_{db,center}\right| + \frac{0.61 * \left|P_{w,coat,mbar} * T_{coat,K} - P_{w,mbar} * T_{db,center,K}\right|}{273 * \Delta P_{a,mbar} * (\nu * \rho)^2} \tag{3.45}$$

where,

P_{mbar} = atmospheric pressure (mbar)

$\Delta P_{a,mbar}$ = difference between the atmospheric pressure and vapor pressure at the center of the barn (mbar)

$P_{w,mbar}$ = vapor pressure at the center of the barn (mbar)

$P_{w,coat,mbar}$ = average of the vapor pressure of the skin and the vapor pressure at the center of the barn (mbar)

T_{coat} = average of the skin temperature and the dry bulb temperature at the center of the barn (°C)

$T_{coat,K}$ = coat temperature (K).

Finally, the potential mass flow rate for evaporation can be calculated (Equation 3.46):

$$\dot{m}_{pot} = (W_{skin} - W_{center}) * h_{v,eq} * 3600 * \frac{1000}{v_{center}} \tag{3.46}$$

where,

$h_{v,eq}$ = the equivalent vapor convective mass transfer coefficient ($R_{v,eq}^{-1}$ m s^{-1}).

The remaining heat from the cow will be dissipated through sweating, if the surrounding environment has the ability to accept added vapor from the cow's surface. Equation 3.47 represents the final amount of energy (W) required to be lost through sweating if the cow is to maintain core body temperature:

$$\dot{Q}_{remain,sweat} = \dot{Q}_{heat,remain} - (\dot{Q}_{cv} + \dot{Q}_{rad}). \tag{3.47}$$

The mass flow rate of the required sweating rate is outlined in Equation 3.48:

$$\dot{m}_{sweat,req} = \frac{\dot{Q}_{remain,sweat} * 1000 * 3600}{h_{fg} * A_{cow}}. \tag{3.48}$$

If the environment is able to absorb all of the required sweat for thermal balance, then the assumption is made that the cow will sweat at the rate required for thermal balance, provided this rate is at or below the maximum possible (W) (Equation 3.49):

$$\dot{Q}_{sweat,max} = \frac{SR_{reqd} * \beta * A_{cow} * h_{fg}}{3600 * 1000}. \tag{3.49}$$

The environment will only absorb the vapor it can handle ($\dot{Q}_{evap}$) and if the energy remaining for sweating ($\dot{Q}_{remain,sweat}$) is greater than $\dot{Q}_{evap}$, the actual sweating rate will be equal to $\dot{Q}_{evap}$. If she is sweating at a rate lower than $\dot{Q}_{evap}$ she will continue to sweat at that rate. The mass flow rate of the evaporated sweat actually being absorbed by the environment is given in Equation 3.50:

$$\dot{m}_{sweat,evap} = \frac{\dot{Q}_{sweat,actual} * 1000 * 3600}{h_{fg} * A_{cow}}. \tag{3.50}$$

If the cow is required to dissipate more energy through sweating than the amount the surrounding environment can evaporate, then this excess energy will be stored (Equation 3.51):

$$\dot{Q}_{storage} = \dot{Q}_{sweat,remain} - \dot{Q}_{sweat,actual}. \tag{3.51}$$

Prior to the need to store energy, the RR is upwardly adjusted up to the maximum RR, allowing for added energy to be lost via respiration in an attempt to maintain core body temperature. For this model, the respiration rate is allowed to increase to a maximum (RR_{max} = 120 BPM), if required for balance.

Finally, if the sweating rate and latent loss via respiration is insufficient to balance energy and the RR has been maximized, the excess energy is stored causing a rise in core body temperature (Equation 3.52) in the 1-hour time increment dictated by the resolution of the TMY3 data set:

$$T_{core,new} = T_{core,current} + TF * \frac{\dot{Q}_{stored}}{Cp_{body} * W_t} \tag{3.52}$$

where,

$T_{core,current}$ = current core temperature entering the current hour of analysis (°C)

$T_{core, new}$ = updated core temperature after 1-hr (°C)

Cp_{body} = specific heat of cow's body (3400 J kg^{-1} °C^{-1}; Monteith, 1973)

TF = time conversion from hour to seconds (3600 s hr^{-1}).

Conversely, if the environment is capable of dissipating more heat than that required for thermal balance, and the core temperature is above thermoneutral (38.5°C), her core body temperature (°C) is allowed to incrementally decrease as shown in Equation 3.53:

$$T_{core} = T_{core} - \frac{(\dot{Q}_{evap} - \dot{Q}_{sweat,remain}) * TF}{m_{cow} * Cp_{body}}. \tag{3.53}$$

Finally, if the environment is again capable of absorbing more vapor than the cows produce through her maximum sweating rate, then sprinkling can be utilized. This model only takes into consideration the potential of sprinkling (Equation 3.54) and determines how much water can to be added to the cow's body:

$$SP_{pot} = \dot{Q}_{evap} - \dot{Q}_{sweat,remain}. \tag{3.54}$$

In reality, producers have their sprinklers running on a timer, which could be adding too much water, therefore, wasting water, or the sprinklers may not be adding enough water to achieve optimal heat loss benefit. The sprinkling potential was used to determine how much sprinkling water (SP_{gal}) could be utilized (gal hr^{-1} cow^{-1}) to reduce heat stress through Equation 3.55:

$$SP_{gal} = \frac{\left(\frac{SP_{pot}}{h_{fg}}\right)}{\rho_{water}} * UC \tag{3.55}$$

where,

ρ_{water} = density of water (997 kg m^{-3})

UC = unit conversions from m^3 s^{-1} to gal hr^{-1} (3.28^3 ft^3 m^{-3} * 7.5 gal ft^{-3} *3600 s hr^{-1}).

During sprinkling events, if the potential heat loss afforded by sprinkling allows for the respiration rate to be reduced, still allowing for thermal balance, the respiration rate is allowed to incrementally decrease accordingly taking advantage of the positive benefit of increase heat loss through sprinkling water evaporation.

Similarly, if evaporative cooling is utilized, the total amount of water (EC_{gal}) needed (gal hr^{-1} cow^{-1}) for a 70% efficient evaporative cooler follows Equation 3.56:

$$EC_{gal} = \frac{(\dot{m} * \Delta W)}{\rho_{water}} * UC \tag{3.56}$$

where,

ΔW = Difference between outdoor humidity ratio and the humidity ratio just after passing through the evaporative cooler (kg_w kg_a^{-1}).

3.4 Model Comparison with Field Data

The previously described barn and cow models were used to compare with collected field data (Gebremedhin et al., 2010) using similar parameters to show the effectiveness of the developed models, using TMY3 Class I stations in Wisconsin (SN: 726435) and California (SN:723890). Tables 3.5 and 3.6 show this comparison. Both the hot and dry and hot and humid conditions were analyzed in the field study comparison at an airspeed of 1 m s^{-1}. Also, the following results from the field study are from shaded cows similar to the situation encountered inside buildings as modeled with this project. The field study was compared to the developed barn and cow models using selected six hour periods from the Wisconsin and California TMY3 data sets. The six hour continuous time periods used from the developed models were selected based on matching the THI levels reported in the field study. The field study was conducted for twelve high-producing Holstein cows. The numbers in the tables are the measured means with the values in parentheses representing the standard deviation.

Table 3.5. Field data comparison with developed cow model for hot and dry conditions (Ave ± SD)

	$T_{db,ave}$ (°C)	THI	SR ($g\ m^{-2}\ hr^{-1}$)	RR (BPM)	T_{core} (°C)	T_{skin} (°C)
Field Study*	35.1	79.6	173.6 (123.2)	95.8 (15)	39.4 (0.5)	36.5 (0.7)
Model (WI)	35.1 (0.9)	83.4 (0.5)	138.5 (8.6)	120 (0)	39.6 (0.3)	36.6 (0.4)
Model (CA)	35.1 (3.5)	79.8 (3.0)	172 (17)	117.2 (6.2)	39.4 (0.1)	36.8 (0.3)

*Gebremedhin et al. (2010)

Table 3.6. Field data comparison with developed cow model for hot and humid conditions (Ave ± SD)

	$T_{db,ave}$ (°C)	THI	SR ($g\ m^{-2}\ hr^{-1}$)	RR (BPM)	T_{core} (°C)	T_{skin} (°C)
Field Study*	29.1	79.6	205.7 (105.4)	71.7 (14.3)	38.8 (0.3)	33.9 (0.8)
Model (WI)	29.1 (4.0)	78.4 (4.1)	122.4 (4.2)	111.8 (20.2)	38.7 (0.3)	35.7 (0.4)
Model (CA)	29.1 (4.2)	75.7 (3.6)	151.6 (8.9)	104 (23.1)	38.6 (0.1)	35.6 (0.6)

*Gebremedhin et al. (2010)

The model's results were within one standard deviation of the measured means for the sweating rate and core body temperature for both conditions (hot and humid, hot and dry), as well as the skin temperature for the hot and dry condition. The respiration rates predicted were within two standard deviations for the hot and dry condition and within three standard deviations for the hot and humid condition, most likely as a result of the developed cow model's provision allowing the cow to reach her maximum respiration rate (120 BPM) before heat is allowed to be stored. The skin temperature for the hot and humid condition were also within three standard deviations of the field data's measured means. Overall however, the developed cow model performed very well and was deemed suitable for evaluating dairy heat stress mitigation across the U.S. using TMY3 Class 1 data as inputs to the barn model; the focus of Chapters 4 and 5.

CHAPTER 4. EVALUATING HEAT STRESS MITIGATION WITH THE DEVELOPED BARN AND COW MODELS

The following chapter presents multiple graphs comparing the predicted results from the model against published data (Albright, 1990). The model was ran for a station in Eau Claire, Wisconsin (SN: 726435) and Fresno, California (SN: 723890). All indication of either state throughout this chapter refers to the results from the model ran at these two stations.

4.1 Environment Evaluated

In order to analyze the developed barn and cow models further, it is important to have an understanding of the difference in predicted cow performance with the various mitigation strategies analyzed with this project. To accomplish this objective, environmental parameters from Wisconsin and California were first evaluated before the barn and cow models were applied to the remaining 213 TMY3 Class 1 stations; the topic of Chapter 5. Figures 4.1 and 4.2 show the general temperature profile trend for Wisconsin and California, respectively, with Figure 4.3 showing a direct comparison of the frequency of the outdoor air temperature distribution for each state.

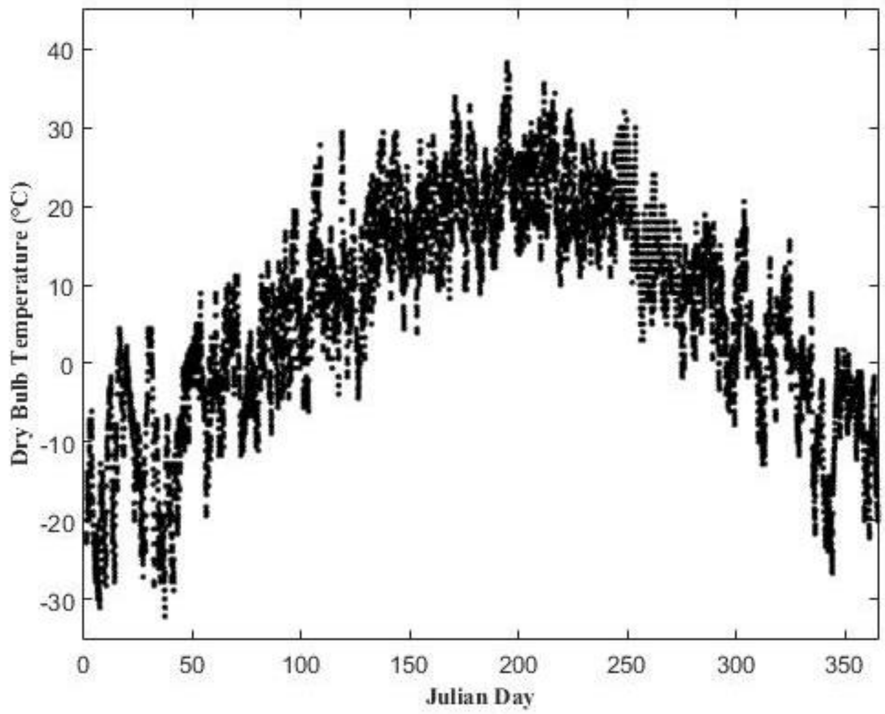


Figure 4.1. Yearly temperature profile for Wisconsin

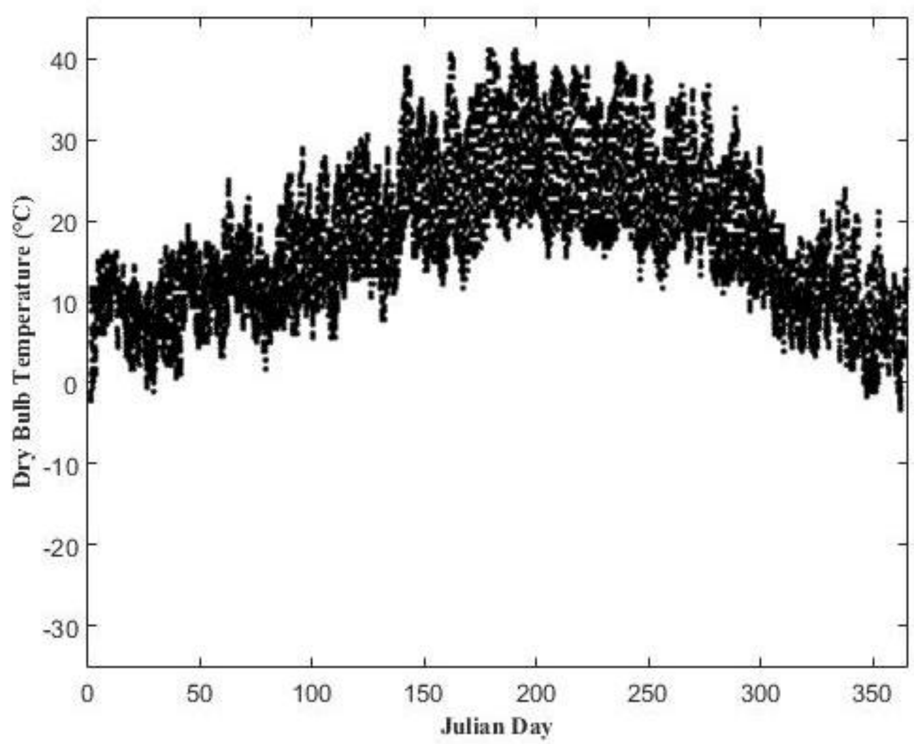


Figure 4.2. Yearly temperature profile for California

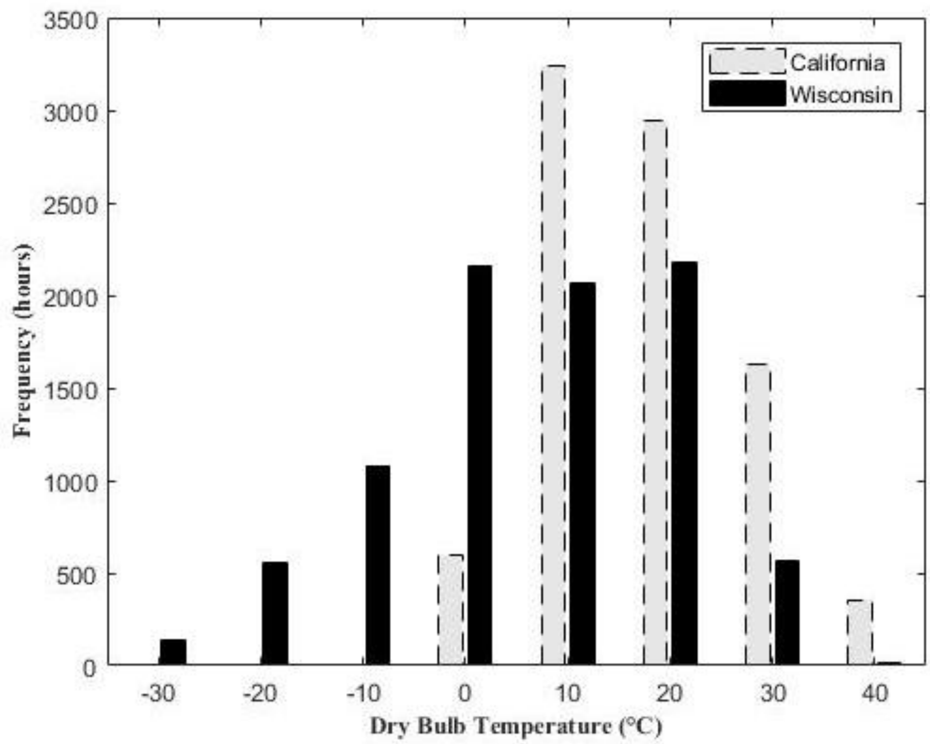


Figure 4.3. Temperature frequency distribution between Wisconsin and California

As expected, California's average yearly temperature (17.9±9.2°C) is higher than Wisconsin's average yearly temperature (6.5±3.7°C). California also has a higher frequency of hours where the temperature is greater than 20°C (4924 hours, 205.1 days) compared to Wisconsin (2756 hours, 114.8 days). A temperature of 20°C is the target temperature modeled for maximizing the ventilation rate in the developed barn model; therefore, the ventilation system in California is running 2168 more hours (90.3 days) at the maximum ventilation rate than Wisconsin.

Since THI is a better assessment of the surrounding environment than just dry-bulb temperature, THI (Equation 2.7) was also plotted for Wisconsin and California. Figures 4.4 and 4.5 show the general THI profile trend for each state with Figure 4.6 showing a direct comparison of the frequency of the THI distribution for each state.

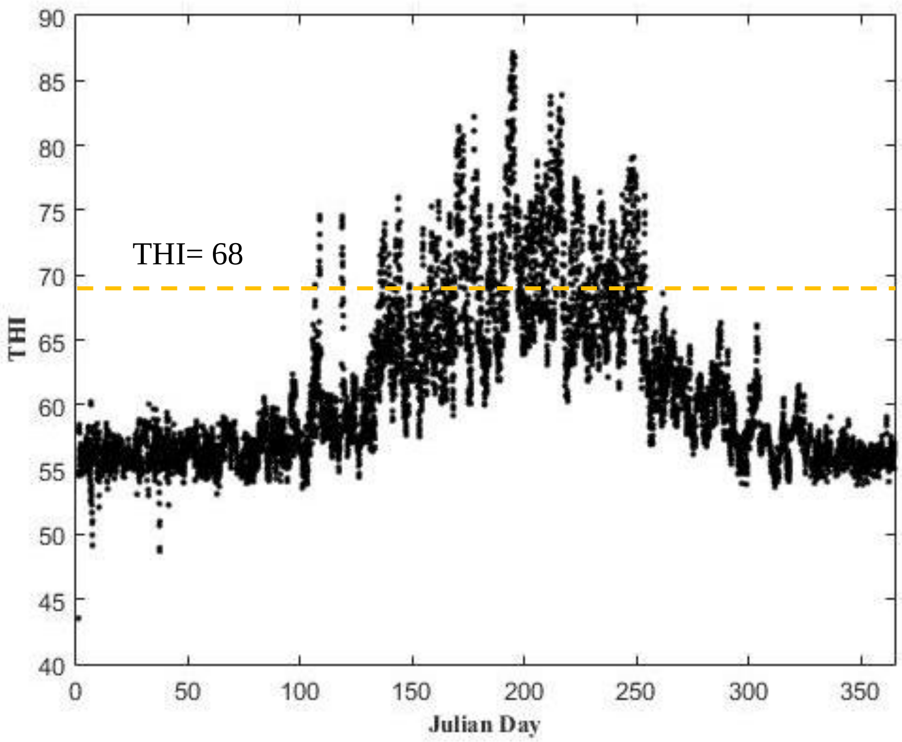


Figure 4.4. Yearly THI profile for Wisconsin

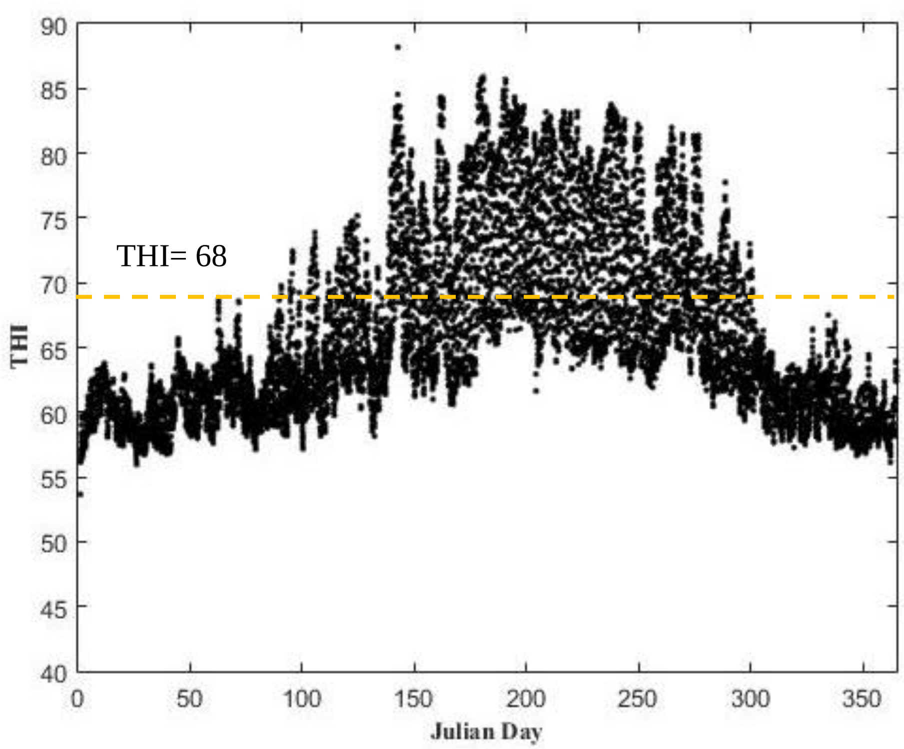


Figure 4.5. Yearly THI profile for California

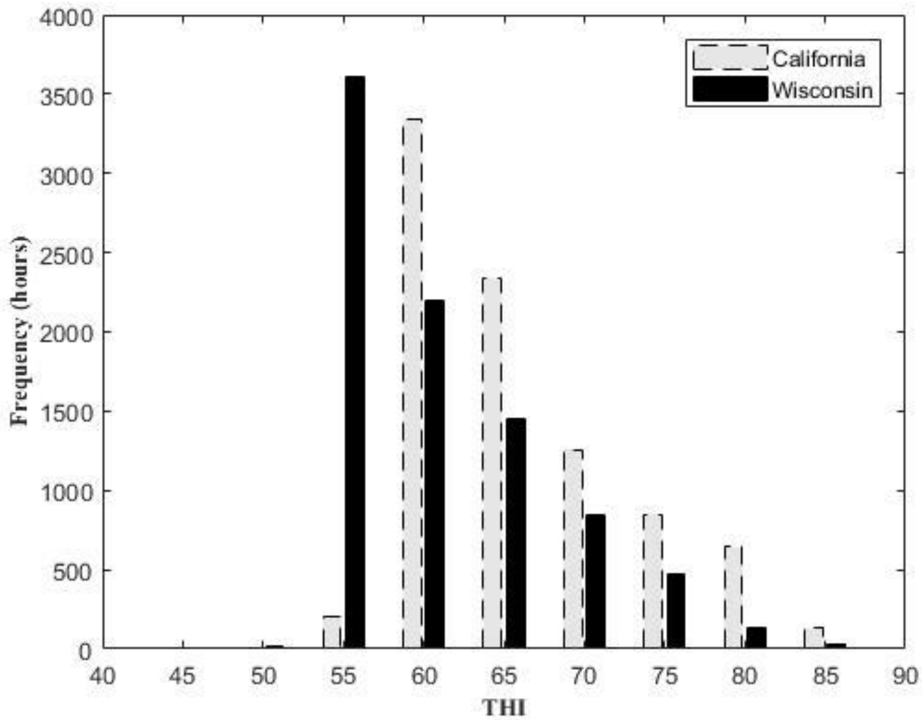


Figure 4.6. THI frequency distribution between Wisconsin and California

The average yearly THI is also higher for California (65.8±6.8) than Wisconsin (61.2±6.3) as well as the peak one-hour THI value for each state (88.2, 87.2 respectively). When the THI is greater than 68 (the generally accepted onset of heat stress), the frequency distribution is also higher for California (2880 hours, 120 days) compared with Wisconsin (1482 hours, 61.75 days). Figure 4.7 gives a better comparison of the THI hours between the two states with and without the use of an evaporative cooler for conditions present at the barn center surrounding the modeled cow, including THI hours above 68.

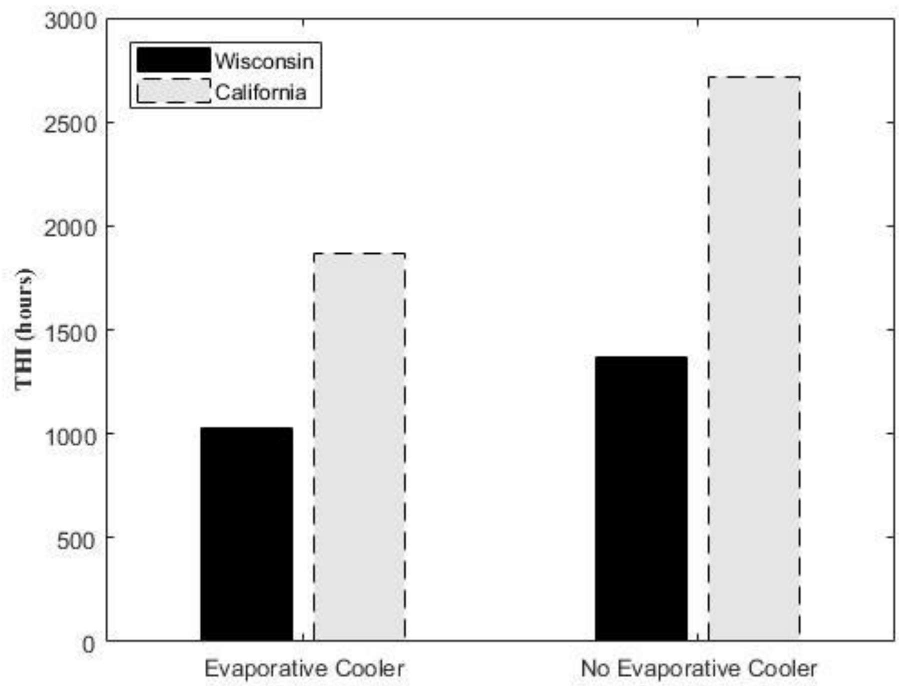


Figure 4.7. The THI hours for the onset of heat stress (THI => 68)

Having the evaporative cooler on decreases the THI hours for the onset of heat stress (THI=68) for both Wisconsin and California. Turning on the evaporative cooler decreases the number of hours where the THI is greater than 68 for Wisconsin from 1372 hours to 1029 hours, but having the evaporative cooler on in California has a major impact on decreasing the number of hours where the THI is greater than 68 (2718 hours to 1870 hours). Overall, California has more heat stress hours as indicated by the THI index than Wisconsin, which indicates a hotter or hotter and more humid environment.

4.2 Evaluating the Developed Cow Model

This section uses the previously summarized Wisconsin and California TMY3 data to investigate the predicted cow energy exchange derived from the relations outlined in Chapter 3. Figures 4.8-4.11 show the comparison between the sensible and latent heat loss predicted by the developed cow and barn models with published data for the summer months (June through August; Julian Days 152 to 244) in Wisconsin and California without and with evaporative cooling. The cow model predictions were derived using the developed barn model at the LVC ventilation case (see Table 3.2) consistent with the calorimetry conditions used for the published data (Albright, 1990; Appendix 5-1). It should be noted that in all non-evaporative cooling results, the developed cow model allows for added sprinkling if the surrounding environment is capable of evaporating excess "sweat". As a consequence of this modeling provision, the "with evaporative cooling" cases are more representative of the calorimetry conditions used for comparison.

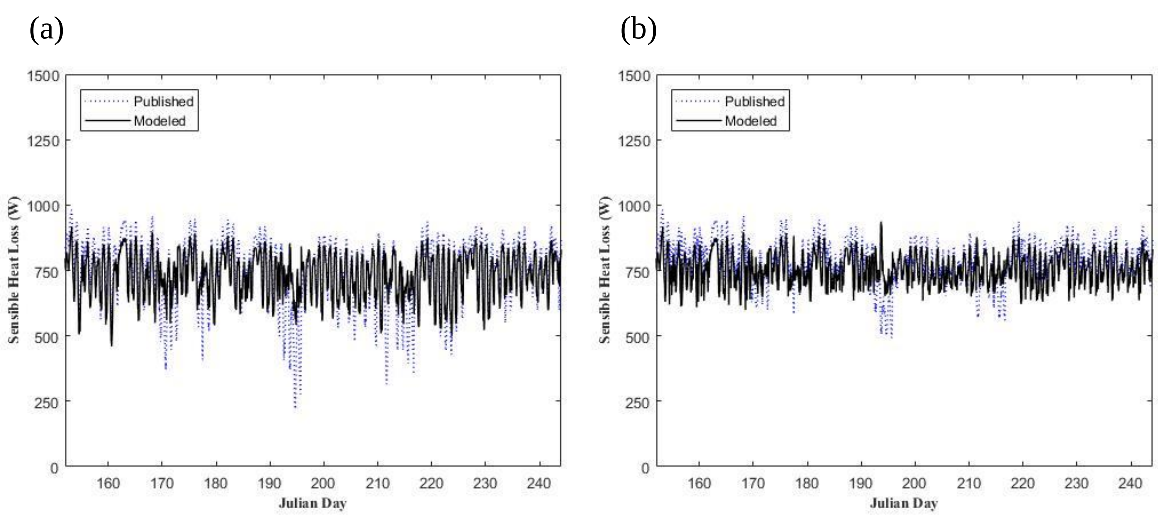


Figure 4.8. Sensible heat loss in Wisconsin (a) without evaporative cooling and (b) with evaporative cooling

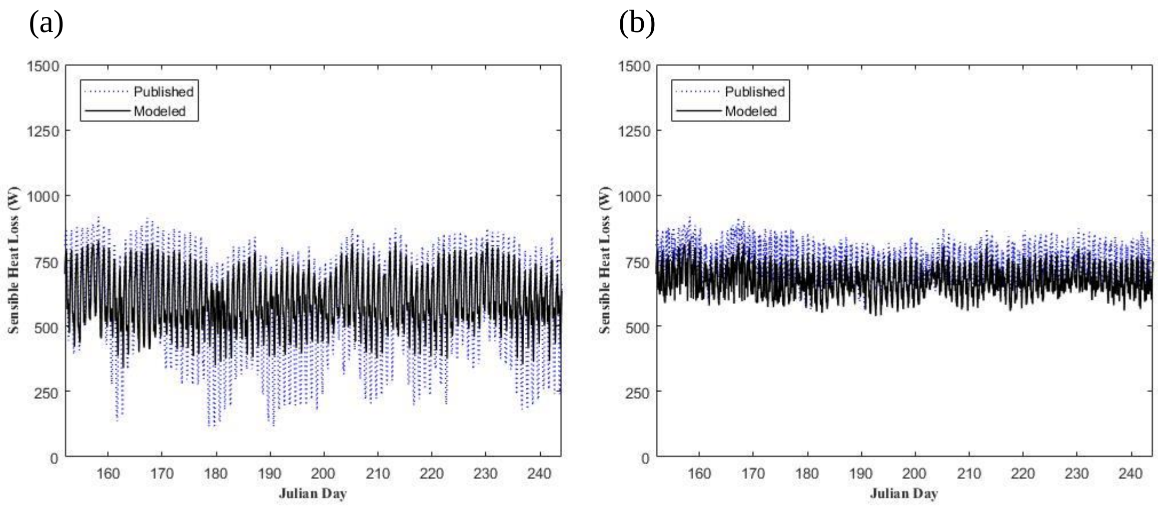


Figure 4.9. Sensible heat loss in California (a) without evaporative cooling and (b) with evaporative cooling

The predicted sensible heat loss using the developed barn and cow models without evaporative cooling for Wisconsin was 732±85 W and with evaporative cooling was 757±58 W compared to 745±130 W and 787±82 W respectively for the published data (JD 152-244). The predicted sensible heat loss for California without evaporative cooling was 598±104 W and with evaporative cooling was 677±55 W, compared to 573±196 W and 745±70 W respectively for published data (JD 152-244). The average sensible heat loss in Wisconsin for both conditions was higher than California, a result of the higher average temperature in California over this same period (27.2±6.6°C) compared to Wisconsin (20.9±5.3°C). The sensible heat loss increased for both states when evaporative cooling was added because of the decrease in barn center temperature, thus increasing the skin-to-ambient temperature gradient. The evaporative cooler decreased the ambient air's temperature enough to allow the cow to give off more sensible heat. Figures 4.10 and 4.11

show the latent heat loss for both states without and with evaporative cooling through the summer months as well.

(a) (b)

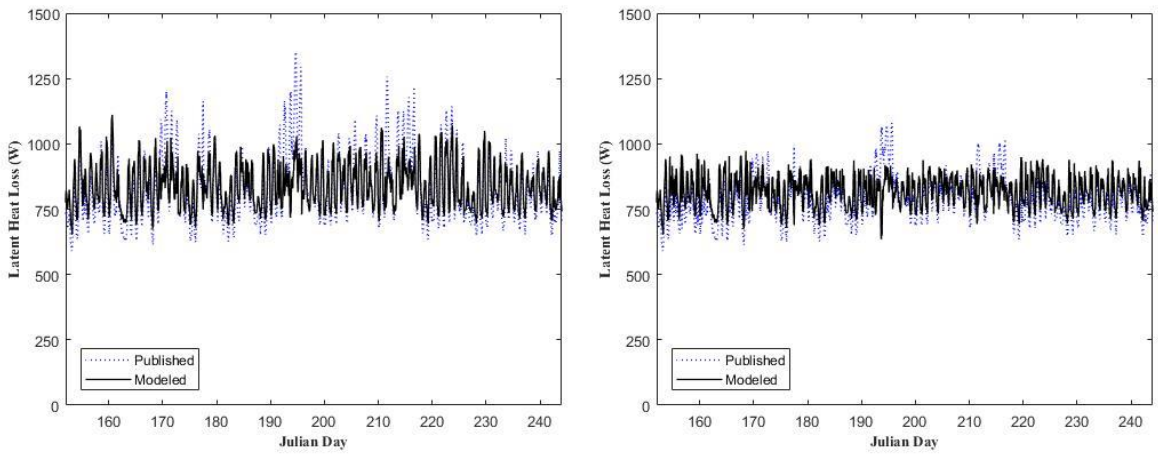


Figure 4.10. Latent heat loss in Wisconsin (a) without evaporative cooling and (b) with evaporative cooling

(a) (b)

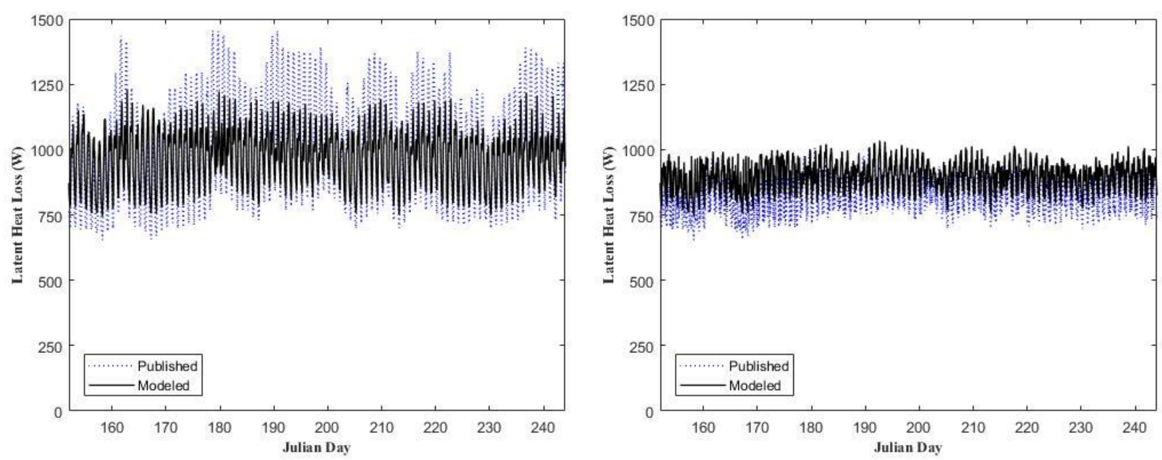


Figure 4.11. Latent heat loss in California (a) without evaporative cooling and (b) with evaporative cooling

The model also predicted latent heat loss for Wisconsin without and with evaporative cooling as 839±85 W and 814±58 W compared to 827±130 W and 785±82 W respectively for the published data (JD 152-244). The predicted latent heat loss for California without and with evaporative cooling was 973±104 W and 895±55 W respectively, compared to

999±196 W and 826±70 W respectively for published data (JD 152-244). The average latent heat loss in Wisconsin for both conditions was lower on average than California, indicating, at least in part, that the humidity ratio in Wisconsin is higher than California over this same period. The latent heat loss decreased for both states when evaporative cooling was added due to the increase in moisture in the air from the evaporative cooler, thus decreasing the vapor gradient between the cow and her environment.

A Welch's (unequal variances) t-test was performed to verify the agreement between predicted and published sensible and latent heat loss. Sixteen data points were randomly selected using the random number function in Microsoft Excel. First, a row number was randomly selected, which was an hour from one of the 92 days in the summer (Julian Days 152-244). This was the beginning hour of one data point. Then, the random number function was used to choose a continuous number of averaging hours between 3 and 8 inclusive. The comparisons were made at the LVCE condition to best match published calorimetry conditions (i.e., no sprinkler use). A two-tailed test was used with a significance level of 0.05 and 30 degrees of freedom ($|t|=2.042$). The results indicated that for the Wisconsin data comparison, no significant difference was found between predicted and published sensible and latent heat loss data ($|t|=0.97$). For California data; however, significant differences were found for both sensible and latent heat loss comparisons ($|t|=3.75$). Reviewing the percent differences for the sixteen randomly selected Wisconsin and California data set at the LVCE strategy, the sensible heat loss was -2.6% for Wisconsin and -10.1% for California compared to the latent heat loss of 2.7% for Wisconsin and 9.4% for California. This discrepancy between the Wisconsin and California results could be due to the drier conditions in California since the average relative humidity for California was 65.7±9.5% compared to the Wisconsin data set average relative humidity of 82.3±9.5%. The environmental conditions at which the published data was collected is unknown; therefore, the assumption could be made that Wisconsin's environmental conditions most closely matches that of the published data's environmental conditions. In summary, since Wisconsin's data set was not significantly different than the published data set, the model was deemed valid and thus acceptable for use.

As outlined in Chapter 3, sensible heat loss can take several paths including heat lost due to ingested feed and water, and sensible heat lost due to respiration, convection, and radiation. Figures 4.12-4.15 show a comparison of the amount of heat lost through each path for Wisconsin and California, comparing the LVCN, LVCE, HVCN and HVCE ventilation cases (Table 3.2; LVC≈0.5 m s^{-1} cow-level airspeed, HVC≈2 m s^{-1}, N=without evaporative cooling, E=with evaporative cooling).

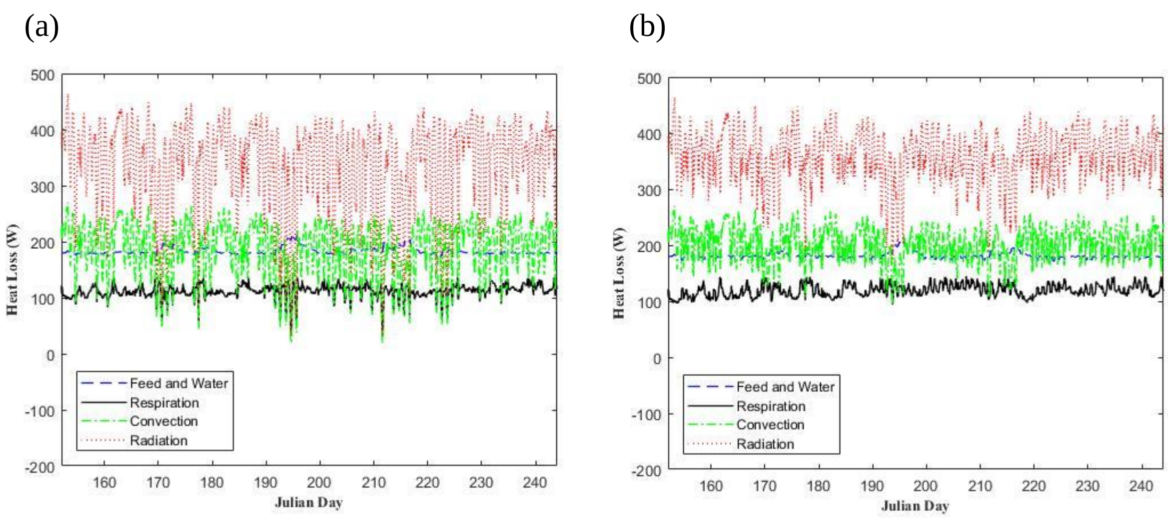


Figure 4.12. Sensible heat loss partitions in Wisconsin for (a) LVCN and (b) LVCE

(a) (b)

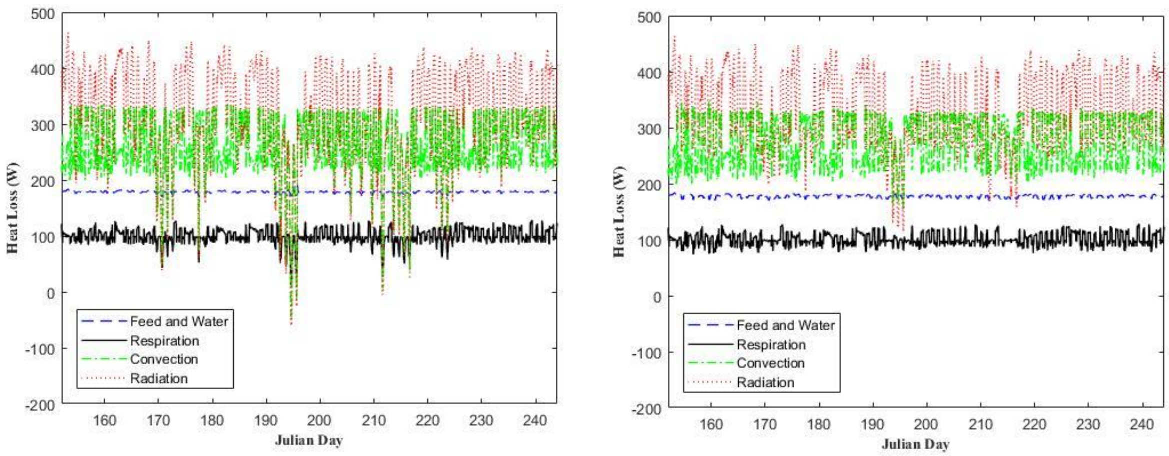


Figure 4.13 Sensible heat loss partitions in Wisconsin for (a) HVCN and (b) HVCE

(a) (b)

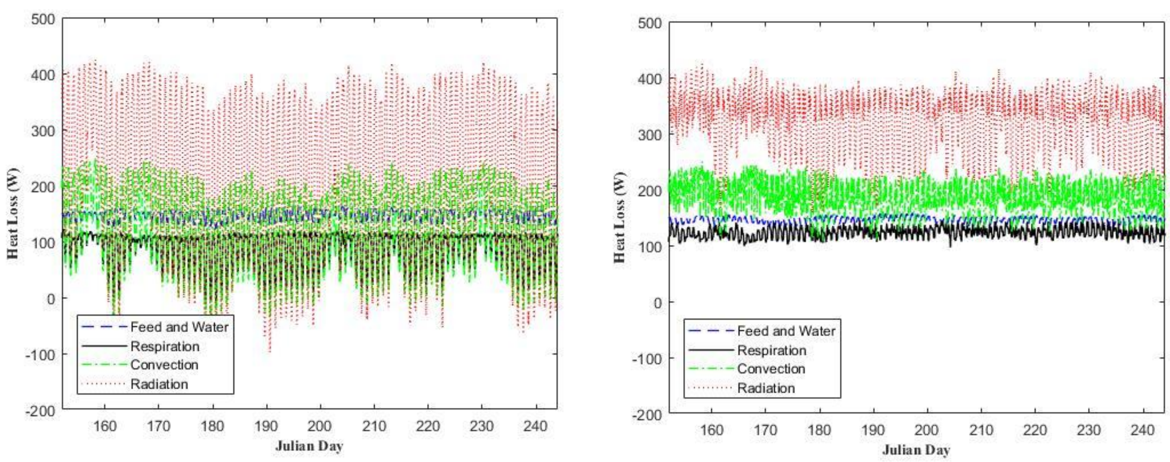


Figure 4.14. Sensible heat loss partitions in California for (a) LVCN and (b) LVCE

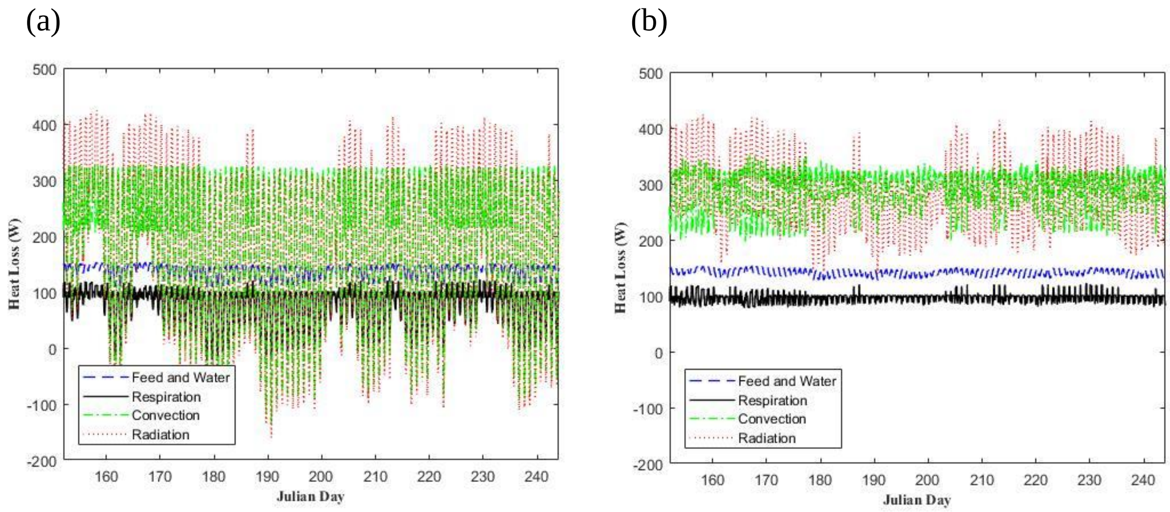


Figure 4.15. Sensible heat loss partitions in California for (a) HVCN and (b) HVCE

The type of ventilation strategy used impacts the amount of sensible heat released through each path. Incorporating evaporative cooling increases the amount of sensible heat released through convection and radiation while the sensible heat loss through feed and water ingestion and respiration stays nearly constant. A negative value for these graphs indicates that the heat “loss” component is in fact a heat gain for the cow, potentially resulting in the need for her to store undissipated heat. For the LVCN and LVCE cases in Wisconsin, the sensible heat lost through feed and water ingestion was 183±6 W and 181±5 W respectively and for respiration was 111±13 W and 119±11 W. Sensible heat loss through convection for the same cooling strategies in Wisconsin was 185±50 W and 203±30 W, and for radiation was 336±77 W and 359±46 W. For California at the LVCN and LVCE strategies, the same pattern followed. The sensible heat lost through feed and water ingestion was 148±9 W and 145±6 W respectively and for respiration was 87±31 W and 123±9 W. Sensible heat loss through convection for the same cooling strategies in

California was 120±74 W and 186±29 W, and for radiation was 227±138 W and 321±53 W.

Convection and radiation are very dependent on ambient temperature, interior barn surface temperature, and airspeed conditions, resulting in more variability than sensible heat lost through respiration and the heat required for ingested feed and water. As the average barn airspeed increases between the LVCN and HVCN ventilation cases for both states, the convective heat loss increases. In turn, radiation heat loss will decrease as convection heat loss increases, which is mainly the result of the cow's skin temperature lowering, reducing the temperature gradient between the cow and her physical surroundings. For Wisconsin, the convective heat lost for the HVCN and HVCE strategies were predicted to be 255±56 W and 278±43 W respectively. For radiation, the sensible heat lost was predicted at 311±88 W and 325±65 W respectively. Like the LVC cases, the sensible heat lost through feed and water ingestion and respiration were nearly the same for both HVCN and HVCE cases. The sensible heat lost through feed and water ingestion was 99±14 W and 101±11 W respectively, and for respiration was 178±2 W and 178±3 W. For California, the convective heat lost for the HVCN and HVCE strategies were 178±123 W and 293±34 W respectively. For radiation, the sensible heat lost was 188±144 W and 273±54 W respectively. Again, the sensible heat lost through feed and water ingestion and respiration were nearly the same for both HVCN and HVCE cases. The sensible heat lost through feed and water ingestion was 136±11 W and 141±6 W respectively, and for respiration was 76±33 W and 97±8 W.

Radiation has the highest impact on the cow's ability to dissipate sensible heat as shown in the previous figures with convection and feed and water ingestion following. When looking at the LVCN strategy for both states, radiation accounted for 46% of the total sensible heat loss in Wisconsin and 38% in California. Convection accounted for 25% of the total sensible heat loss for Wisconsin and 20% in California whereas feed and water ingestion accounted for 25% of the total sensible heat loss in Wisconsin and California. Finally, respiration accounted for 15% in Wisconsin and 14.5% in California.

Latent heat loss only has two paths it can take, namely latent heat lost due to respiration and the evaporation of sweat. For the following graphs, sweating predicted by the cow model is the evaporation potential of the surrounding environment. Figures 4.16-4.19 show a comparison of the amount of heat lost through each latent path for Wisconsin and California for both the LVC and HVC ventilation cases without and with evaporative cooling.

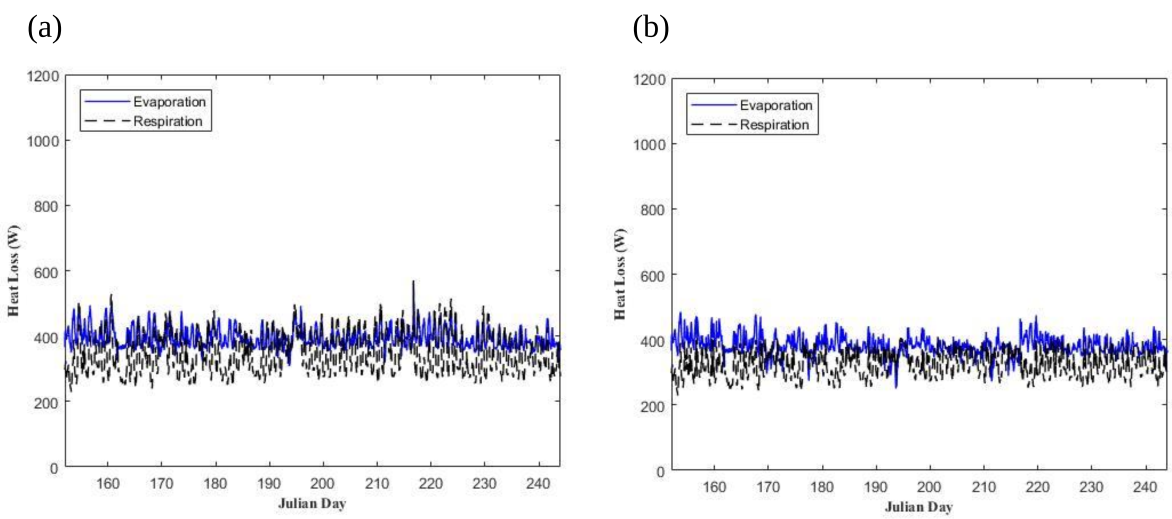


Figure 4.16. Latent heat loss partitions in Wisconsin for (a) LVCN and (b) LVCE

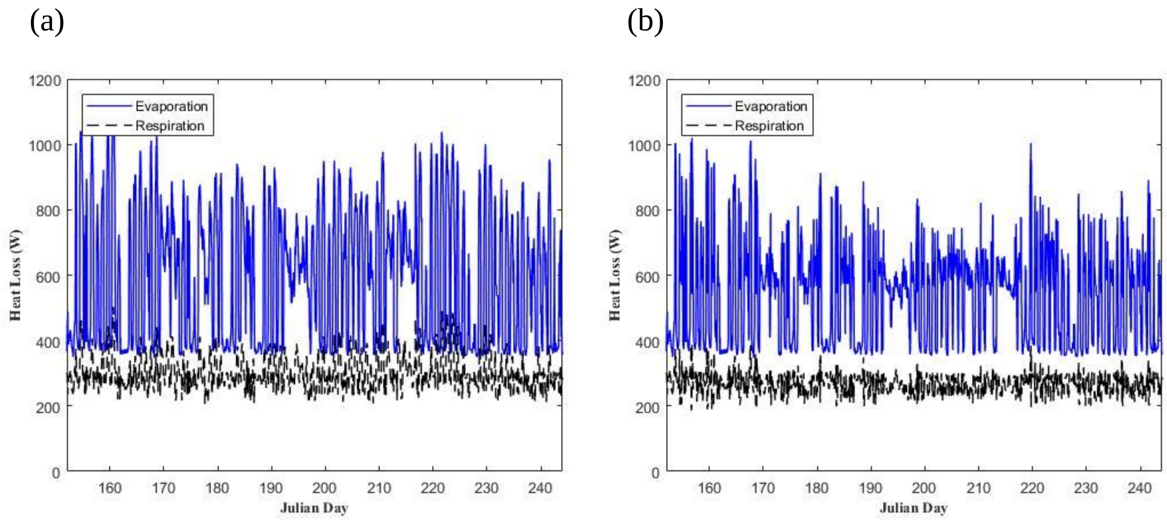


Figure 4.17. Latent heat loss partitions in Wisconsin for (a) HVCN and (b) HVCE

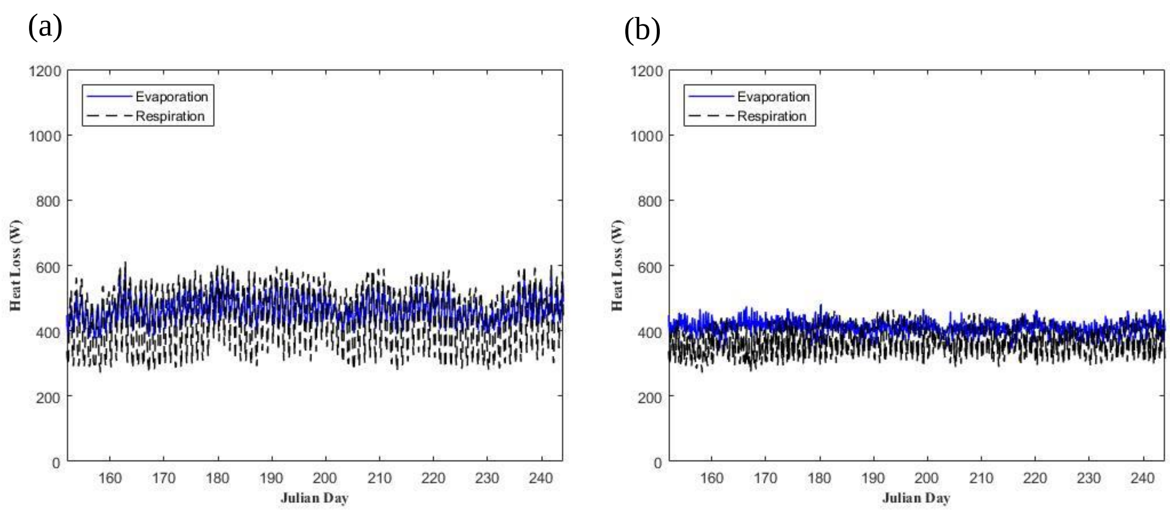


Figure 4.18. Latent heat loss partitions in California for (a) LVCN and (b) LVCE

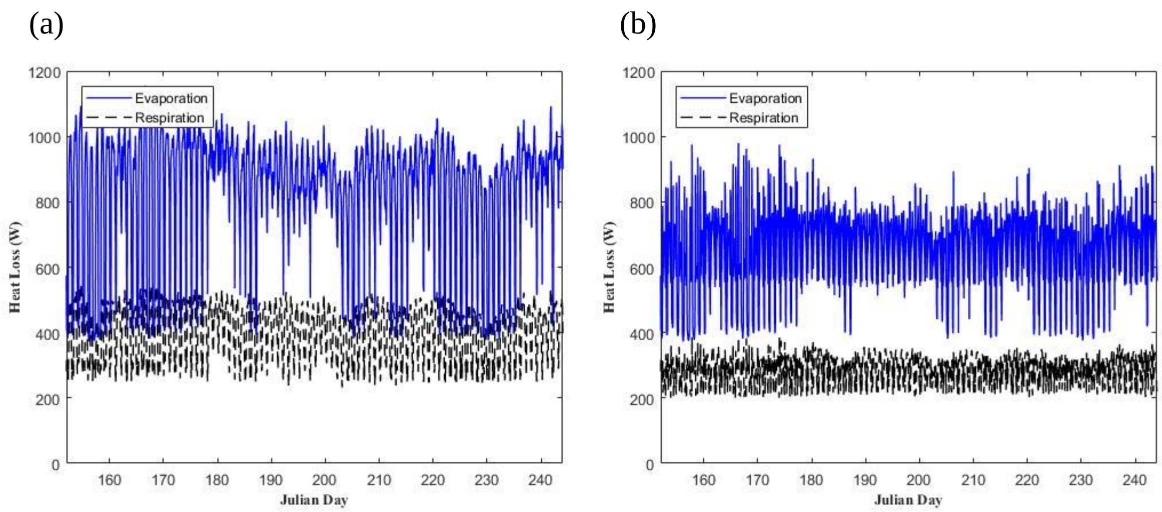


Figure 4.19. Latent heat loss partitions in California for (a) HVCN and (b) HVCE

Evaporative cooler use reduces the cow's ability to dissipate her latent heat due to the increase in moisture in the air. The respired latent heat loss stays fairly consistent for each path during the LVCN and LVCE strategies for each state. In Wisconsin, latent heat lost due to respiration was 345±63 W and 324±38 W respectively, and for evaporation was 397±31 W and 383±28 W respectively. California followed the same pattern where the latent heat lost due to respiration was 448±91 W and 371±44 W respectively, and for evaporation was 464±37 W and 409±21 W respectively. The latent heat loss due to respiration decreases and the latent heat loss due to evaporation increases as the airspeed increases from the LVC to the HVC ventilation cases. For the HVCN and HVCE strategies in Wisconsin, latent heat lost due to respiration was 303±52 W and 268±30 W respectively, and for evaporation was 598±205 W and 533±145 W respectively. California's latent heat lost due to respiration was 400±82 W and 268±41 W respectively, and for evaporation was 833±187 W and 653±120 W respectively.

Respiration and evaporation dissipate latent heat almost equally in a low airspeed environment (LVCN or LVCE). When looking at the LVCN strategy for both states, respiration accounts for 41% of the total latent heat loss for Wisconsin and 46% in California. Evaporation accounts for 47% of the total latent heat loss for Wisconsin and 48% in California.

For the developed cow model, the sweating rate directly correlates to the evaporation rate, which is dependent on the moist air properties and airspeed at the modeled cow located at the center of the barn. Therefore, the cow could sweat more than the environment can handle, but the sweat would pool on the skin surface providing no cooling benefit due to evaporation. The modeled sweating rate (i.e., evaporation rate) of the cow was compared to a published equation which is dependent on the cow's skin temperature (Equation 4.1) that was developed by Thompson, Fadel, et al., (2011):

$$SR_{published} = 0.75 * e^{0.15(T_{skin})}. \tag{4.1}$$

This correlation between skin temperature and sweating rate was developed by applying a best fit line to sweating rate data from 12 different studies, which was then evaluated against three previously published equations (Thompson, Fadel, et al., 2011). The comparison for the modeled cow's sweating (i.e., evaporation) rate versus the published sweating rate in Wisconsin and California for the LVC and HVC ventilation cases can be seen in Figures 4.20 – 4.23 without and with evaporative cooling.

(a) (b)

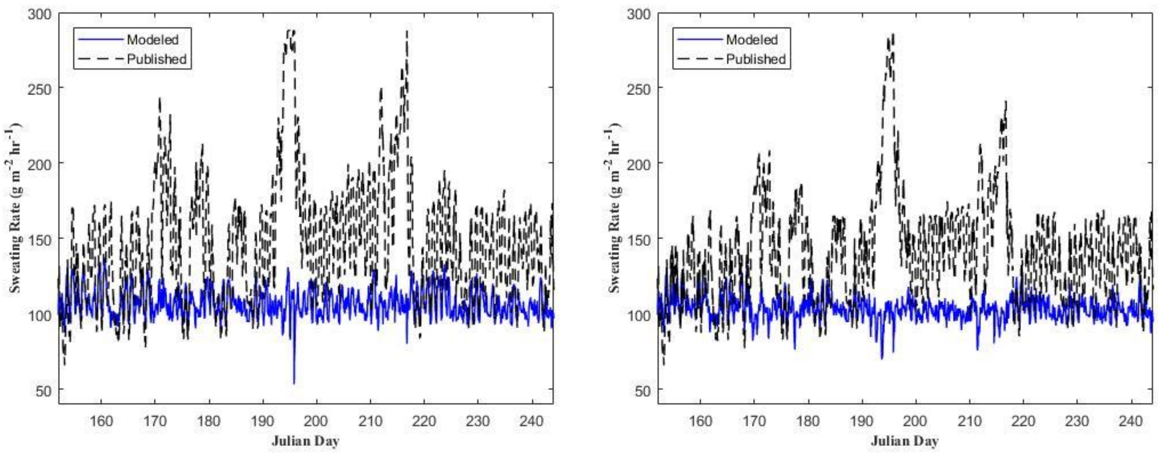


Figure 4.20. Sweating rate comparison in Wisconsin for (a) LVCN and (b) LVCE

(a) (b)

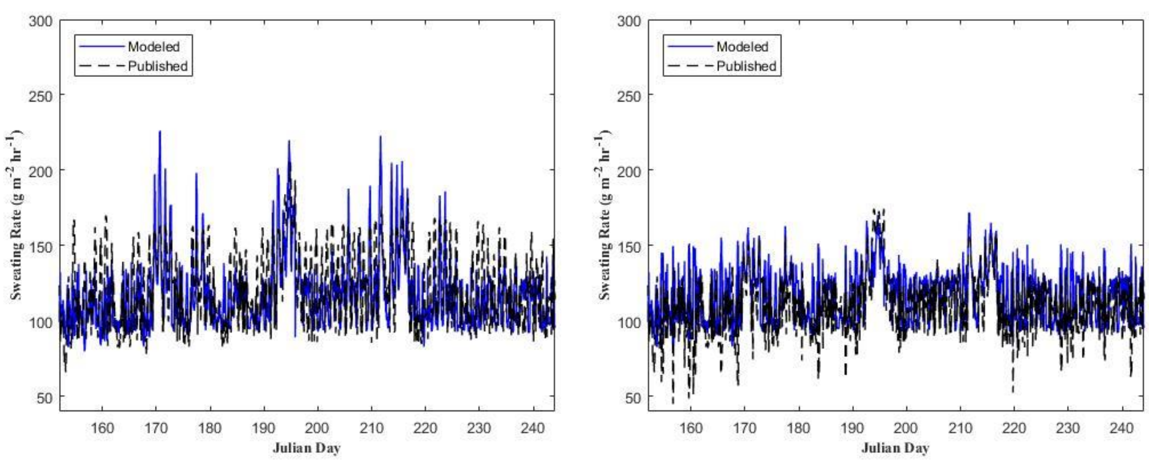


Figure 4.21. Sweating rate comparison in Wisconsin for (a) HVCN and (b) HVCE

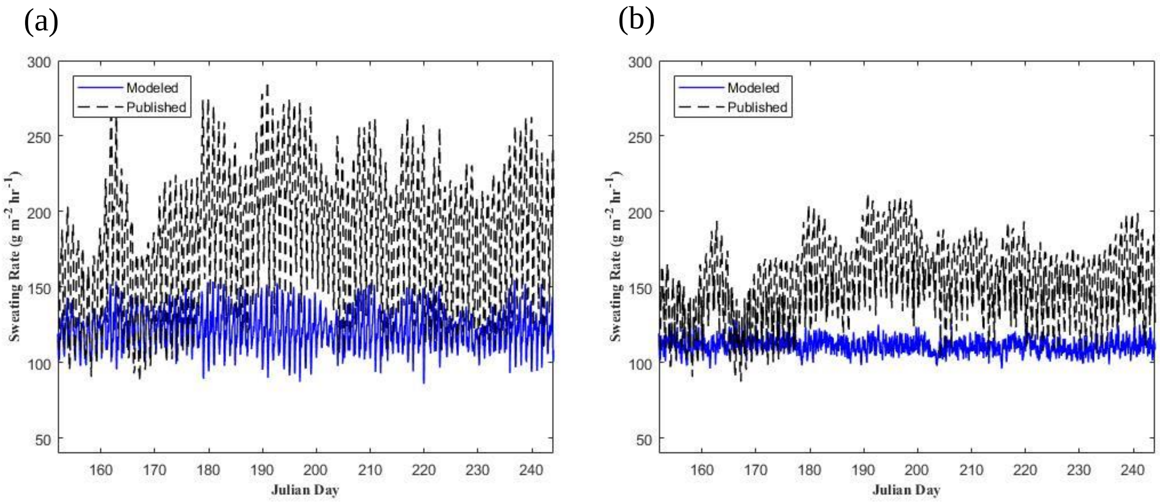


Figure 4.22. Sweating rate comparison in California for (a) LVCN and (b) LVCE

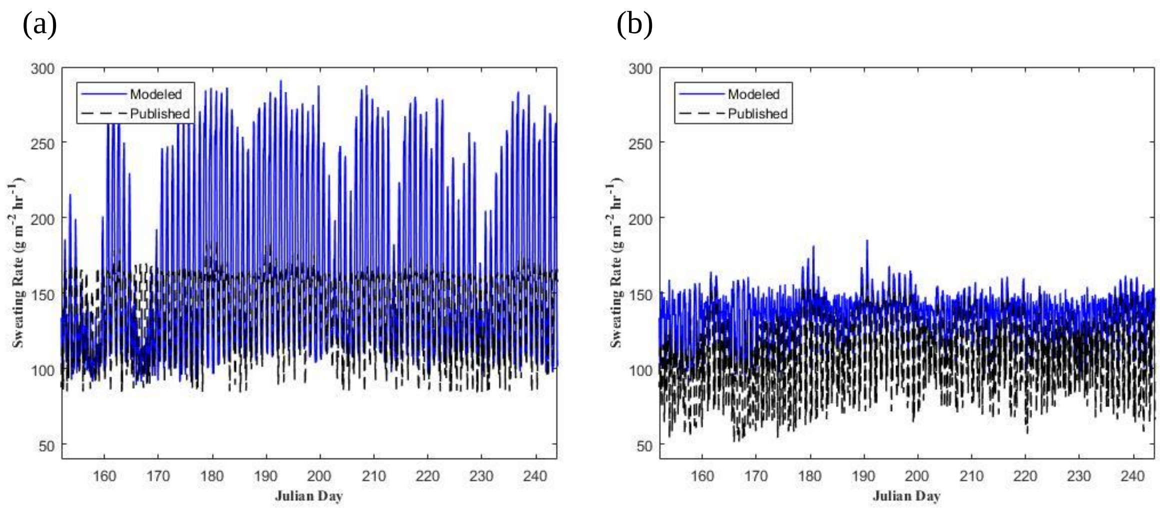


Figure 4.23. Sweating rate comparison in California for (a) HVCN and (b) HVCE

The published values are in general higher than the modeled results during the LVC ventilation case. For Wisconsin at the LVCN and LVCE strategies, the published sweating rates were 148±43 g m^{-2} hr^{-1} and 141±36 g m^{-2} hr^{-1} respectively, and for the modeled sweating (i.e., evaporation) rates were 107±9 g m^{-2} hr^{-1} and 103±8 g m^{-2} hr^{-1}. For California at the LVCN and LVCE strategies, the published sweating rates were 185±43 g m^{-2} hr^{-1} and 153±26 g m^{-2} hr^{-1} respectively, and for the modeled sweating (i.e., evaporation) rates were

122±12 g m^{-2} hr^{-1} and 111±5 g m^{-2} hr^{-1}. This trend is the result of the model equating sweating rate with the actual evaporation rate versus the published sweating rate summarized with Equation 4.1. The difference between the modeled and published data (Equation 4.1) indicates, as one possible explanation, that unevaporated sweat is pooling on her skin since the environment is unable to evaporate excess sweat. The sweating rates and evaporation rates are lower during the evaporative cooling cases due to the moisture increase in the ambient air from the evaporative cooler which decreases the moisture's concentration gradient between the air and the cow's skin.

In comparison with the HVC ventilation case, the environment is capable of evaporating significantly more sweat produced by the cow, which is one potential reason why the agreement between modeled and published is better as shown in Figures 4.21 and 4.23. For Wisconsin at the HVCN and HVCE strategies, the published sweating rates were 123±25 g m^{-2} hr^{-1} and 111±19 g m^{-2} hr^{-1} respectively, and for the modeled sweating (i.e., evaporation) rates were 117±23 g m^{-2} hr^{-1} and 117±18 g m^{-2} hr^{-1}. For California at the HVCN and HVCE strategies, the published sweating rates were 143±26 g m^{-2} hr^{-1} and 111±24 g m^{-2} hr^{-1} respectively, and for the modeled sweating (i.e., evaporation) rates were 160±59 g m^{-2} hr^{-1} and 134±16 g m^{-2} hr^{-1}. When the modeled evaporation rate exceeds the published sweating rate the difference shows how much extra water could be added via sprinkling to cool the cow.

4.2.1 One week analysis

A week in July (July 16^{th} – 22^{nd}; Julian Days 197 to 203) was also randomly chosen to further evaluate the model in more detail. Figures 4.24 and 4.25 show the outdoor temperature profile as well as the roof and mean radiant temperature profile for Wisconsin and California at the LVCN and LVCE conditions.

(a) (b)

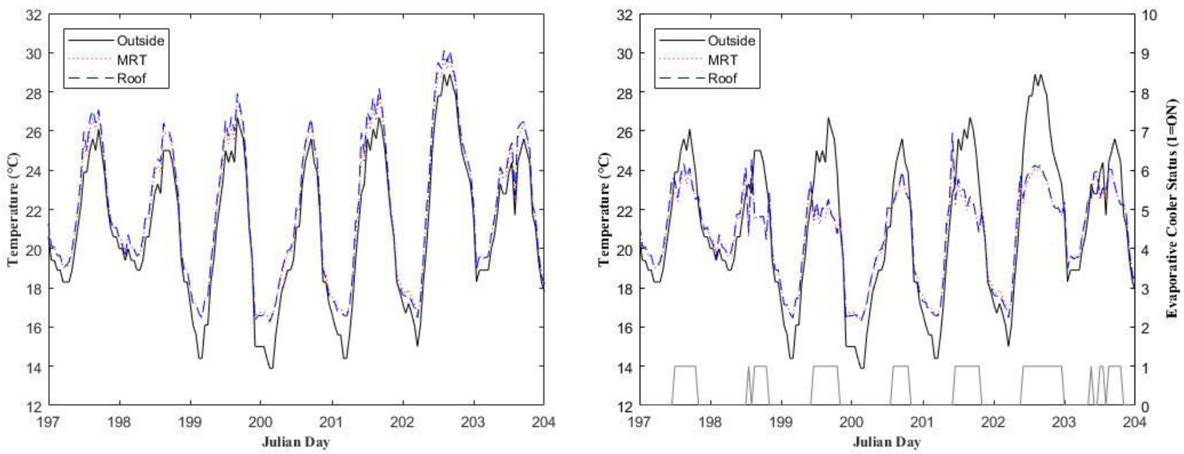


Figure 4.24. Temperature profiles in Wisconsin for (a) LVCN and (b) LVCE

(a) (b)

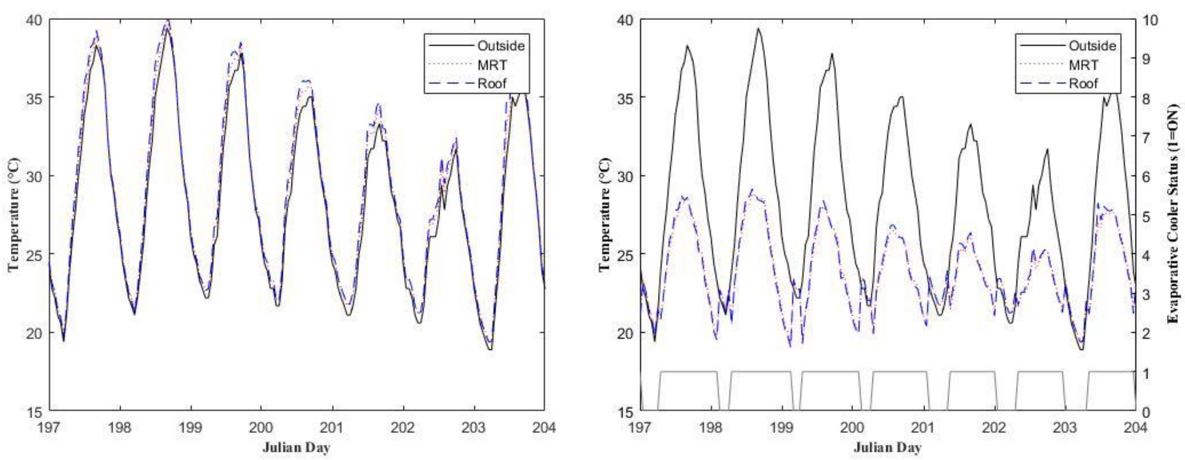


Figure 4.25. Temperature profiles in California for (a) LVCN and (b) LVCE

The roof temperature is the inside temperature of the roof facing the housed cows, and the mean radiant temperature (MRT) is the equivalent temperature that the cows are exposed to radiatively from their surroundings. When evaporative cooling is implemented (shown by the secondary y-axis), the roof and mean radiant temperature decrease due to the decreased temperature inside the barn. For both conditions in Wisconsin (LVCN, LVCE), the average outdoor temperature over the 7-day period was 21±3.7 °C. The MRT for the LVCN strategy was 22.0±3.6 °C and the roof temperature was 22.1±3.8 °C while for the LVCE strategy the MRT and roof temperatures were 20.7±2.2 °C and 20.8±2.4 °C respectively. For both conditions in California, the average outdoor temperature over the 7-day period was 28±5.4 °C. The MRT for the LVCN strategy was 28.8±5.5 °C and the roof temperature was 29.0±5.6 °C while for the LVCE strategy the MRT and roof temperatures were 23.9±2.5 °C and 23.9±2.6 °C respectively. The MRT and roof temperatures decreased when evaporative cooling was implemented due to the decrease in in-barn air temperature. The developed barn model also accounts for heat build-up through the barn in the direction of airflow, which is shown in Figures 4.26 and 4.27 for both states.

(a) (b)

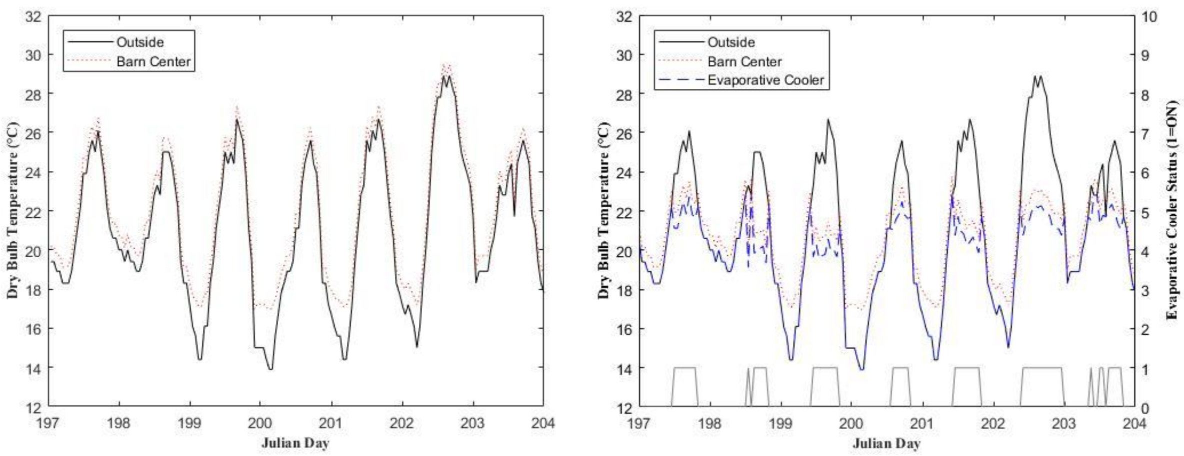


Figure 4.26. Temperature profiles in Wisconsin for (a) LVCN and (b) LVCE

(a) (b)

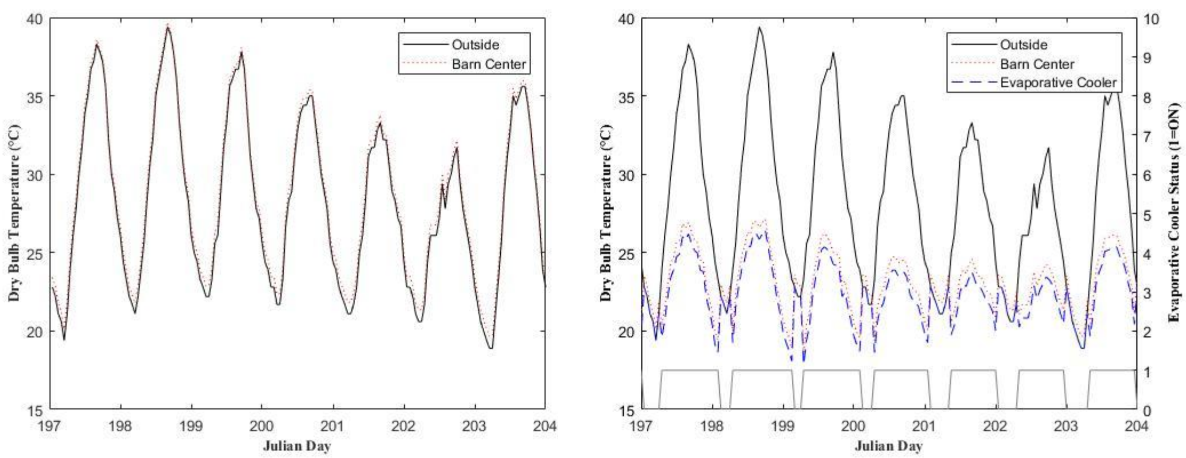


Figure 4.27. Temperature profiles in California for (a) LVCN and (b) LVCE

After moist air enters the barn, it begins to pick up sensible heat given off by the cattle as it moves axially through the barn. The barn center denotes the temperature at the center of the barn, and for the LVCE strategy, the evaporative cooler temperature is the air temperature just after it passes through the evaporative cooler. For the LVCN strategy in Wisconsin, the average barn center temperature reaches 22±3.4 °C. When the evaporative cooler is turned on (LVCE), it first decreases the air temperature to an average of 19.6±2.3 °C and then rises to an average barn center temperature of 20.6±1.9 °C. For the LVCN strategy in California, the average barn center temperature reaches 28.7±5.3 °C. When the evaporative cooler is turned on (LVCE), it first decreases the air temperature to an average of 22.3±2.0 °C and then rises to an average barn center temperature of 23.1±2.0 °C. The secondary y-axis in both Figures 4.26b and 4.27b show when the evaporative cooler is on or off. When it is off, the outdoor temperature and evaporative temperature are the same as shown in figures 4.26a and 4.27a. Figures 4.28 and 4.29 also show the airflow and airspeed profiles through the barn for Wisconsin and California for the LVCN and HVCN strategies.

(a) (b)

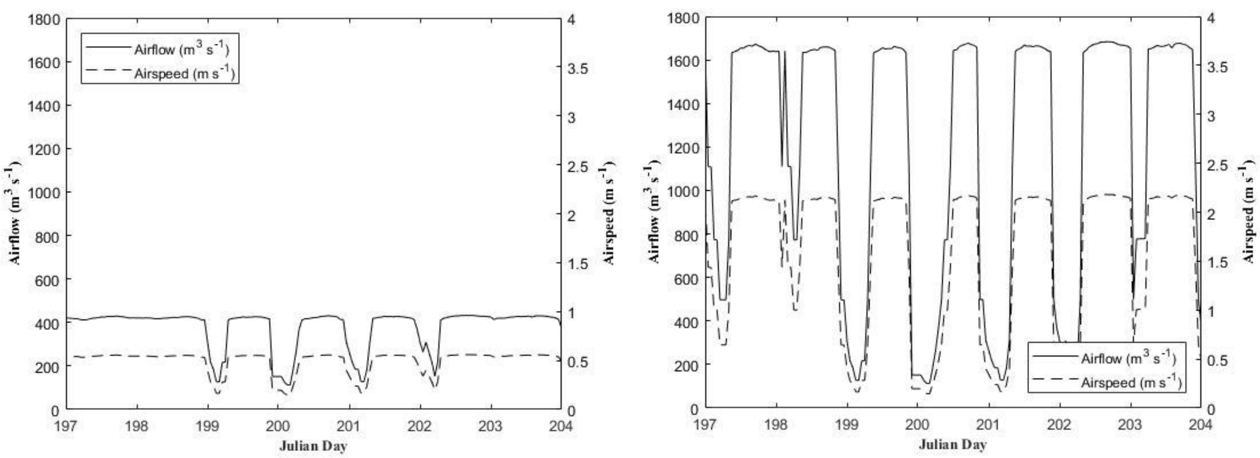


Figure 4.28. Airflow and airspeed profiles in Wisconsin for (a) LVCN and (b) HVCN

(a) (b)

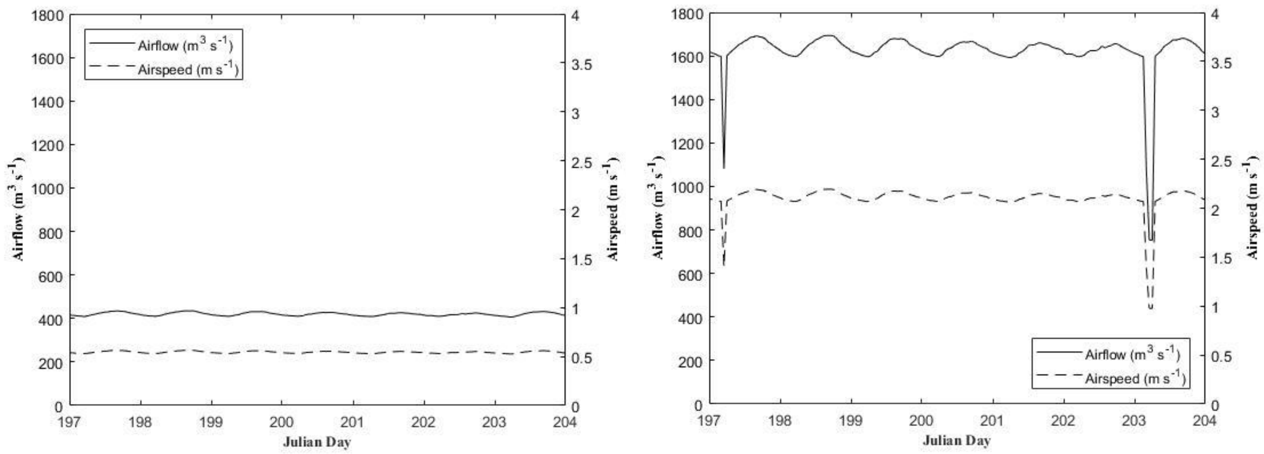


Figure 4.29. Airflow and airspeed profiles in California for (a) LVCN and (b) HVCN

For Wisconsin, the average airflow and airspeed for the LVCN strategy are 380±92 m^3 s^{-1} and 0.5±0.1 m s^{-1} respectively. For the HVCN strategy, the airflow and airspeed increase to 1170±613 m^3 s^{-1} and 1.5±0.8 m s^{-1} respectively. For California, the average airflow and airspeed for the LVCN strategy are 420±8 m^3 s^{-1} and 0.5±0.0 m s^{-1} respectively. For the HVCN strategy, the airflow and airspeed increase to 1620±116 m^3 s^{-1} and 2.1±0.2 m s^{-1} respectively. Figures 4.30-4.33 show the LVCN and LVCE comparison of sensible and latent heat loss for a week in July for both states.

(a) (b)

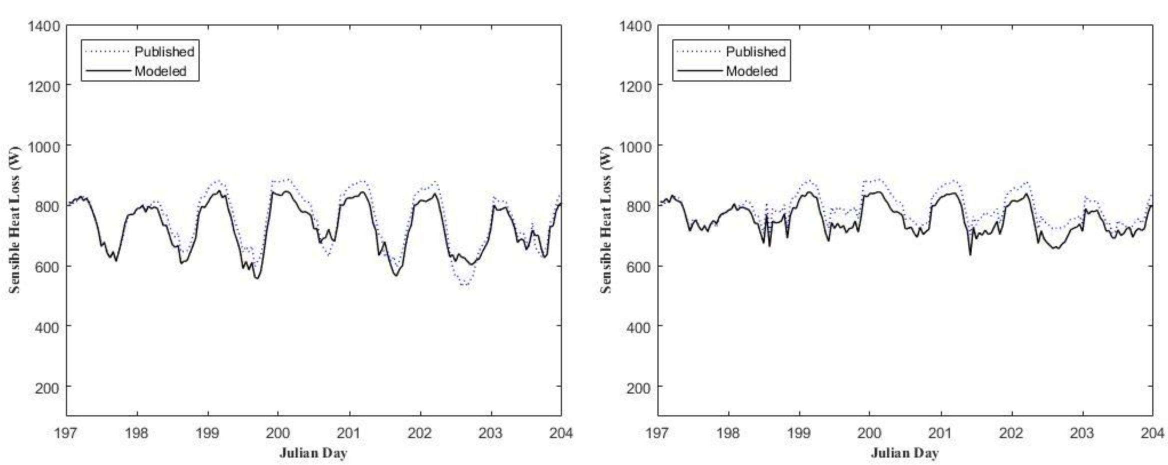


Figure 4.30. Sensible heat loss for a week in July in Wisconsin at (a) LVCN and (b) LVCE

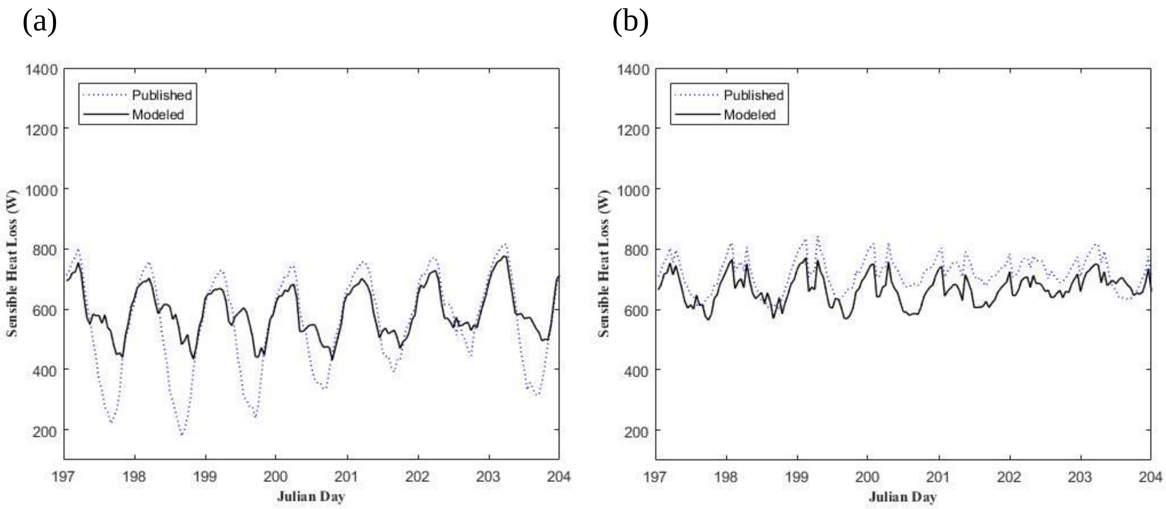


Figure 4.31. Sensible heat loss for a week in July in California at (a) LVCN and (b) LVCE

For the analyzed week, the model predicted sensible heat loss for Wisconsin without and with evaporative cooling as 729±82 W and 757±51 W respectively compared to 750±94 W and 790±51 W for the published data (JD 197-203). The predicted sensible heat loss for California without and with evaporative cooling was 590±83 W and 668±49 W respectively, compared to 547±166 W and 720±56 W respectively for published data (JD 197-203).

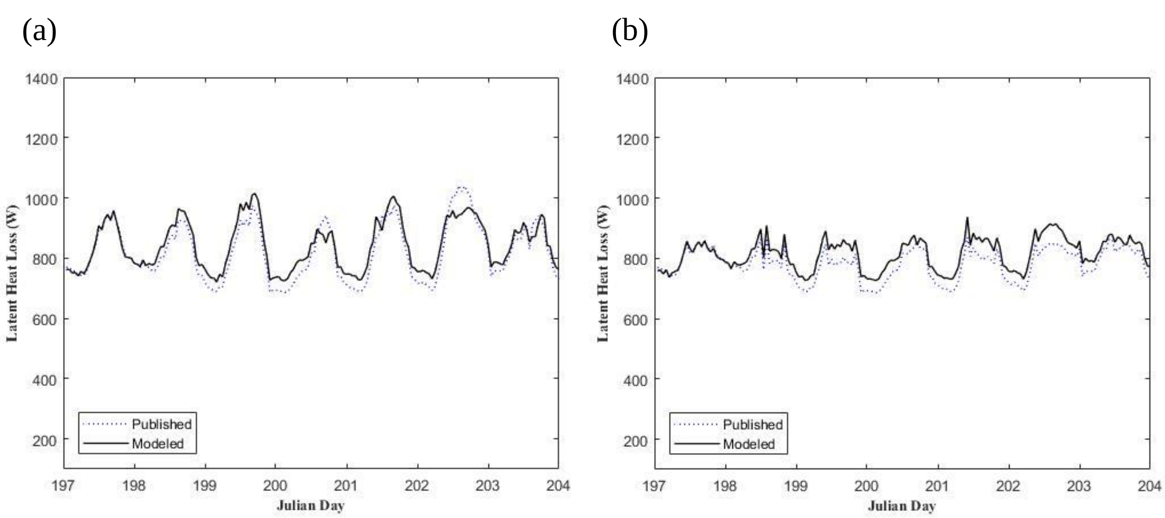


Figure 4.32. Latent heat loss for a week in July in Wisconsin at (a) LVCN and (b) LVCE

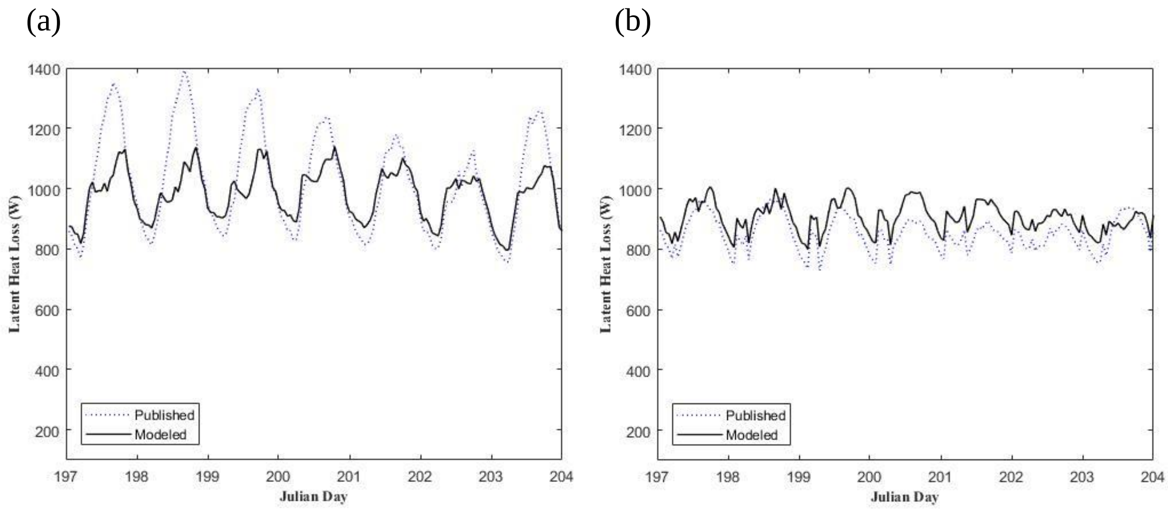


Figure 4.33. Latent heat loss for a week in July in California at (a) LVCN and (b) LVCE

The model predicted latent heat loss for Wisconsin without and with evaporative cooling as 842±82 W and 815±51 W respectively compared to 822±94 W and 781±51 W for the published data (JD 197-203). The predicted latent heat loss for California without and with evaporative cooling was 981±83 W and 904±49 W respectively, compared to 1024±166 W and 852±56 W respectively for published data (JD 197-203).

In order to more thoroughly compare the model to published data, box and whisker plots were created for both states to create a better visual of the data distribution for each ventilation strategy between states. Figures 4.34 and 4.35 show this comparison. Recall that relative to published data, the modeled cases with evaporative cooling better match the conditions under which published calorimetry data was collected. Without evaporative cooling, the developed cow model allows for added sprinkling, not accounted for in the published data used here for comparison.

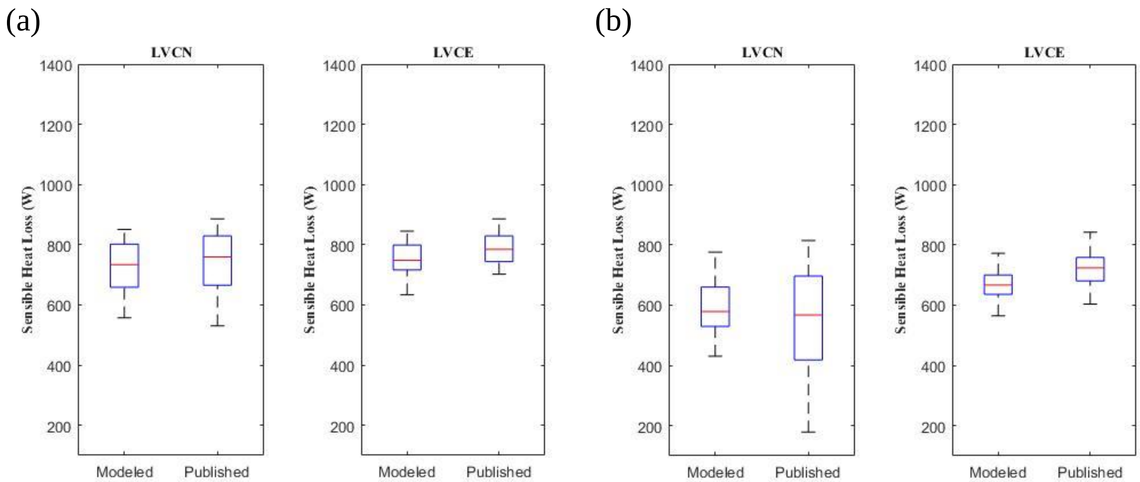


Figure 4.34. Sensible heat loss comparison for a week in July for (a) Wisconsin and (b) California

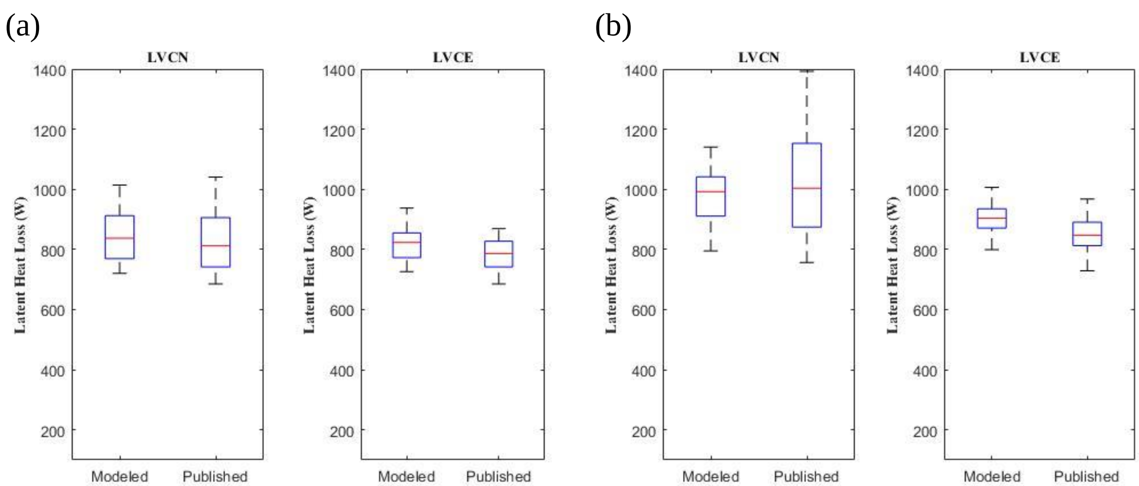


Figure 4.35. Latent heat loss comparison for a week in July for (a) Wisconsin and (b) California

The solid middle line within the box is the median of the data with ± one standard deviation creating the box. The whiskers extend out to the minimum and maximum data point.

A direct comparison of the sensible and latent heat predicted by the model against published data was also performed. Figures 4.36 and 4.37 show the data comparison with a best fit line for the predicted model and published data for both states at the LVCN and LVCE cases, respectively.

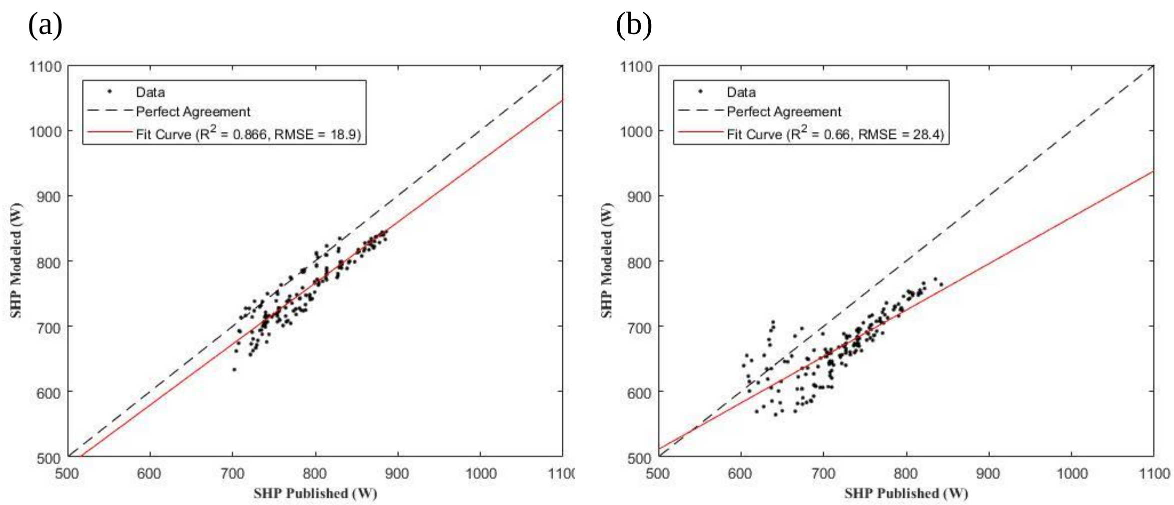


Figure 4.36. Sensible heat loss comparison for (a) Wisconsin and (b) California with evaporative cooling for a week in July at the LVCN strategy

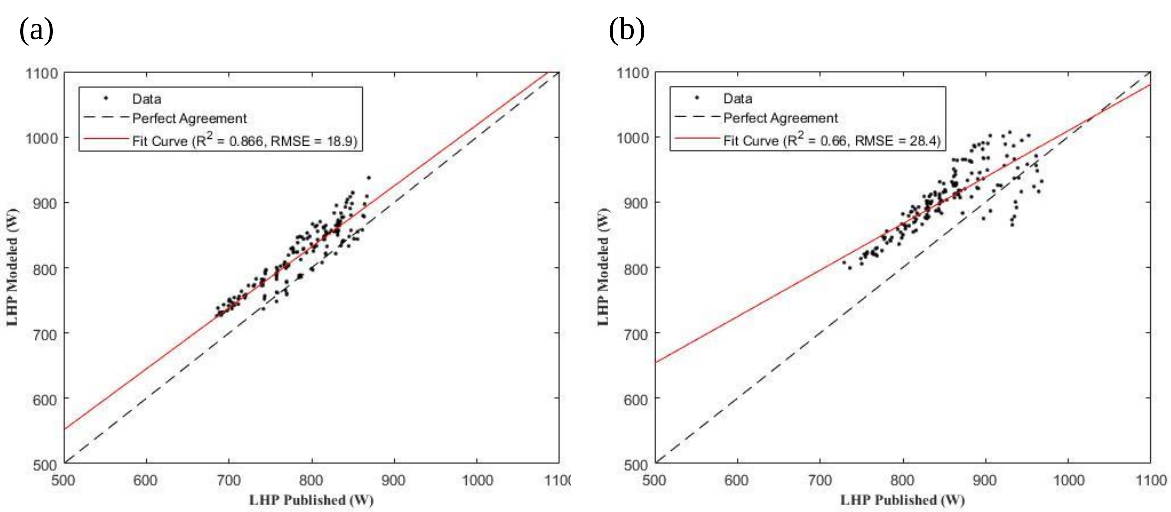


Figure 4.37. Latent heat loss comparison for (a) Wisconsin and (b) California with evaporative cooling for a week in July at the LVCE strategy

The coefficient of determination (R^2) and root mean square error (RMSE) was consistent for sensible and latent heat loss for each state but varied between states. Wisconsin has a high R^2 value of 0.866, whereas California has an R^2 of 0.66. There were 168 total data points plotted which resulted in a higher RMSE for both Wisconsin (18.9 W) and California (28.4 W).

The cow's respiration rates were also compared amongst the varied ventilation cases (LVC and HVC) without and with evaporative cooling. A cow's respiration rate can help a producer determine if she is heat stressed. As previously mentioned, the maximum respiration rate used in this model was 120 BPM (Renaudeau et al., 2012) with the minimum respiration rate fixed at 20 BPM.

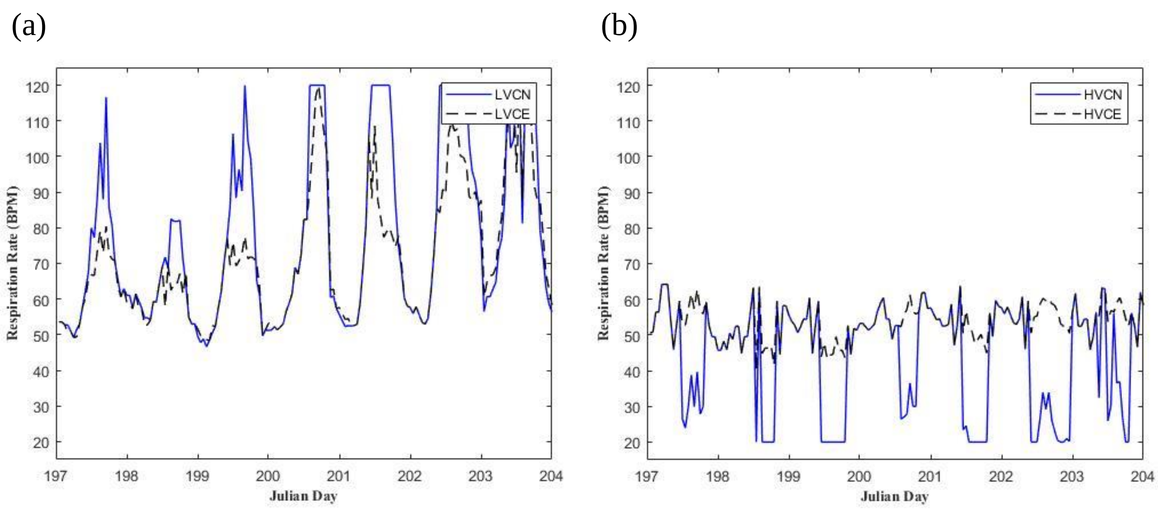


Figure 4.38. Respiration rates for Wisconsin for (a) LVC and (b) HVC for a week in July

(a) (b)

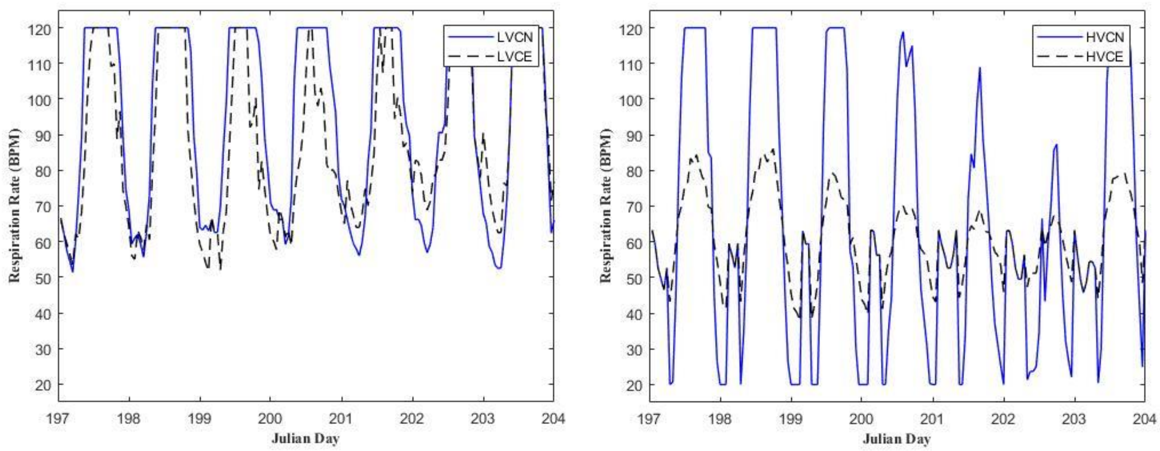


Figure 4.39. Respiration rates for California for (a) LVC and (b) HVC for a week in July

The airspeed over the cow impacts her respiration rate. Regardless of the cooling strategy, increasing the airspeed over the cow decreases her required respiration rate. For Wisconsin, the cow's predicted respiration rate for LVCN was 78±25 BPM and was 44±15 BPM for HVCN. In California, her respiration was predicted to be 95±26 BPM for LVCN and was 65±36 BPM for HVCN. In both states, turning on the evaporative cooler has minimal impact on the cow's respiration rate. In Wisconsin, the cow's respiration rate for LVCE dropped slightly to 72±19 BPM and increased to 54±5 BPM for HVCE, and for California dropped to 89±23 BPM and 61±12 BPM respectively.

Overall, the predicted sensible and latent heat loss from the developed barn and cow models were within one standard deviation of published data (Albright, 1990) throughout the summer and for the one week in July analyzed. Radiation had the highest impact on sensible heat loss to her environment and respiration and the potential for sweat to evaporate also highly impacted the cow's ability to latently dissipate heat.

4.3 Mitigation Methods Evaluated

There are several means of mitigating heat stress in the dairy industry as described in Chapter 2 as well as indices used to determine if the cow is heat stressed. Several equations were used to determine if the cow is heat stressed and thus if a mitigation method such as evaporative cooling or sprinklers is required or beneficial. The following sections outline the model's ability to predict various indicators of heat stress and any changes that result with the various mitigation strategies analyzed in this project.

4.3.1 Using THI

The current method of assessing heat stress for dairy cattle is THI (Equation 2.7). THI was plotted against the capacity to dissipate heat formulation (CDH; Equation 2.16) to predict the adequacy of THI as a heat stress indicator. Figure 4.40 plot CDH against THI for Wisconsin and California comparing the LVC and HVC ventilation cases with and without evaporative cooling. A CDH that falls below 1.00 is an indication that energy must be stored to balance THP and is one possible measure of heat stress. Therefore, the THI level at which CDH falls below 1.00 can be used to assess, in theory, the adequacy of a mitigation strategy.

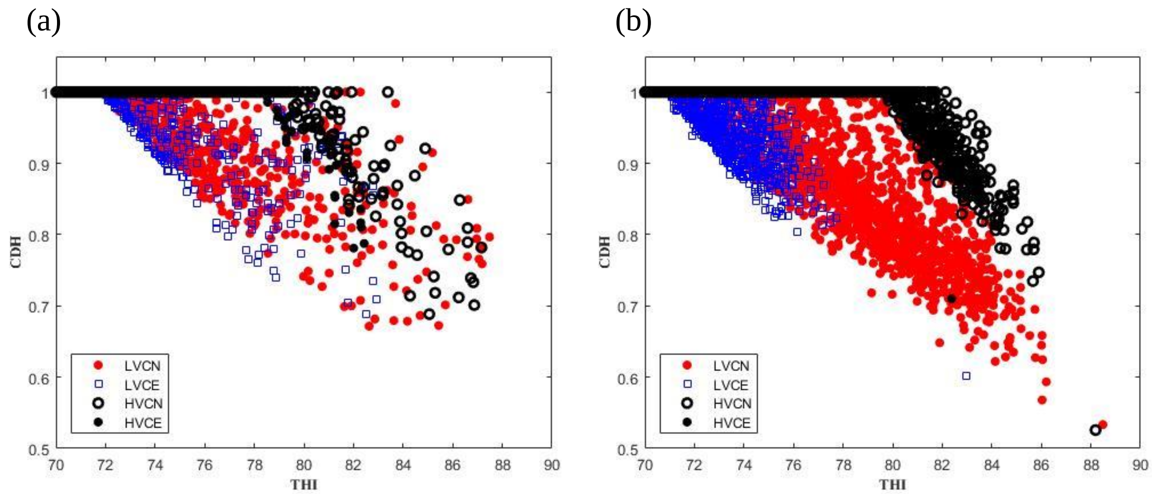


Figure 4.40. CDH vs THI in (a) Wisconsin and (b) California

In Wisconsin, CDH drops below 1.00 when THI=72 for both strategies in the LVC condition where as in California the same happens when THI=71. It is interesting to note that this CDH level occurs at the THI stress level that had been used in the dairy industry up until 2006 when a re-evaluation of THI was done that determined the onset of heat stress began at a THI of 68 (Collier, Zimbelman, et al., 2006). If the airspeed levels are increased (HVCN), THI=80 and 81 before CDH drops below 1.00 for Wisconsin and California, respectively. With evaporative cooling (HVCE), CDH drops below 1.00 when THI=79 for Wisconsin and 82.5 for California. Using CDH as an indicator of heat stress is a direct reflection of the developed cow model, and if accepted as representative of the cow's thermal exchange, is also an indication of heat stress. If an alternative heat stress indicator is used, such as THI (Equation 2.7), then CDH should fall below 1.00 at the same "indicator level" regardless of mitigation method used. Clearly, THI as used in this project, exhibited a wide range of "THI levels" when CDH fell below 1.00. These results point out the power of airspeed control in a dairy barn, and the lack of accountability of airspeed in the currently accepted THI.

4.3.2 Using CCI

A new index that incorporates airspeed is CCI (Equation 2.15). Theoretically, all CDH values should overlap at any CCI (like that described above for THI), but since the CCI equation used in this model does not incorporate radiation, previously mentioned, this perfect overlap does not occur. Figure 4.41 shows the CDH plotted against CCI for Wisconsin and California.

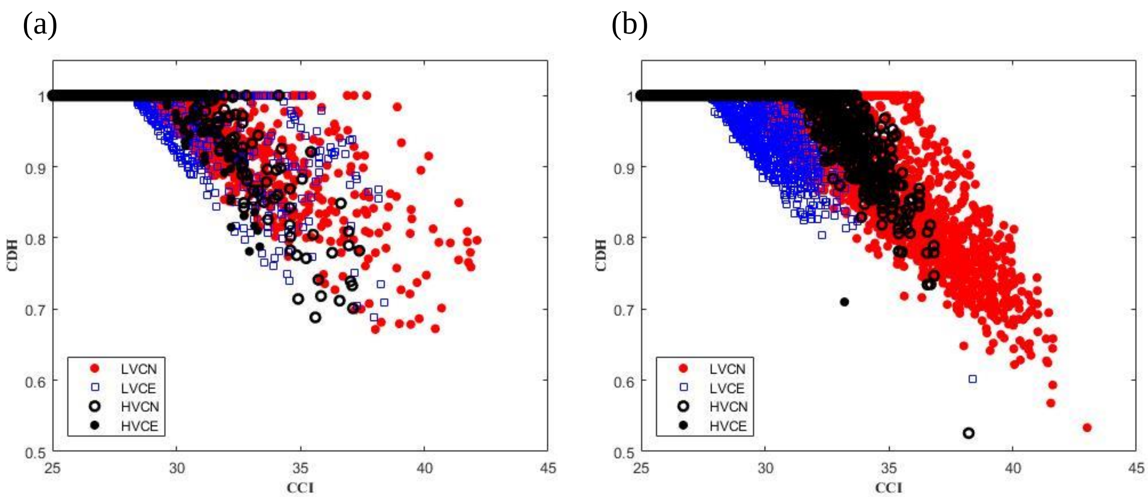


Figure 4.41. CDH vs CCI in (a) Wisconsin and (b) California

In Wisconsin, CDH drops below 1.00 when CCI=29 for the LVCN and 28 for the LVCE conditions where as in California the same happens when CCI=29.5 and 28 respectively. If the airspeed levels are increased (HVCN), CCI=31 before CDH drops below 1.00 for Wisconsin and 32 for California. With evaporative cooling (HVCE), CDH drops below 1.00 when CCI=30 for Wisconsin, and 33.5 for California. The “critical” CCI values for each ventilation case are more closely matched than those predicted with THI, which indicates that CCI is a better, but not perfect, index to assess heat stress. The CCI’s maximum range amongst cooling strategies when CDH dropped below 1.00 for Wisconsin and California was 2, whereas the THI's maximum range was 8 for Wisconsin and 10 for California, indicating that CCI appears to be a better heat stress indicator than THI.

4.3.3 Using CDH

A cow's CDH can directly indicate heat stress depending on the magnitude of the CDH value when it falls below 1.00, indicating that she is unable to dissipate internally produced heat to her surroundings. To further explore CDH, the CDH distribution from June through August (Figures 4.42 and 4.43) was plotted to show the magnitude of heat stress in each state for each cooling strategy studied with this project.

(a) (b)

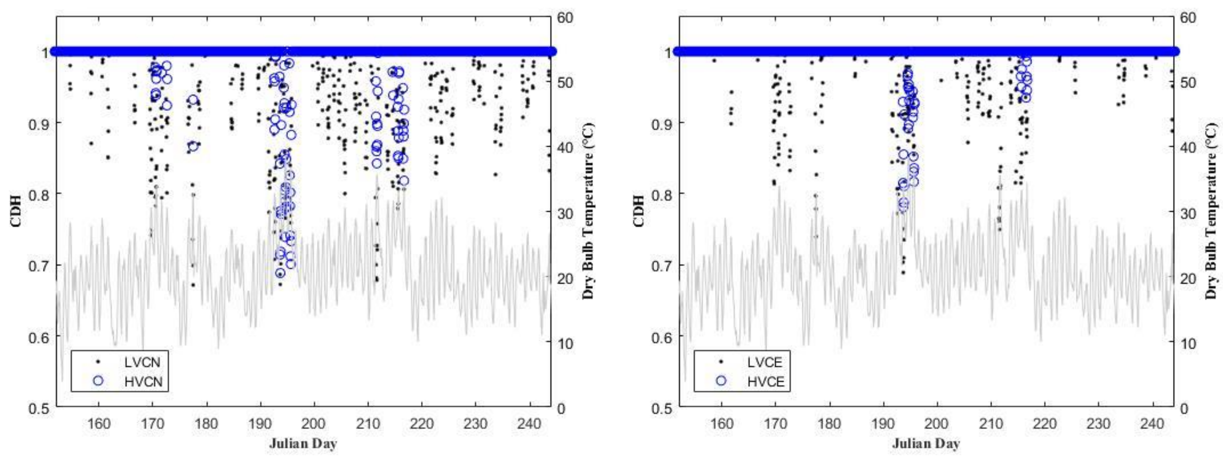


Figure 4.42. CDH distribution from June through August in Wisconsin (a) without evaporative cooling and (b) with evaporative cooling

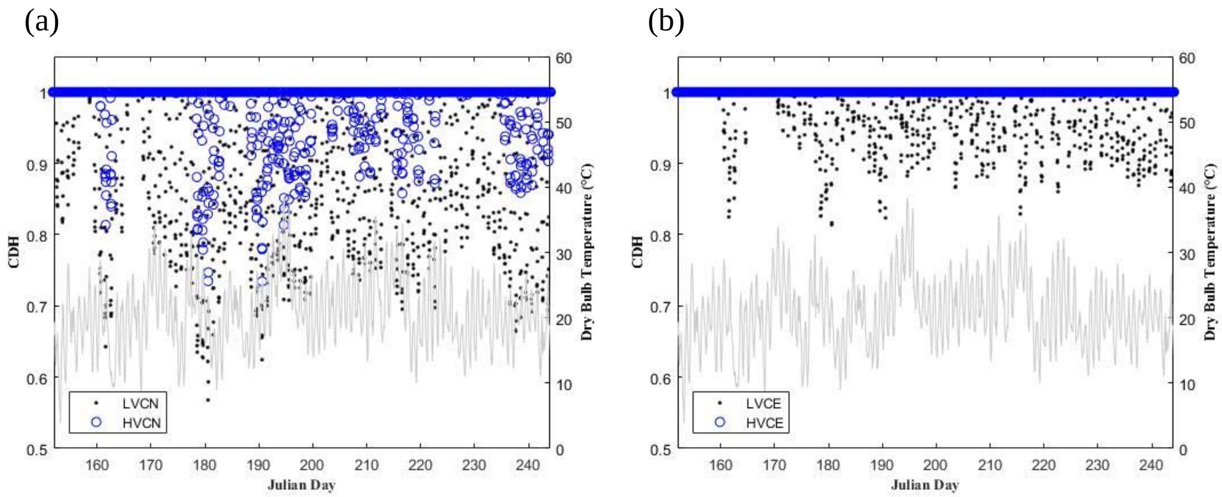


Figure 4.43. CDH distribution from June through August in California (a) without evaporative cooling and (b) with evaporative cooling

The CDH values inversely follows the outdoor dry bulb temperature trend on the secondary y-axis for all cooling strategies in Wisconsin and California. When the temperature increases, CDH decreases, which directly shows the cow's inability to dissipate heat when the dry bulb temperature is high. The lowest CDH value that occurs in Wisconsin and California is 0.67 and 0.56, respectively, which occurs at the LVCN strategy for both states. This result indicates that the cow is less able to dissipate heat at the LVCN case. Implementing evaporative cooling for both states decreases the frequency of occurrences where the CDH < 1.00, and in California, the CDH stays at 1.00 for the HVCE strategy. Figure 4.44 shows the number of hours throughout the year for each cooling condition where CDH < 1.00 for both states.

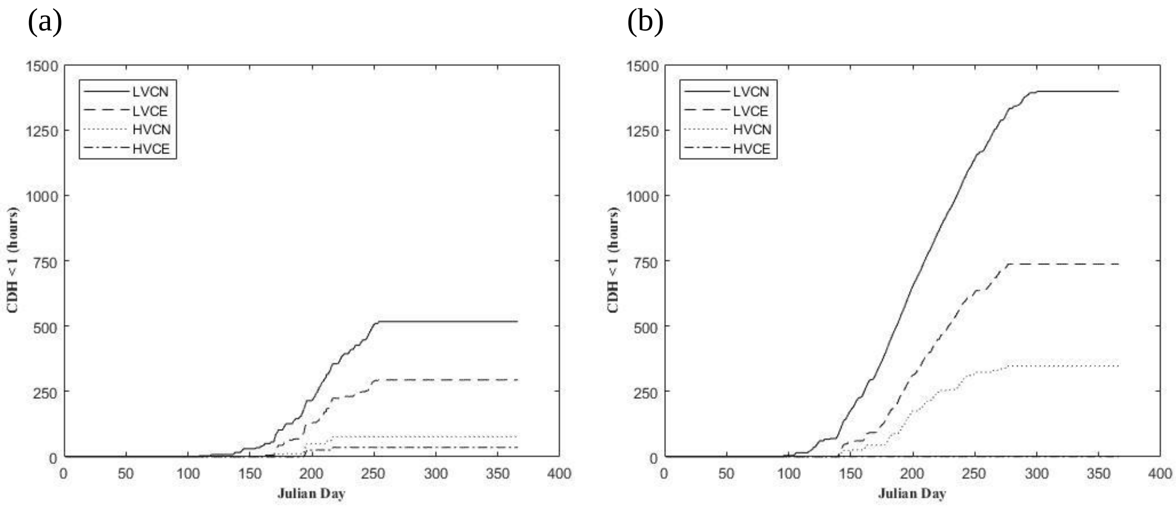


Figure 4.44. Yearly hours where CDH < 1 for (a) Wisconsin and (b) California

The LVCN strategy has the highest annual number of hours where the CDH < 1.00 for both Wisconsin (517 hours) and California (1397 hours). When a cooling strategy is used such as the HVCE strategy, it reduces the annual number of hours where CDH < 1.00 to 36 for Wisconsin and 1 for California. Although the HVCE strategy significantly reduces the annual number of hours where CDH < 1.00, it is important to consider the amount of resources used for this strategy, such as water and fan energy. The next section will look at the amount of water and fan energy needed for each cooling system.

4.3.4 Water and fan energy usage

Although certain cooling strategies may seem to reduce heat stress more optimally, it is important to keep in mind the amount of water and fan energy each strategy uses. Certain regions in the U.S. are more deprived of water than others, which may cost more to extract. Figure 4.45 shows the comparison of water usage between each cooling strategy for Wisconsin and California.

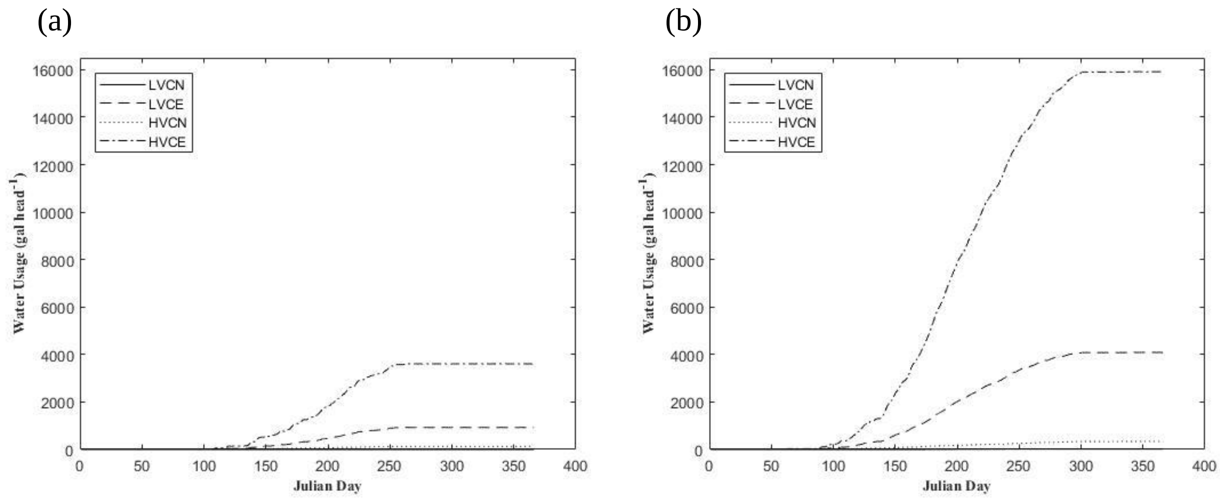


Figure 4.45. Yearly cooling water usage per cow in (a) Wisconsin and (b) California

Evaporative cooling requires significantly more water than sprinklers for both states. The HVCE strategy in Wisconsin annually consumes a model estimated 3,607 gallons cow^{-1}, compared with California at 15,919 gallons cow^{-1}. Using the HVCN strategy, where sprinkling is implemented, the potential for sprinkling requires 127 gallons cow^{-1} in Wisconsin and 336 gallons cow^{-1} for California. The sprinkling water use levels presented only consider the absolute minimum required to match the ability of the surrounding environment to evaporate water. Figure 4.46 shows the fan energy usage comparison between both states for the LVC and HVC strategies.

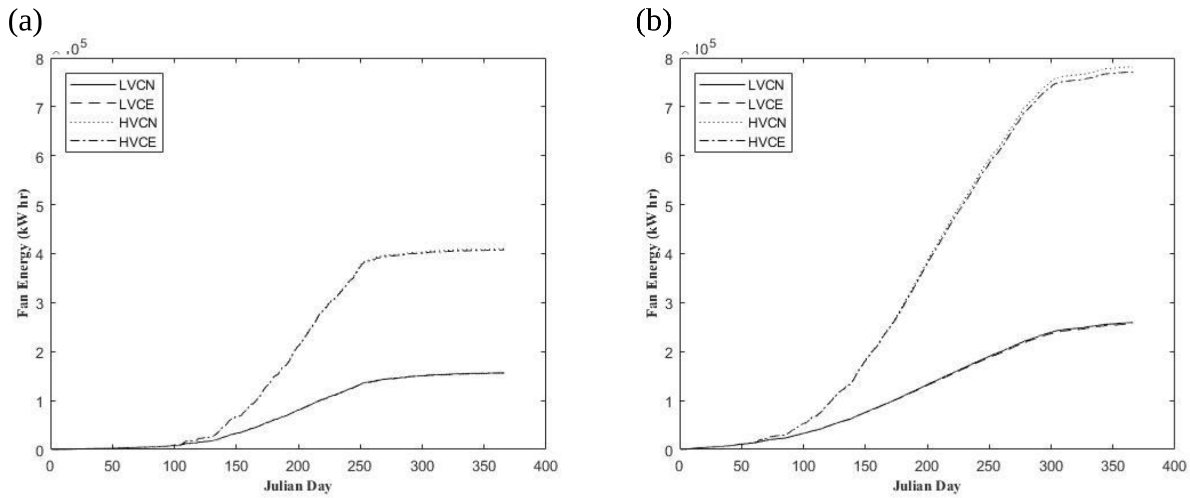


Figure 4.46. Yearly fan energy usage in (a) Wisconsin and (b) California

For both states, implementing an evaporative cooler reduces the fan energy consumption. In Wisconsin, the HVCN strategy requires is 409,950 kW-hr of fan energy consumption to achieve an airspeed of about 2 m s^{-1} through the barn versus 407,620 kW-hr at the HVCE strategy. California's fan energy consumption at the HVCN strategy requires 782,070 kW-hr and 771,590 kW-hr at the HVCE strategy. The fan energy consumption is less when the evaporative cooler is turned on due to the decrease in ambient temperature at the cow level. Since the ventilation systems is set to keep the barn's temperature at 20°C, the fans are not running as long as they are when there is no evaporative cooler use.

Since water is scarce and energy prices vary in certain regions throughout the country, the increased cooling potential from a selected cooling system needs to be taken into consideration before implementing increased fan energy, an evaporative cooler, or sprinklers into a facility.

CHAPTER 5. GEOSPATIAL EVALUATION OF MITIGATION METHOD PERFORMANCE

Chapter 5 provides results for all 215 TMY3 Class 1 stations, using the barn and cow models described in Chapter 3, and the procedures and results described in Chapter 4 where Wisconsin (SN:726435) and California (SN:723890) were used as test cases. ArcMap 10.4 (*ArcGIS 10.4*, 2016) was used to interpolate the data between each station to produce a contour map of the U.S. based on selected parameters.

5.1 Environment Evaluated

With all of the weather data from each TMY3 station, multiple graphs can be created to evaluate the environment throughout the U.S. Since climates are greatly affected by elevation, an elevation map of the U.S. was also plotted (Figure 5.1).

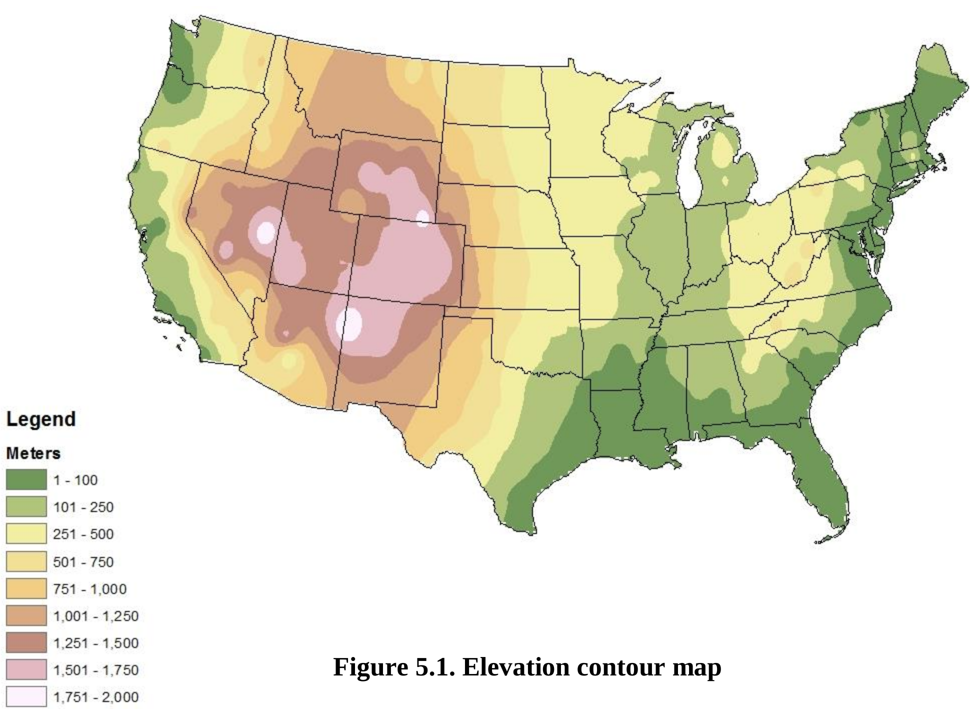


Figure 5.1. Elevation contour map

A map plotting the temperature for each station can also be created to evaluate the environment. The following U.S. contour figures show the annual amount of time (hours) where the dry bulb temperature at the center of the barn is greater than 25°C. Figure 5.2 shows the annual number of hours the barn’s temperature would be greater than 25°C without evaporative cooling using the LVC strategy. In other words, Figure 5.2 shows the worst case scenario if no cooling strategy is used. Figure 5.3 shows the annual number of hours the barn’s center temperature is greater than 25°C if the best cooling strategy is implemented. The best cooling strategy was found by minimizing the amount of annual hours where the barn’s temperature is greater than 25°C. Figures 5.4 and 5.5 put these hours into a yearly percentage.

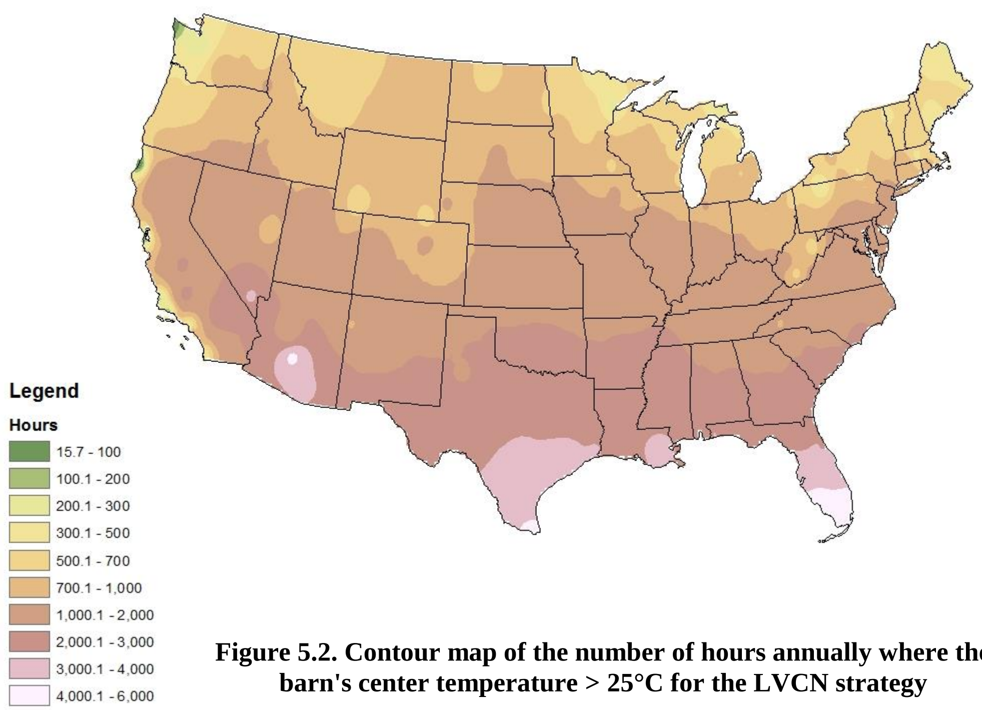


Figure 5.2. Contour map of the number of hours annually where the barn's center temperature > 25°C for the LVCN strategy

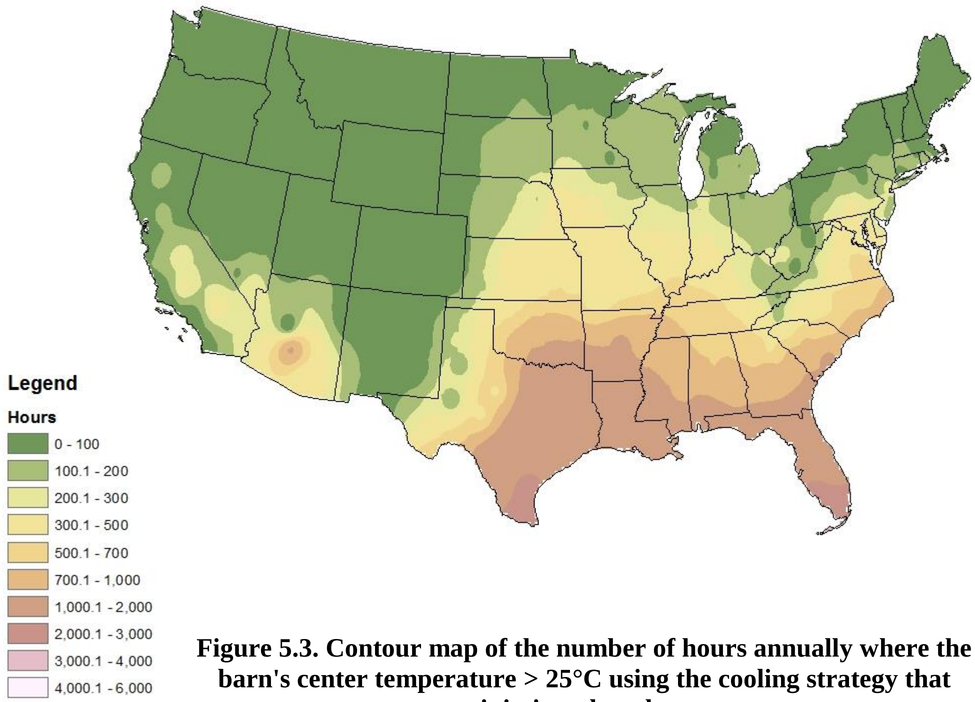


Figure 5.3. Contour map of the number of hours annually where the barn's center temperature > 25°C using the cooling strategy that minimizes these hours

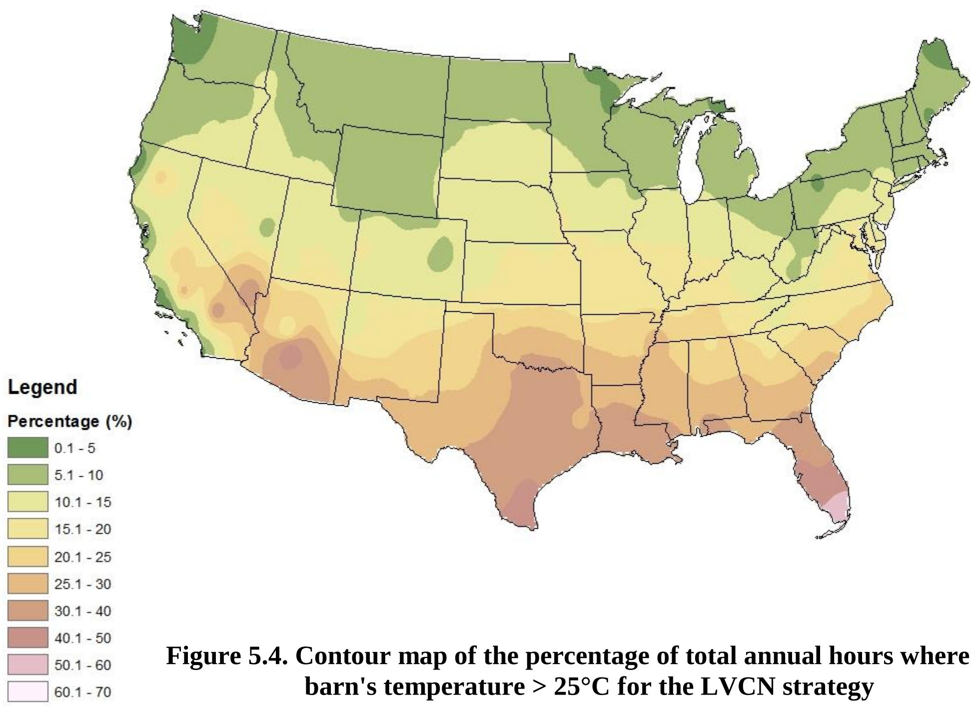


Figure 5.4. Contour map of the percentage of total annual hours where the barn's temperature > 25°C for the LVCN strategy

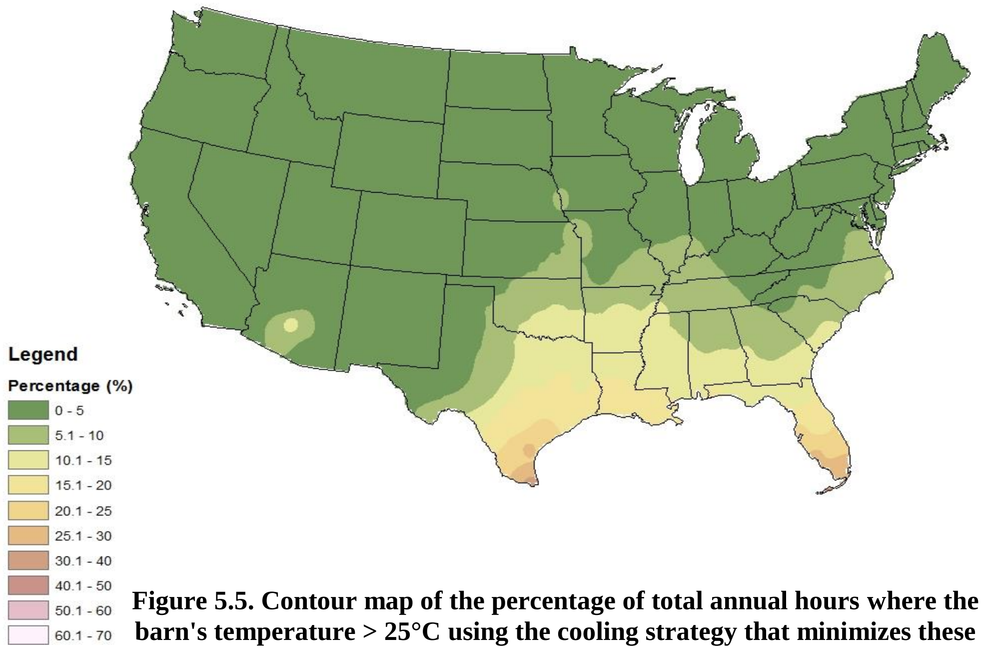


Figure 5.5. Contour map of the percentage of total annual hours where the barn's temperature > 25°C using the cooling strategy that minimizes these percentages

Implementing the best cooling strategy drastically reduces the number of hours where the barn’s center temperature is greater than 25°C. When the LVCN strategy is used, the majority of the country has over 700 hours throughout the year where the barn’s center temperature is greater than 25°C, but when the best cooling strategy is used for each TMY3 Class 1 station, the majority of the country drops far below 700 hours. In other words, when using the LVCN strategy, greater than 10% of the time throughout the year the barn’s temperature is greater than 25°C, and when the best cooling strategy is used, that time decreases to below 10%, with less than 5% of the annual hours where the barn’s center temperature is greater than 25°C, which is equivalent to 437.5 hours or 18 days throughout the year for the majority of the U.S.

The next several maps show the THI distribution throughout the U.S. Figure 5.6 outlines the maximum one-hour THI value that occurs at each station throughout the year and the level of heat stress associated with that value. Refer back to Table 2.5 for each threshold value.

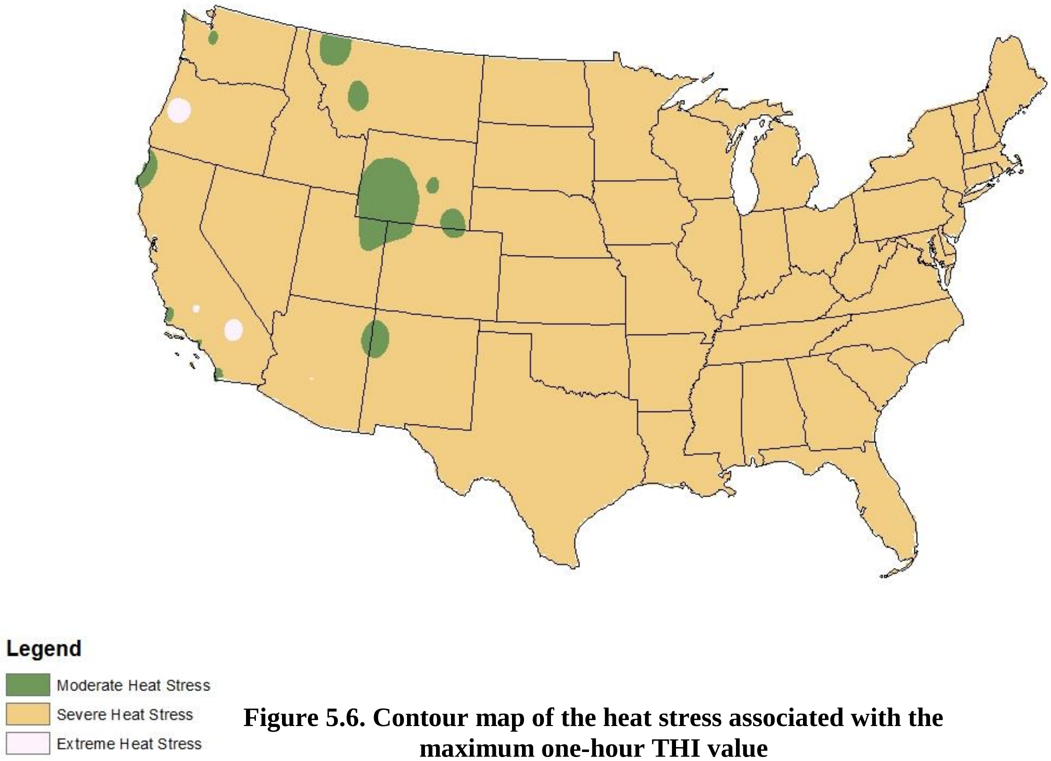
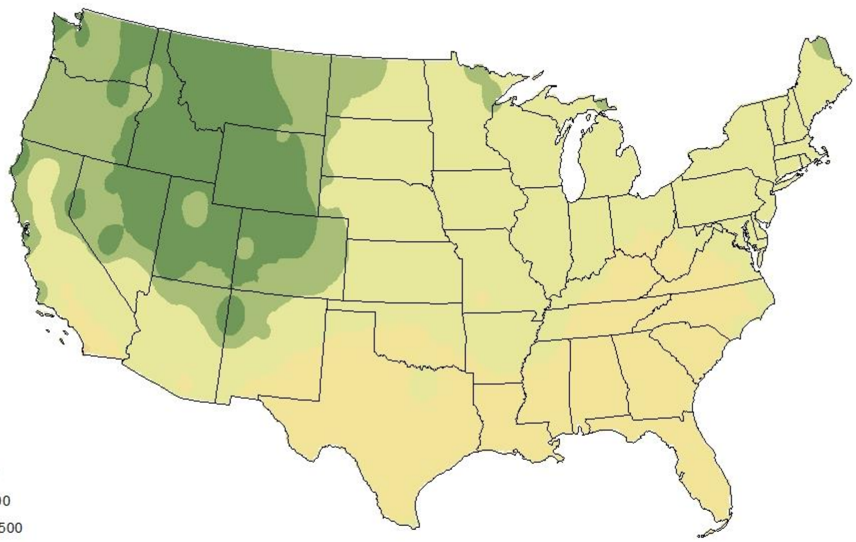


Figure 5.6. Contour map of the heat stress associated with the maximum one-hour THI value

The majority of the country's annual one-hour peak THI value falls in the severe heat stress category (THI = 79 to 89). This map only considers this maximum THI value for one hour, so if the THI decreases over the next few hours, the cow would most likely be able to dissipate her heat without causing any stress. This map does however show the magnitude of possible heat stress cases that occur throughout the year.

Figures 5.7 and 5.8 show the number of hours where the THI is between 68 and 71, and 72 and 77 respectively. As the THI increases from 68-71 and 72-77, the number of hours almost doubles in the Southeastern U.S. Figures 5.9 and 5.10 put these hours into a yearly percentage.



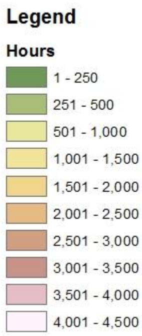
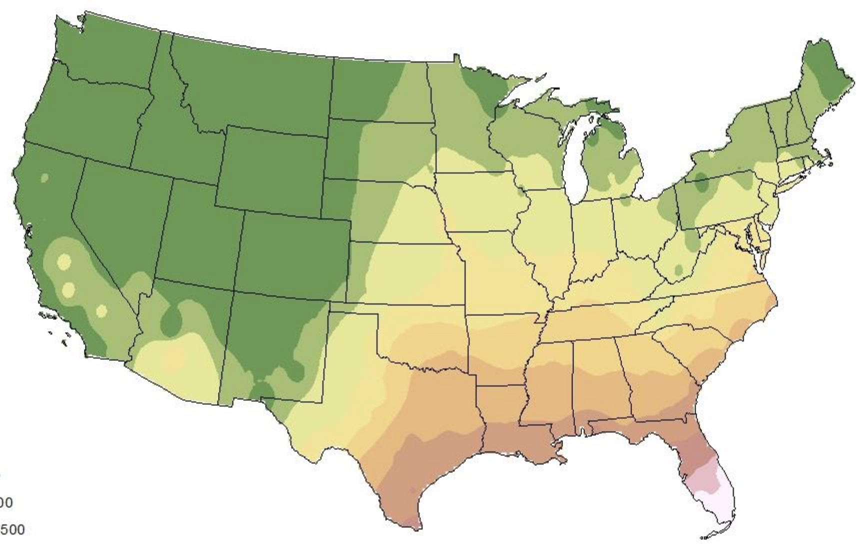


Figure 5.7. Contour map of the number of hours annually where the THI is between 68 and 71



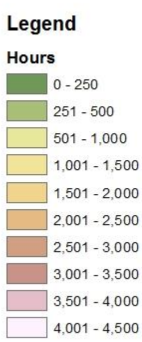


Figure 5.8. Contour map of the number of hours annually where the THI is between 72 and 77

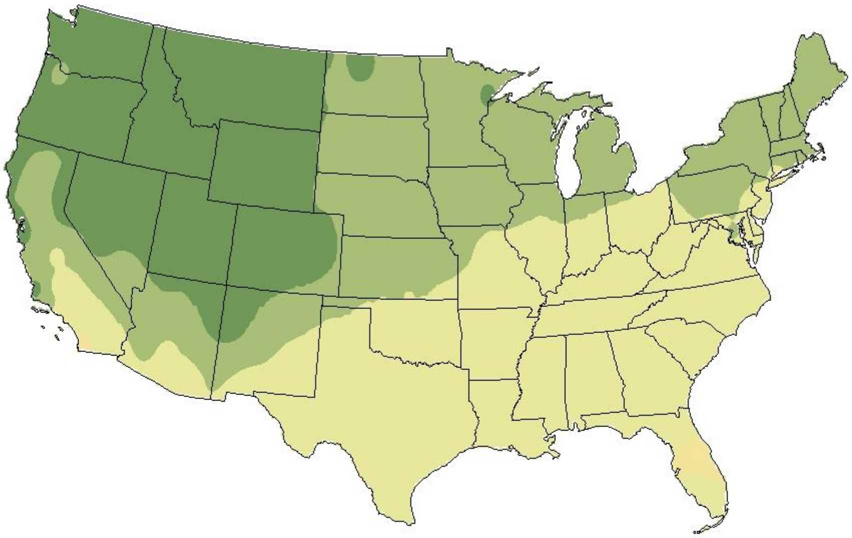


Figure 5.9. Contour map of the percentage of total annual hours where the THI is between 68 and 71

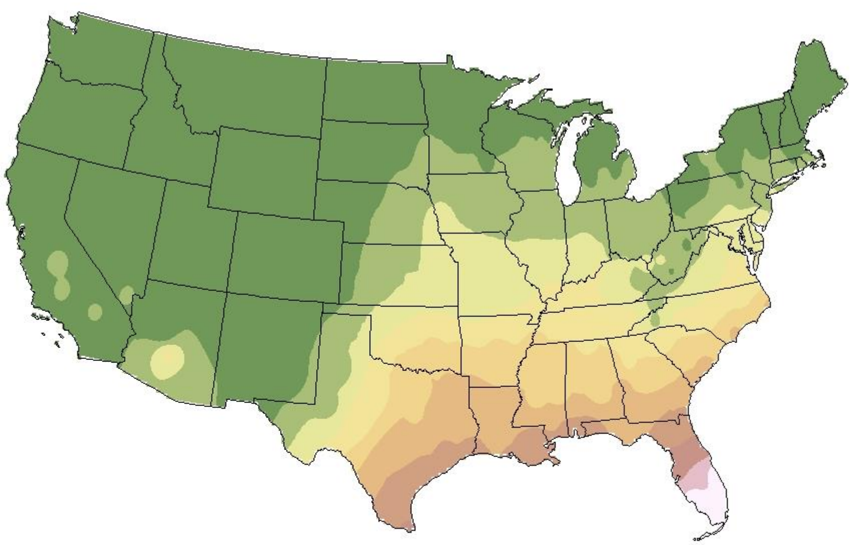


Figure 5.10. Contour map of the percentage of total annual hours where the THI is between 72 and 77

The maximum number of annual hours where the THI is between 68 and 71 is 1620 hours (SN: 722900, San Diego, CA), but when the THI increases to a value between 72 and 77, the maximum number of hours is 4600 hours (SN: 722020, Miami, FL). The majority of the country experiences less than 15% of the total annual hours with a THI value between 68 and 71 whereas almost half of the country has less than 5% of total annual hours when the THI falls between 72 and 77.

5.2 Mitigation Methods Evaluated

The following section shows multiple maps with recommended mitigation strategies that reduces heat stress most optimally for that particular parameter. The cooling strategies will change for each thermal index evaluated, and a final series of mitigation maps will be presented in Chapter 6 that producers can use to implement the best cooling strategy in their facility for their location with given availability to resources.

5.2.1 Using THI

The maps in this section will suggest cooling strategies to reduce the THI in the barn. In order to do this, code was written that looped through each cooling strategy for each station and found the minimum number of hours where the THI was within a selected range. Then, for that minimized time, the associated cooling strategy was plotted. Table 5.1 shows an example from California and Wisconsin. In California, the minimal number of hours where the THI is greater than 68 is 1806 hours, which occurs at the VHVCE strategy, and for Wisconsin, this also occurs at the VHVCE strategy with 1171 hours. Using this method, Figure 5.11 shows the best cooling strategy, regardless of resource use, based on THI levels greater than 68.

Table 5.1. Example of two stations to determine best cooling strategy for selected parameters (THI; SN:723890=CA and SN:726435=WI)

SN	Latitude	Longitude	Evap Case	Velocity	THI hours (>68)
723890	36.783	-119.717	N	LVC	2923
723890	36.783	-119.717	E	LVC	2214
723890	36.783	-119.717	N	MVC	2798
723890	36.783	-119.717	E	MVC	2010
723890	36.783	-119.717	N	HVC	2718
723890	36.783	-119.717	E	HVC	1870
723890	36.783	-119.717	N	VHVC	2688
723890	36.783	-119.717	E	VHVC	1806[&]
726435	44.867	-91.483	N	LVC	1493
726435	44.867	-91.483	E	LVC	1337
726435	44.867	-91.483	N	MVC	1403
726435	44.867	-91.483	E	MVC	1234
726435	44.867	-91.483	N	HVC	1372
726435	44.867	-91.483	E	HVC	1184
726435	44.867	-91.483	N	VHVC	1363
726435	44.867	-91.483	E	VHVC	1171[&]

[&]Selected as best for each condition

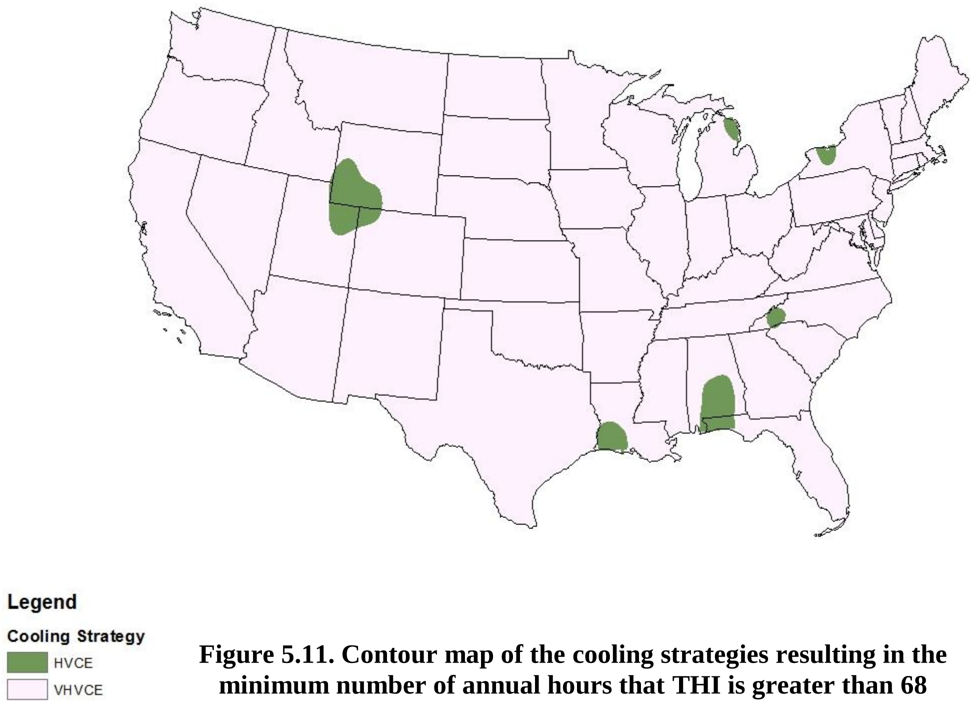


Figure 5.11. Contour map of the cooling strategies resulting in the minimum number of annual hours that THI is greater than 68

Based on the maps above, the minimum number of THI hours greater than 68 results with a cooling strategy of HVCE and VHVCE throughout the entire U.S. Resource use (fan energy and water) will also be a factor in any cooling decision, and Chapter 6 will address this issue.

A cow becomes heat stressed when she is unable to dissipate her heat after several hours. Figure 5.12 quantifies the number of events where THI is between 68 and 71 for 8, 10, 12, and 14 continuous hours. One event accounts for one 8-hour continuous period where the THI does not go below a value of 68. The THI range of 68-71 was selected because this is the mild heat stress range.

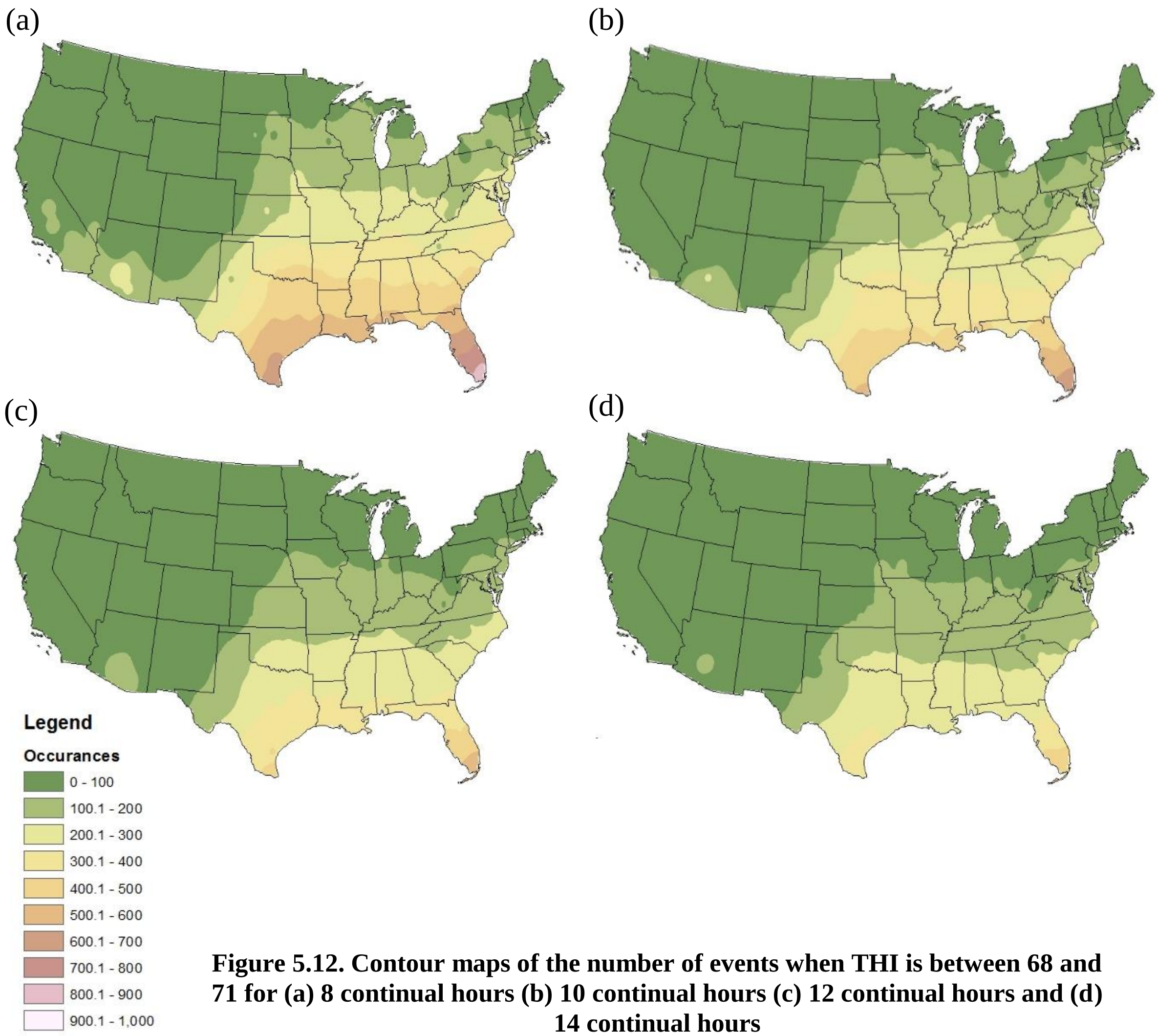


Figure 5.12. Contour maps of the number of events when THI is between 68 and 71 for (a) 8 continual hours (b) 10 continual hours (c) 12 continual hours and (d) 14 continual hours

As the length of continual hours increases, the number of events of those continual hours decreases. If a cow can dissipate her heat up until a continual 14 hour period of the environment having a THI between 68 and 71, the number of events where this occurs is less than 100 for the majority of the U.S.

5.2.2 Using CCI

The same principal as used in the previous section was applied to CCI. Figure 5.13 shows the number of one-hour events when the CCI is between 25 and 29.

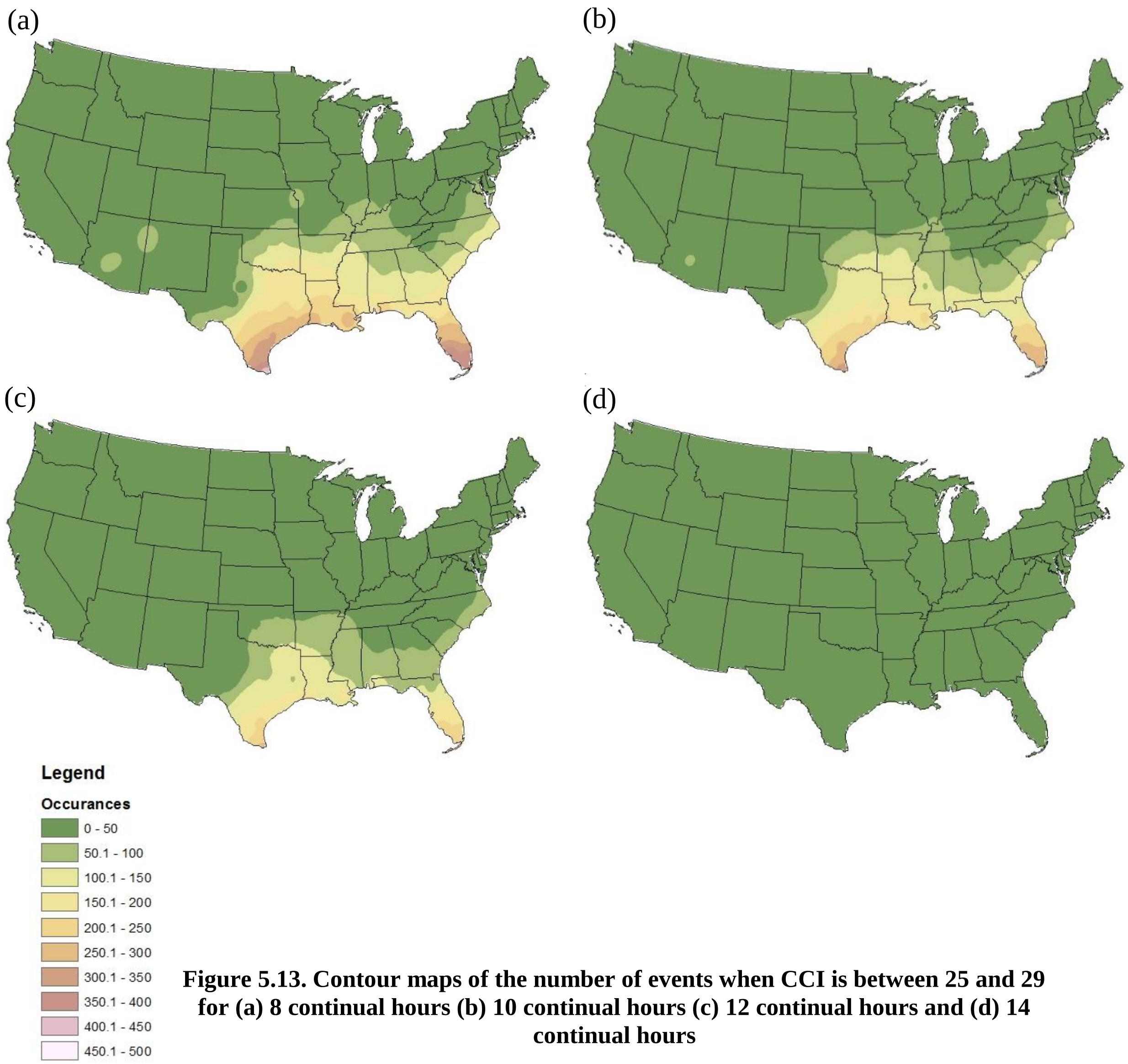


Figure 5.13. Contour maps of the number of events when CCI is between 25 and 29 for (a) 8 continual hours (b) 10 continual hours (c) 12 continual hours and (d) 14 continual hours

Again, as the length of continual hours increases, the number of events of those continual hours decreases. Above 8 continual hours, the majority of the country encounters less than 50 events where the CCI is between 25 and 29. There are no events throughout the U.S. when the CCI is between 25 and 29 for 14 continual hours using the existing TMY3 Class 1 data set.

5.2.3 Using CDH

The number of hours where the CDH drops below 1.00 was also plotted for several magnitudes below 1.00 such as 0.9, 0.8, and 0.7. Figure 5.14 shows the comparison.

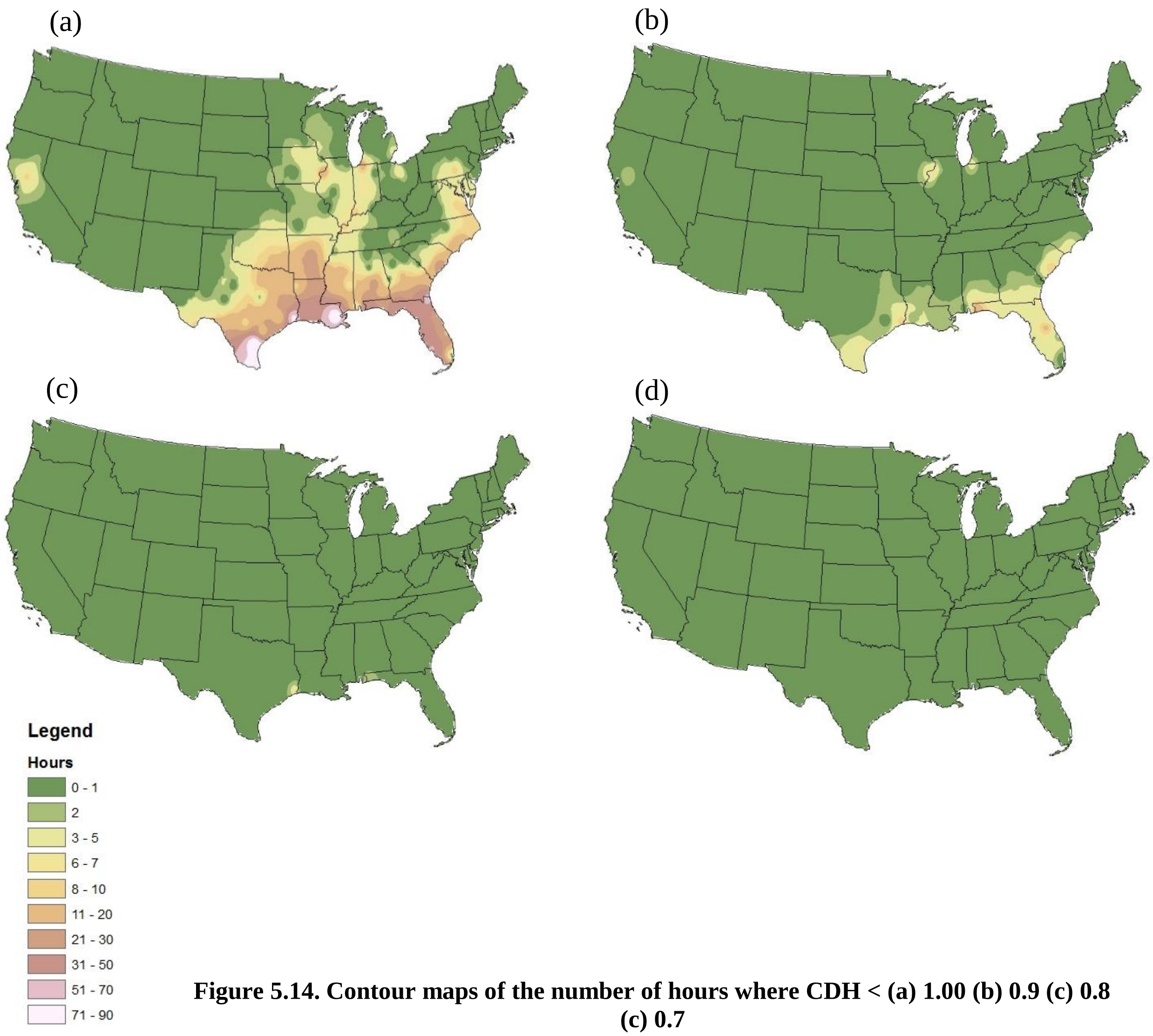
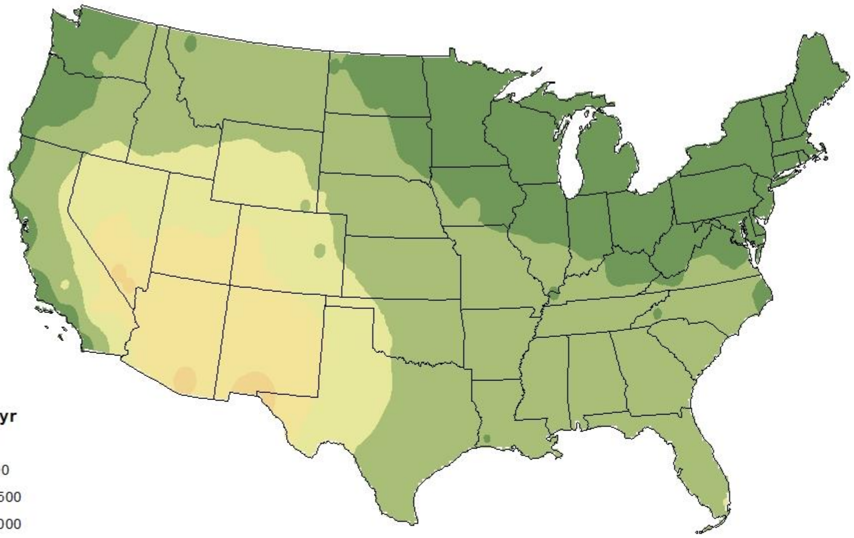


Figure 5.14. Contour maps of the number of hours where CDH < (a) 1.00 (b) 0.9 (c) 0.8 (c) 0.7

The Southeast and up through part of the Midwest encounter multiple hours throughout the year where the CDH < 1.00. Like the number of occurrences where THI or CCI is between a particular value, occurrence numbers decrease as the number of continual hours increases. Almost the entire U.S. has less than one hour throughout the year where the CDH < 0.7 using the current TMY3 Class 1 data set.

5.4.4 Water and fan energy usage

Water and fan energy use is very important to take into consideration. Some locations throughout the U.S. are more deprived of water, which may increase the price of water due to water scarcity, and energy prices vary throughout the U.S as well. The following figures show the units of water (gallons) required and units of fan energy (kW-hr) based on selected cooling strategy. Figure 5.15 and 5.16 compare water usage between each cooling method. The amount of water for the potential for sprinkling is shown in Figure 5.15, while the water usage for evaporative cooling is shown in Figure 5.16. The maximum amount of water used throughout the year, for cooling purposes only, as expected occurs at the VHVCE strategy during evaporative cooling (Figure 5.16).



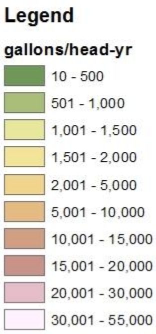
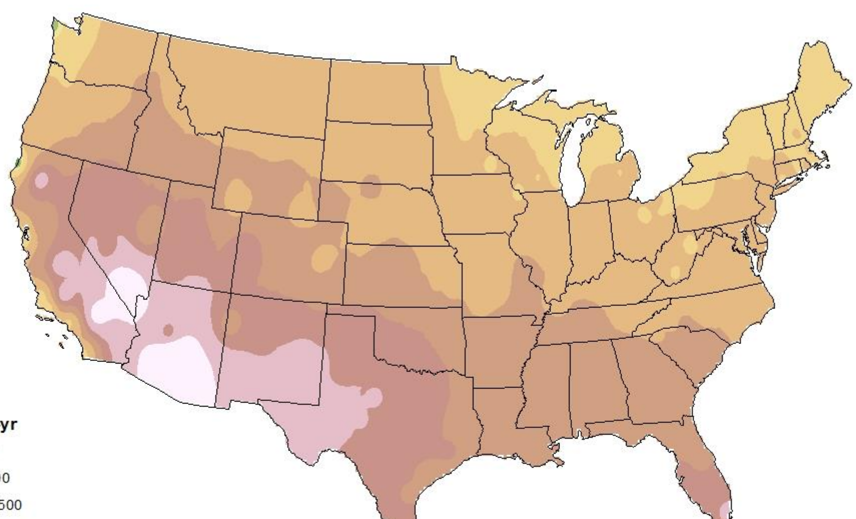


Figure 5.15. Contour map of the yearly sprinkler water usage per cow, assuming water is only applied at the rate that can be evaporated



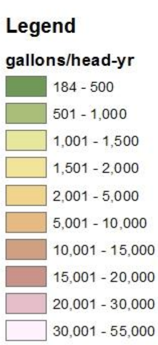


Figure 5.16. Contour map of the yearly evaporative cooler water usage per cow if the VHVCE cooling strategy is implemented

Evaporative cooling requires significantly more water than the water that would be required for the potential for sprinkling. The majority of the country would require less than 2,000 gallons per head if water was only used for the potential for sprinkling whereas that water requirement more than doubles for the majority of the U.S. when evaporative coolers are used. Figure 5.17 shows the maximum amount of fan energy consumption, which results from the VHVCN strategy. The LVCE strategy requires the least amount of energy consumption as expected.

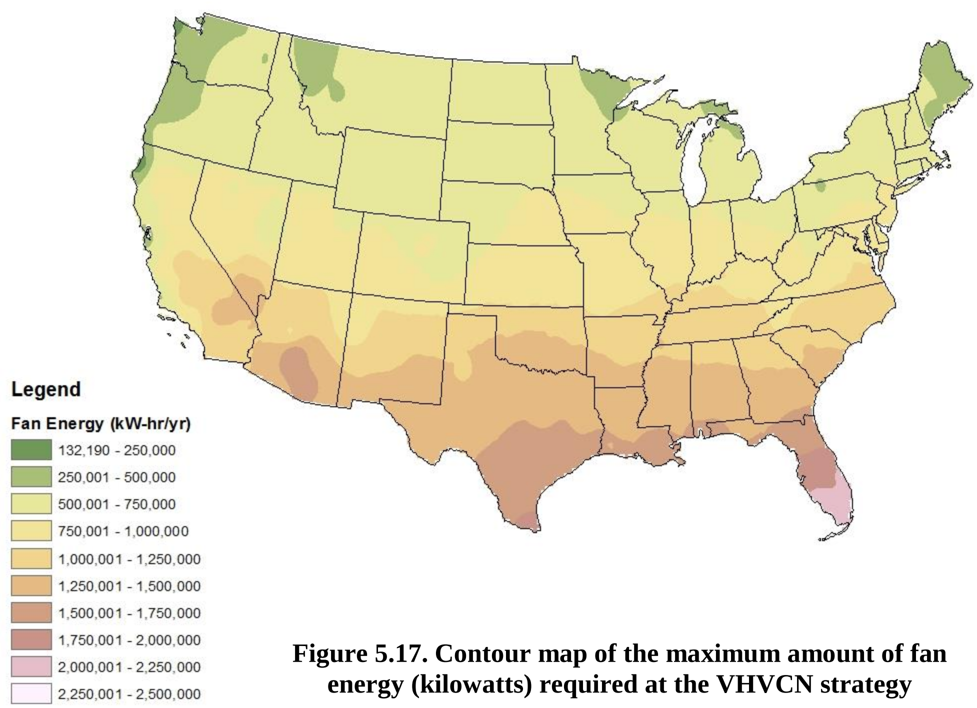


Figure 5.17. Contour map of the maximum amount of fan energy (kilowatts) required at the VHVCN strategy

The energy use only takes into consideration the fans that are operating. The non-evaporative cooler strategies have higher energy consumption at the same airspeeds because the fans are running longer compared with evaporative cooler use. Since the evaporative cooler decreases the indoor temperature, the fans don’t need to run as long since it is not as hot at the cow level.

Although decreasing water and fan energy usage may seem ideal, the benefit of the increased cooling potential needs to also be considered. For example, a cooling system may require more water and/or fan energy, but it could also have a better cooling potential, which would reduce heat stress for the cattle. Decreasing the amount of energy use may cost less, but higher airspeeds may decrease heat stress as well. Less heat stress means higher conception rates and more milk production, which directly correlates to a higher profit. Multiple considerations need to be accounted for when looking at the most optimal cooling system for particular regions, which will be looked at further in the next chapter.

CHAPTER 6. SUMMARY

The overall goal of this project was to determine which heat stress mitigation strategy has the greatest cooling potential for dairy cattle dependent on historical climatic conditions. A model was developed (Chapter 3) that analyzed the thermoregulation of a single cow in the center of a full, uniformly distributed barn using TMY3 Class 1 data. The model's predicted results were compared to published data (Chapter 4) to determine the validity of the model. Once the model was deemed valid, it was compiled for all 215 stations throughout the United States to create U.S. contour plots of multiple different parameters (Chapter 5). This chapter uses the model to achieve the overall goal of the project as well as looks at future work that could be done to improve the model.

6.1 Final Heat Stress Mitigation Maps

In order to determine the best overall cooling strategy to implement in a facility throughout multiple climates, several parameters need to be taken into consideration. First of all, the purpose of a cooling system is to cool an animal; therefore, the cooling potential of a particular system is the greatest factor to consider. Since CCI was determined to be a better thermal index to assess heat stress (Chapter 4) and dry matter intake decrease is a direct physiological response of heat stress (Chapter 2), a feed intake equation was created as a function of CCI. This was done by applying the feed intake equation as a function of temperature (Appendix G) and assuming a relative humidity of 50% and an airspeed of 0.5 m s^{-1} to calculate CCI (Table 6.1).

Table 6.1. CCI calculated from dry bulb temperature, 50% RH, and 0.5 m s^{-1}

Temperature (°C)	DMI (kg day^{-1})	CCI
-16	25.8	-14.0
-15	24.8	-13.0
-6	23.8	-3.8
-5	3.3	-2.8
4	23.3	6.5
5	23.1	7.6
14	22.9	17.2
15	22.2	18.3
24	22.2	28.1
25	21.1	29.2
34	20.0	39.2
35	20.0	40.4
36	17.2	41.5
37	14.4	42.6
40	13.3	46.0

The dry matter intake was then plotted against CCI (Figure 6.1) to create a correlation equation (Equation 6.1):

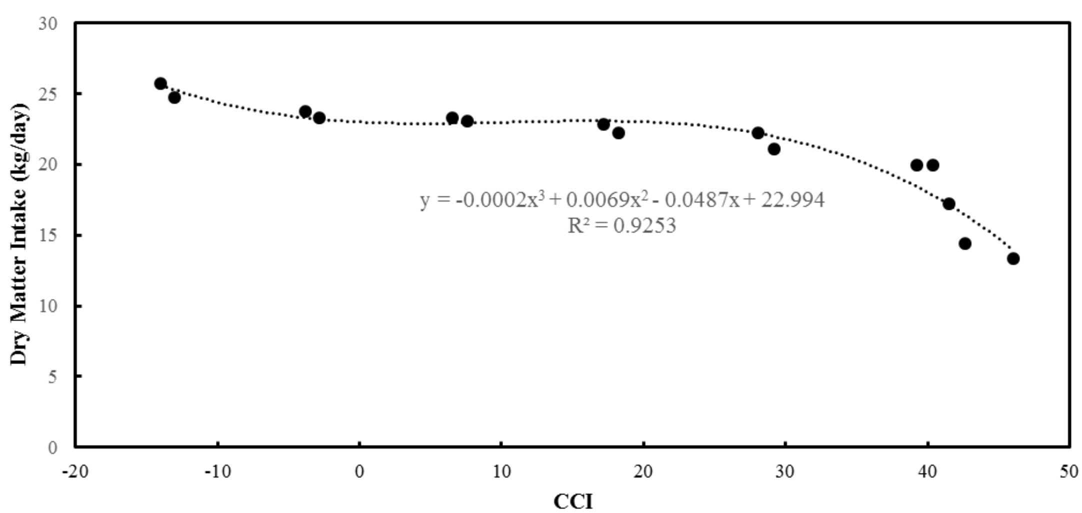


Figure 6.1. DMI vs CCI correlation graph

$$FI = -0.0002 * CCI^3 + 0.0069 * CCI^2 - 0.0487 * CCI + 22.994 \quad (6.1)$$

where,

FI = feed intake (kg cow^{-1} day^{-1}).

The calculated feed intake from CCI was then multiplied by a feed efficiency of 1.5 kg milk per kg feed (Hutjens, Michael F., 2005) to determine the milk production for that cow, which can be directly related to a return on investment for the producer.

Another consideration to determine the best cooling strategy, previously mentioned, is the resource use such as water or energy requirements to accomplish any given level of heat stress mitigation. Due to continual fluctuation in milk prices, and cost of water and energy as well as the dispersion of those prices throughout the U.S., a map looking at the most economical cooling system was not feasible to create. Instead, an equation was developed (Equation 6.2) to determine a return on investment (ROI) per cow with each consideration that could later be modified depending on the fluctuation in prices by location

$$ROI = N_{cows}[(M_{pr} * UC_w * M_{tot}) - (W_{pr} * W_{cool}) - (E_{pr} * E_f)] \quad (6.2)$$

where,

E_{pr} = Energy price (\$ kW^{-1} hr^{-1})

E_f = Fan energy use (kW hr cow^{-1} yr^{-1})

M_{pr} = Milk prices (\$ 100^{-1} lbs^{-1})

M_{tot} = Total milk weight (kg cow^{-1})

N_{cows} = Number of cows

ROI = Return on investment (\$ yr^{-1})

UC_w = Weight unit conversion (2.2 lbs kg^{-1})

W_{pr} = Water price (\$ $gallon^{-1}$)

W_{cool} = Cooling water use (gallons cow^{-1} yr^{-1}).

This equation can then be applied to the following U.S. contour maps to determine which cooling strategy producers should consider in their facility in order to most optimally reduce heat stress in their dairy cattle. This heat stress reduction will increase conception rates and milk production for the cattle and overall increase the operation's profit. The LVCN strategy was used as a baseline condition to compare all other strategies against. Figure 6.2 shows the baseline for feed intake and milk production as well as the cooling water, total water (drinking + cooling), and fan energy use that occurs during the LVCN strategy.

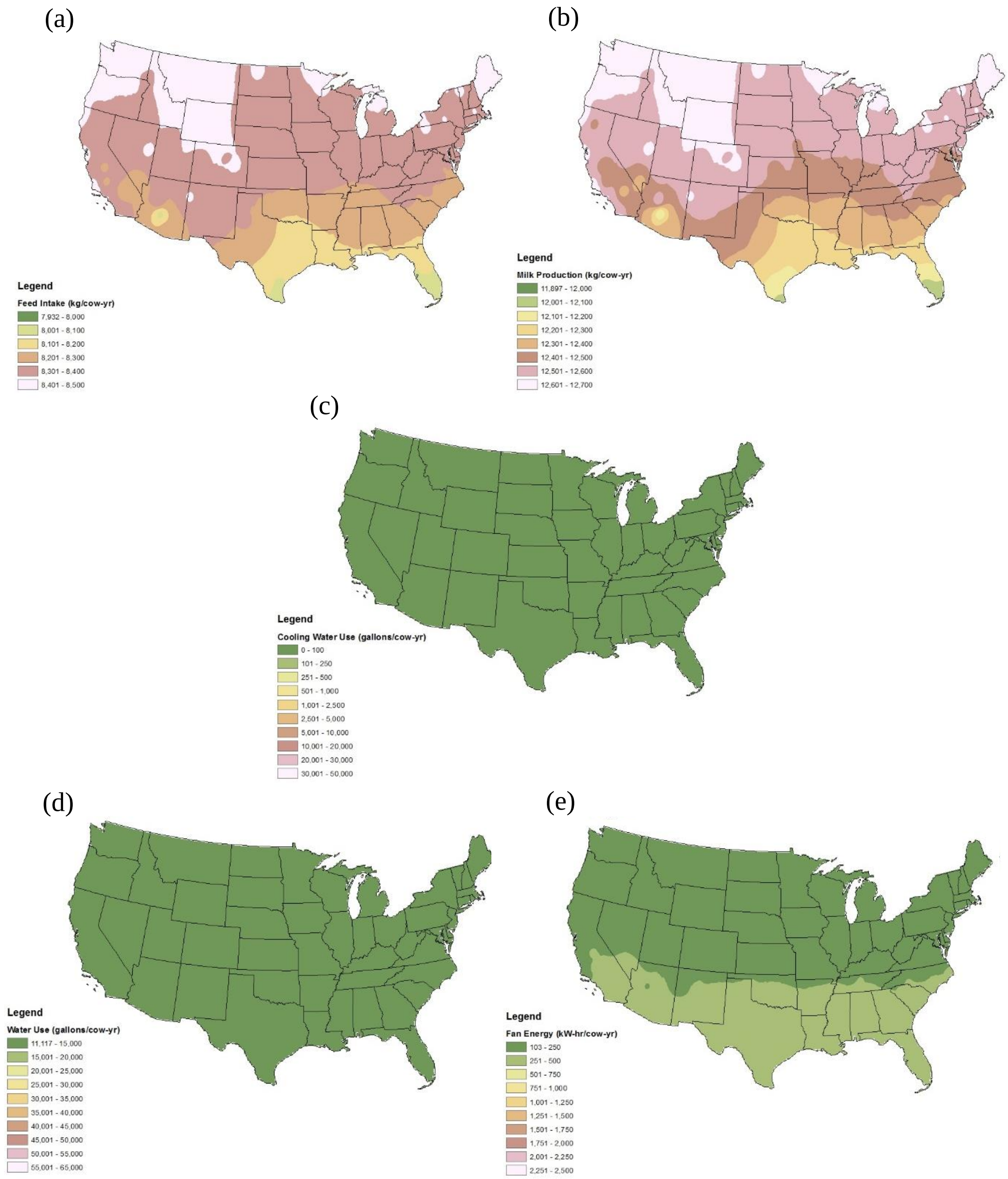


Figure 6.2. Contour maps showing the (a) feed intake (b) milk production (c) cooling water use (d) total water use, and (e) fan energy use at the LVCN strategy

The next several maps (Figures 6.3-6.9) show the increase in feed intake and milk production when a specific cooling strategy is used along with the amount of water needed for each cooling strategy, the total amount of water used (cooling + drinking), and the fan energy needed per cow.

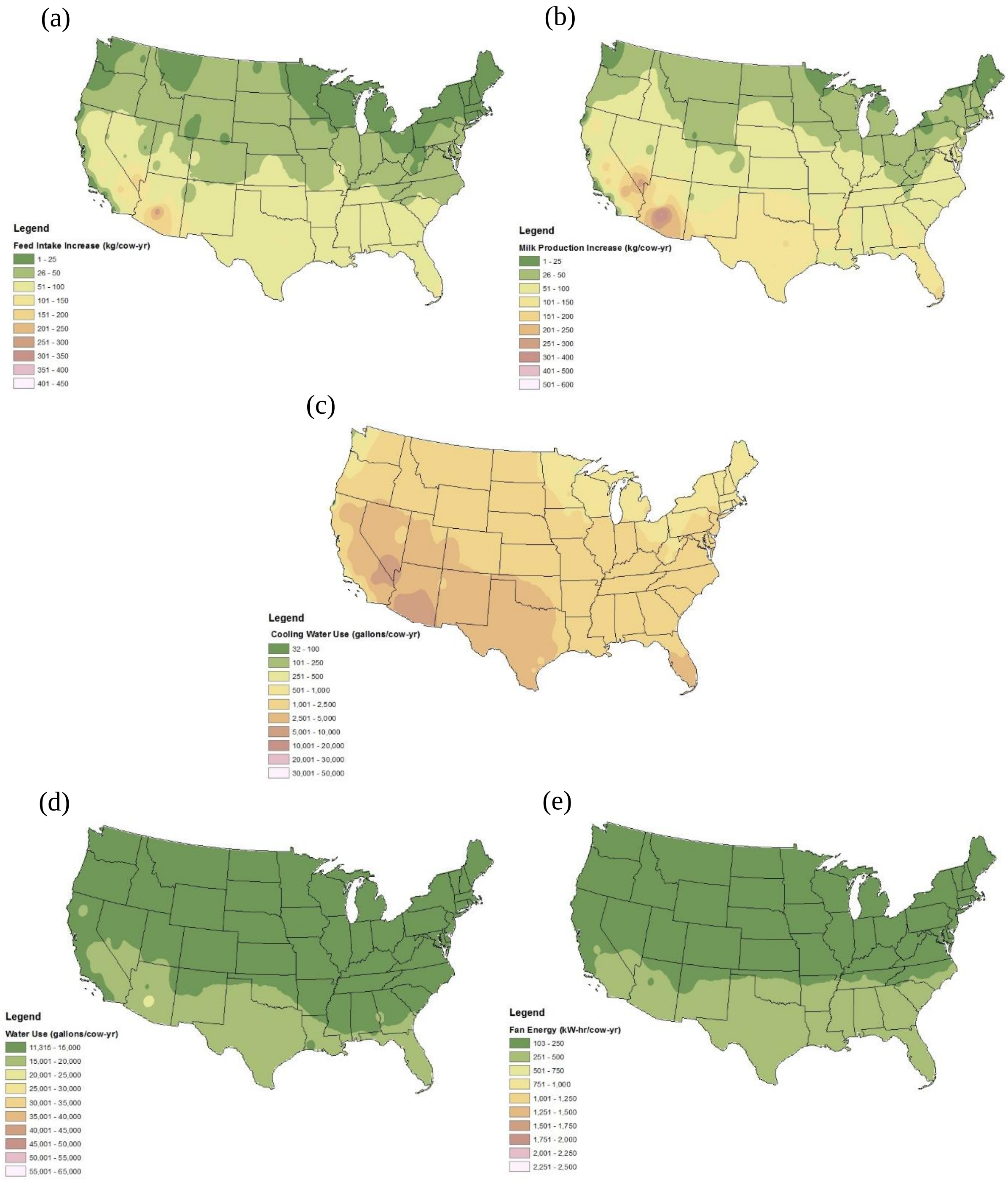


Figure 6.3. Contour maps showing the (a) increased feed intake and (b) increased milk production at the LVCE strategy compared to the LVCN strategy as well as the (c) cooling water use (d) total water use and (e) fan energy use at the LVCE strategy

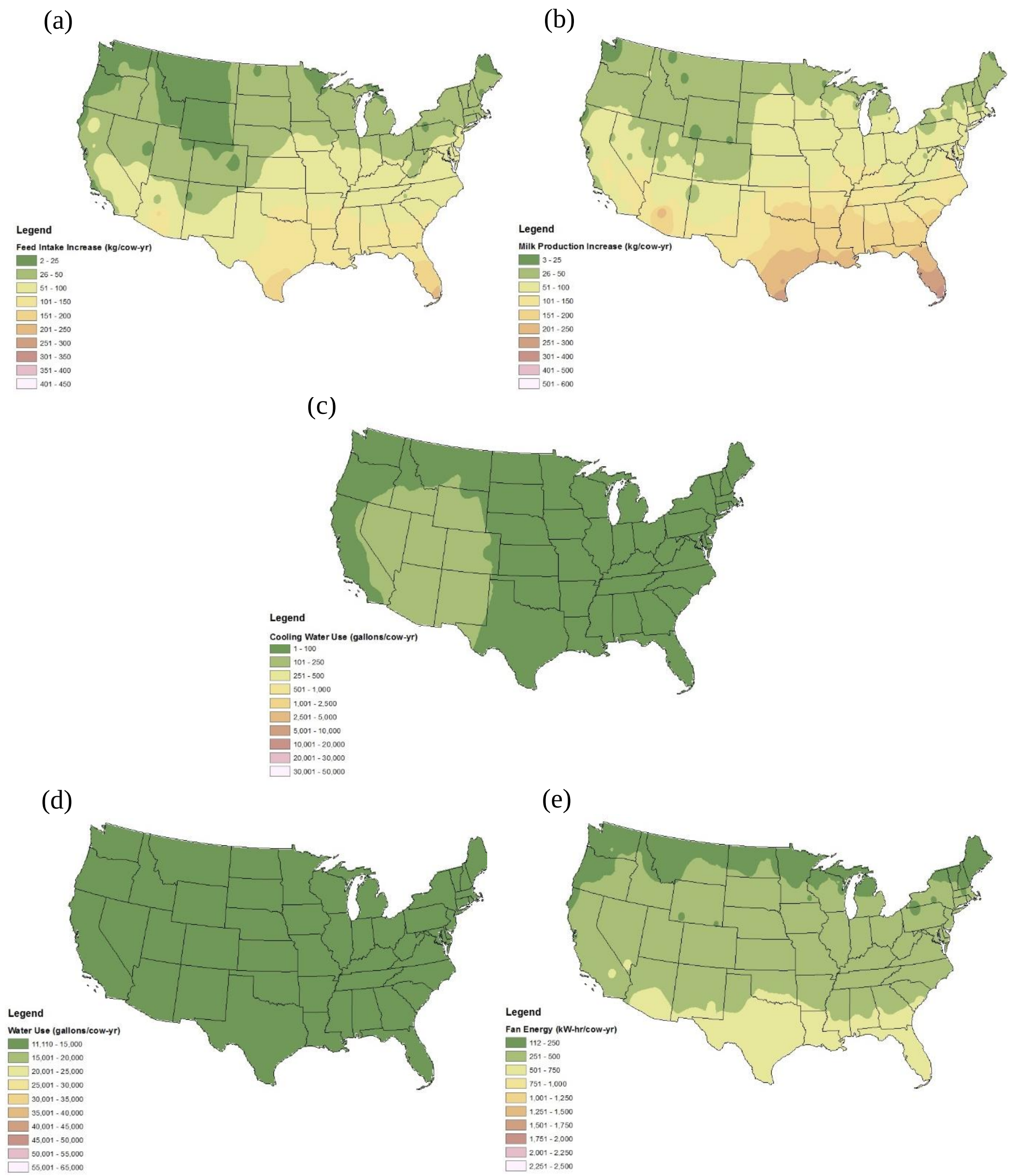


Figure 6.4. Contour maps showing the (a) increased feed intake and (b) increased milk production at the MVCN strategy compared to the LVCN strategy as well as the (c) cooling water use (d) total water use and (e) fan energy use at the MVCN strategy

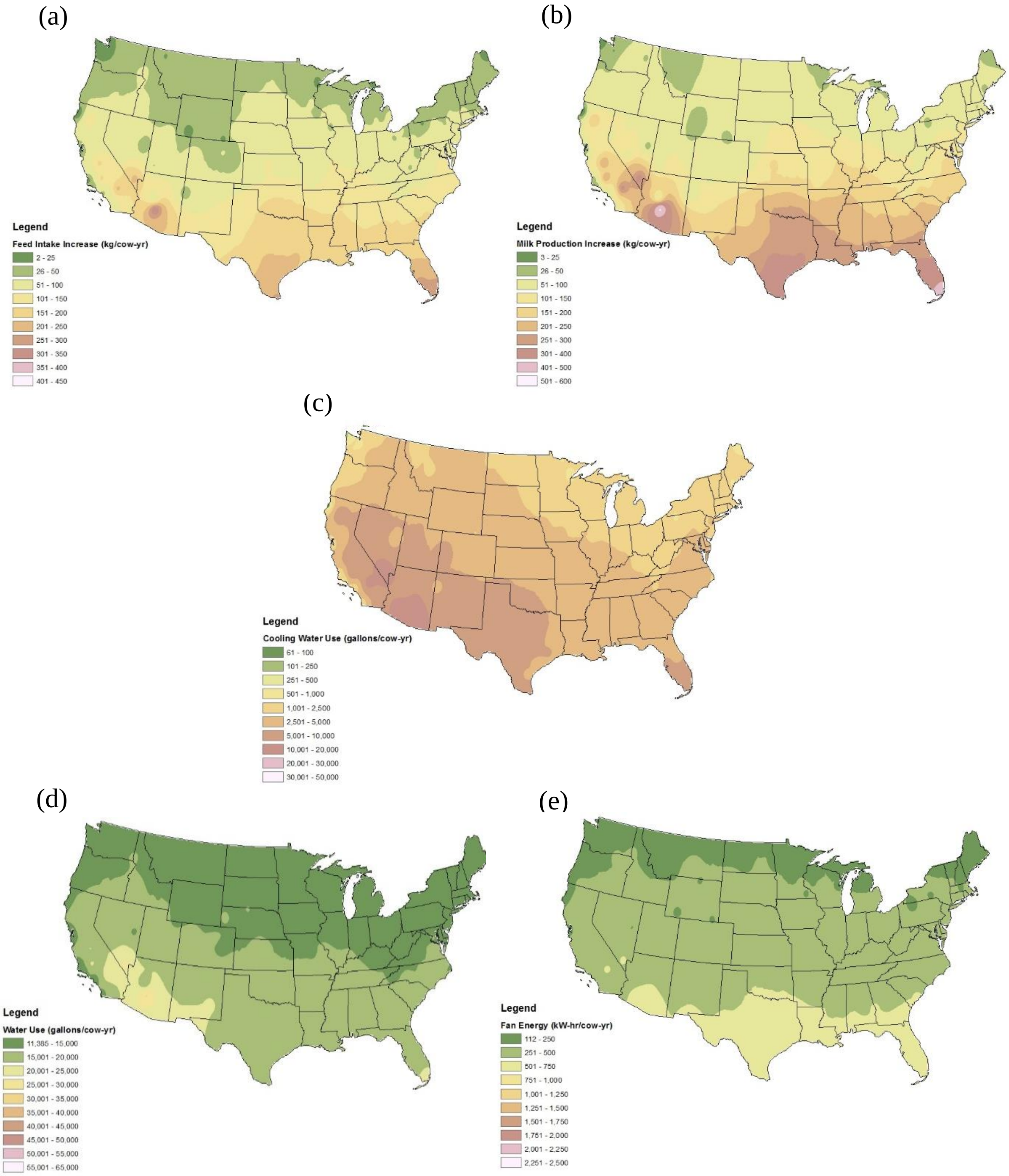


Figure 6.5. Contour maps showing the (a) increased feed intake and (b) increased milk production at the MVCE strategy compared to the LVCN strategy as well as the (c) cooling water use (d) total water use and (e) fan energy use at the MVCE strategy

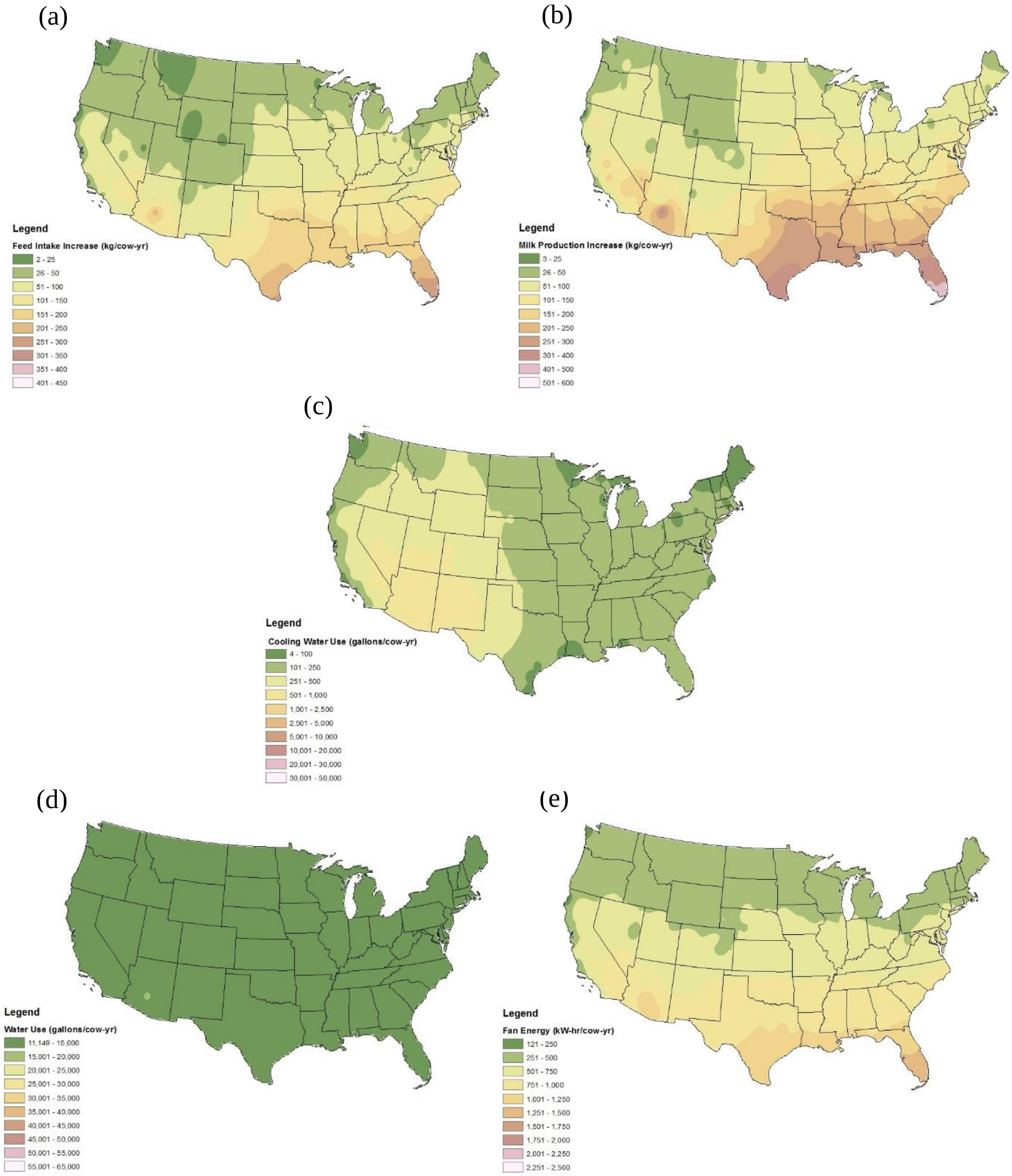


Figure 6.6. Contour maps showing the (a) increased feed intake and (b) increased milk production at the HVCN strategy compared to the LVCN strategy as well as the (c) cooling water use (d) total water use and (e) fan energy use at the HVCN strategy

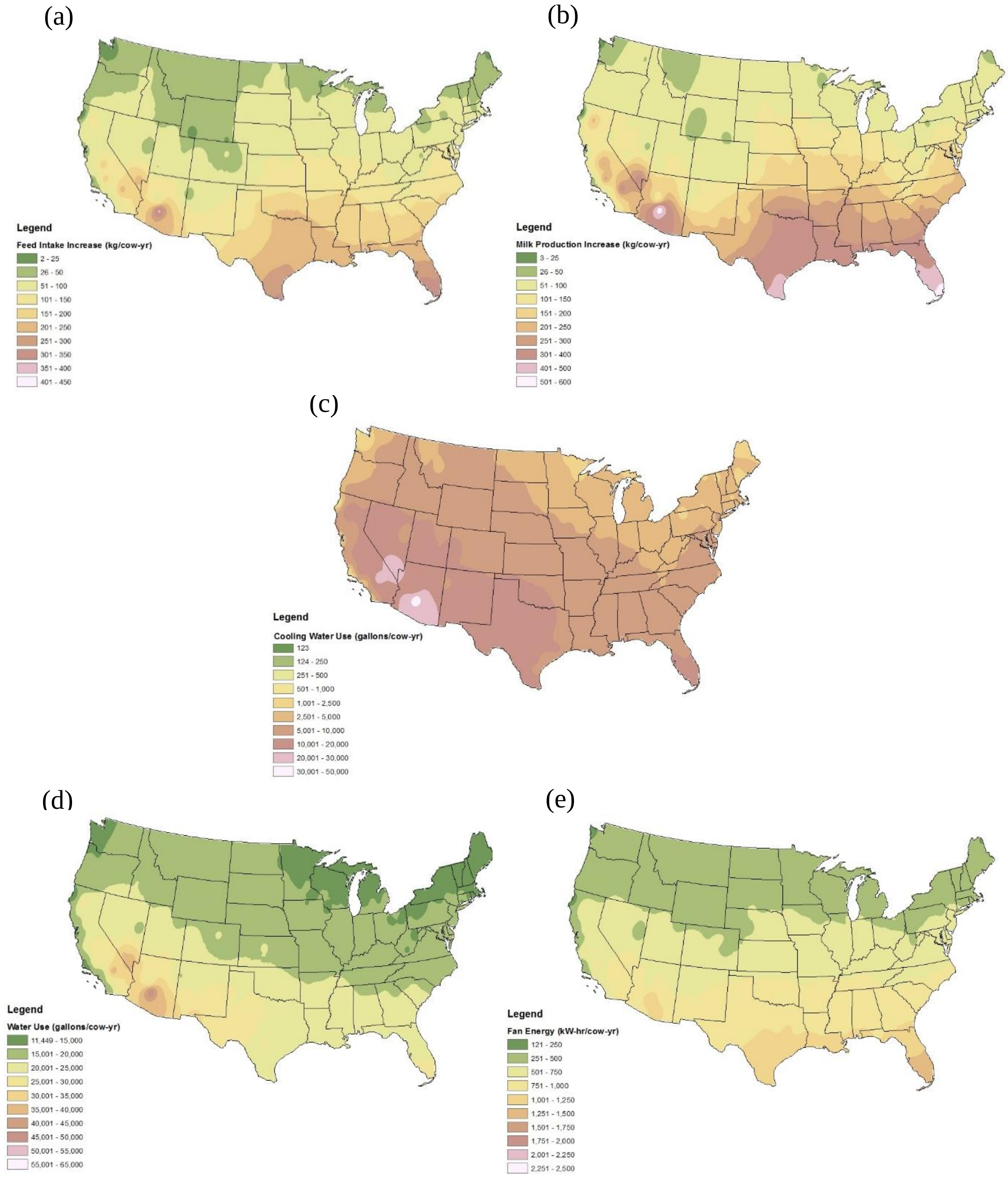


Figure 6.7. Contour maps showing the (a) increased feed intake and (b) increased milk production at the HVCE strategy compared to the LVCN strategy as well as the (c) cooling water use (d) total water use and (e) fan energy use at the HVCE strategy

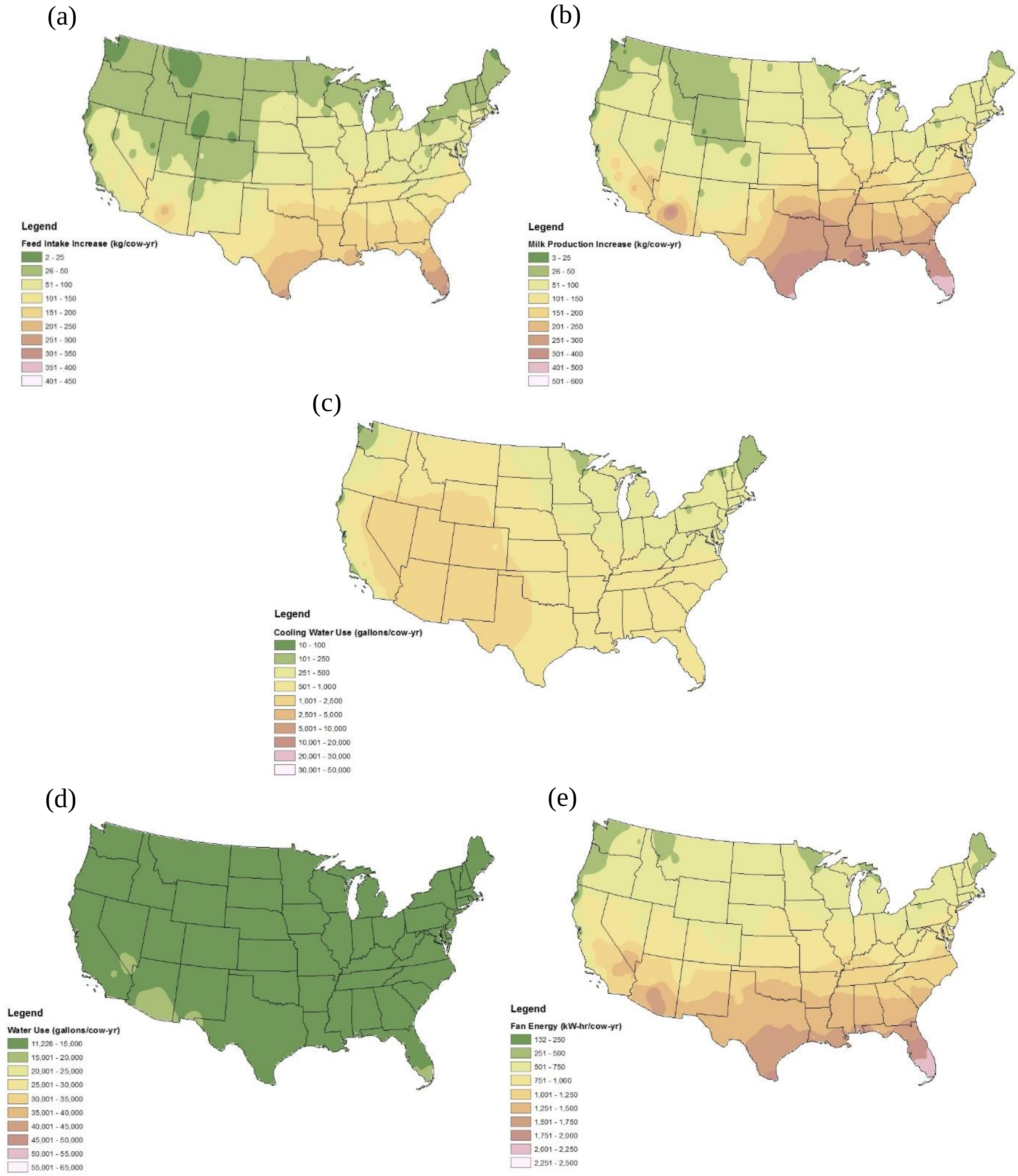


Figure 6.8. Contour maps showing the (a) increased feed intake and (b) increased milk production at the VHVCN strategy compared to the LVCN strategy as well as the (c) cooling water use (d) total water use and (e) fan energy use at the VHVCN strategy

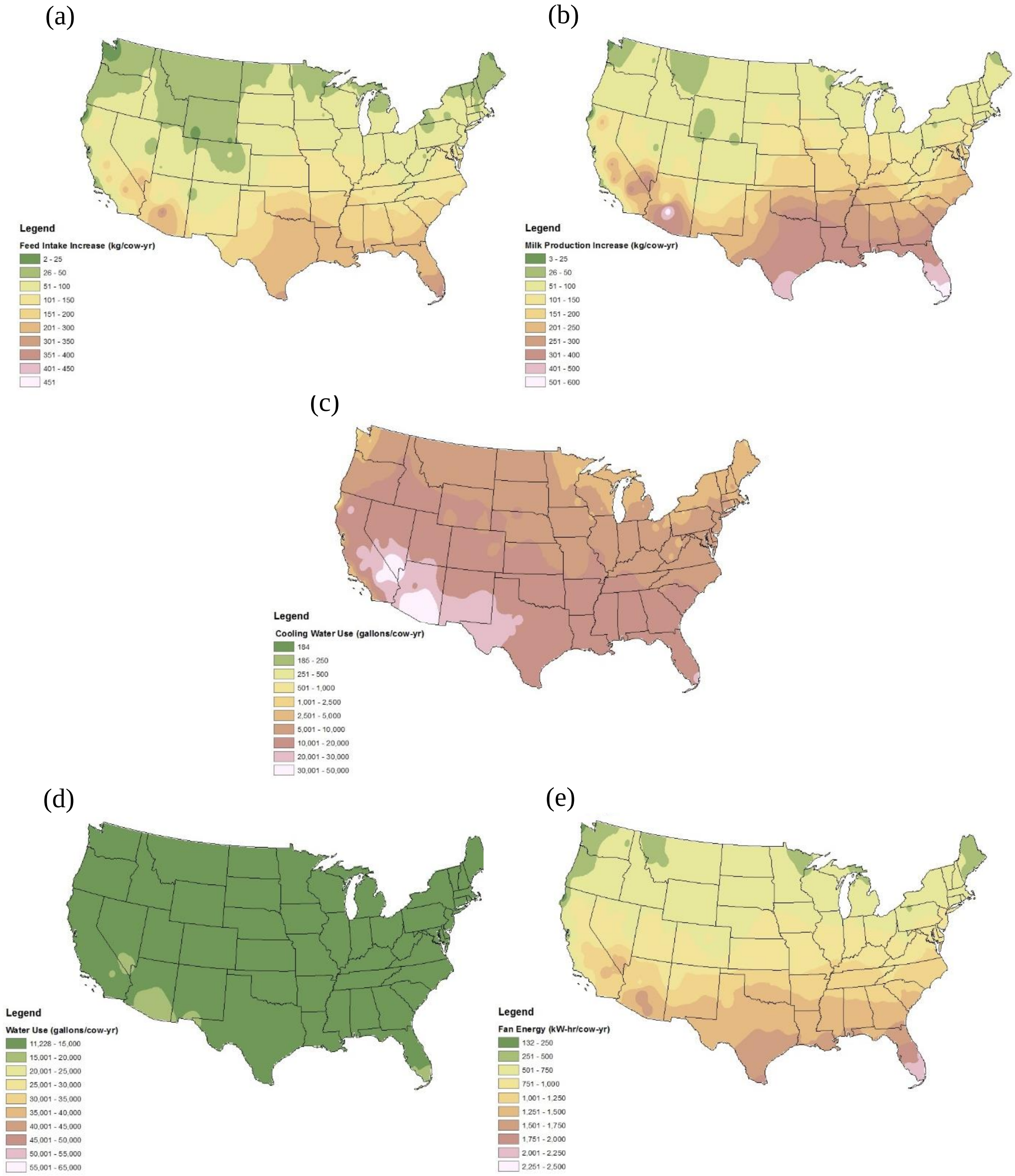


Figure 6.9. Contour maps showing the (a) increased feed intake and (b) increased milk production at the VHVCE strategy compared to the LVCN strategy as well as the (c) cooling water use (d) total water use and (e) fan energy use at the VHVCE strategy

It would be up to the producer to apply equation 6.2 to the previous maps in order to determine which strategy is the most economical to implement in their region. They could use the following steps to determine which cooling strategy to use in their area:

1. Determine the increase in milk production in your region for each strategy,
2. Multiply the milk production increase by your milk price, or a milk price you are comfortable with for long term planning,
3. Take the cooling water use for that same strategy and subtract the cooling water use for the LVCN strategy,
4. Multiply this water usage by the water price in your region,
5. Repeat steps 3 and 4 for fan energy use, and finally,
6. Using Equation 6.2, determine the ROI for each strategy.

For example, for the stations in Wisconsin and California that were analyzed in Chapter 4, the single ROI can be calculated using equation 6.2 for each of the cooling strategies. Table 6.2 outlines each of the cooling strategies with the values for feed intake, milk production, water and fan use and the resulting ROI for that strategy. The ROI's were calculated by using constants for both states. A milk price of \$0.363 kg^{-1}, which is the U.S. average milk price for April 2017 (USDA, 2017) was assumed. Due to the fluctuation in water cost throughout the U.S, water was assumed to be \$0.0015 $gallon^{-1}$ for both states and fan energy cost was set at \$0.12 $kW\text{-}hr^{-1}$ (Jiang, 2011). The total ROI column in Table 6.2 shows the increase or decrease in ROI from the LVCN strategy, which is denoted as "Baseline."

Table 6.2. Example of finding the total ROI for two stations, CA (SN: 723890) and WI (SN: 726435) by using equation 6.2

Station Number	**Cooling Strategy**	**Feed Intake, or Increase^{+} (kg cow^{-1} yr^{-1})**	**Milk Production, or Increase^{+} (kg cow^{-1} yr^{-1})**	**Total Water Use (gallon cow^{-1} yr^{-1})***	**Cooling Water Use (gallon cow^{-1} yr^{-1})**	**Fan Energy Use (kW hr cow $^{-1}$ yr^{-1})**	**ROI ($ yr^{-1})**	**Total ROI Increase or Decrease ($ yr^{-1})**
723890	LVCN	8,277	12,415	13,129	18	260	4,475,497	Baseline
723890	LVCE	115^{+}	172^{+}	16,347	4,089	257	4,532,071	56,574$^{+\&}$
723890	MVCN	83^{+}	125^{+}	13,127	64	441	4,499,108	23,611^{+}
723890	MVCE	158^{+}	237^{+}	20,153	7,955	436	4,528,548	53,052^{+}
723890	HVCN	113^{+}	170^{+}	13,376	336	782	4,474,004	-1,493
723890	HVCE	169^{+}	254^{+}	28,087	15,919	772	4,482,491	6,994^{+}
723890	VHVCN	120^{+}	180^{+}	14,016	984	1,143	4,433,529	- 41,968
723890	VHVCE	171^{+}	257^{+}	36,033	23,874	1,127	4,428,816	-46,681
726435	LVCN	8,387	12,580	11,544	2	157	4,547,798	Baseline
726435	LVCE	23^{+}	34^{+}	12,289	926	156	4,558,821	11,023^{+}
726435	MVCN	33^{+}	49^{+}	11,535	19	248	4,554,711	6,913^{+}
726435	MVCE	47^{+}	71^{+}	13,136	1,803	2467	4,559,930,	12,132$^{+\&}$
726435	HVCN	43^{+}	65^{+}	11,631	127	410	4,540,855	-6,943
726435	HVCE	54^{+}	81^{+}	14,927	3,607	408	4,541,571	-6,227
726435	VHVCN	46^{+}	68^{+}	11,854	354	586	4,520,548	-27,250
726435	VHVCE	55^{+}	82^{+}	16,725	5,409	583	4,518,452	-29,346

*Includes drinking water and water used for the cooling strategy

$^{+}$Denotes an increase from the original value at the LVCN strategy

$^{\&}$Denotes the highest ROI for that station

The total water use column in Table 6.2 includes the water needed by the cow for cooling by either sprinkler potential or evaporative cooling pad requirement as well as their drinking water. Since the cattle will be drinking water regardless of the cooling strategy, it was neglected in the total ROI calculation. After calculating the total ROI for each cooling strategy for each station in California and Wisconsin, the most economical cooling strategy for California was the LVCE strategy and for Wisconsin was the MVCE strategy. The same procedure can be replicated for each station to determine the most economical cooling strategy for dairy cattle, with fixed water and fan energy costs. Figure 6.10 shows a contour map of the U.S. using the same costs for milk, water, and fan energy as used in Table 6.2.

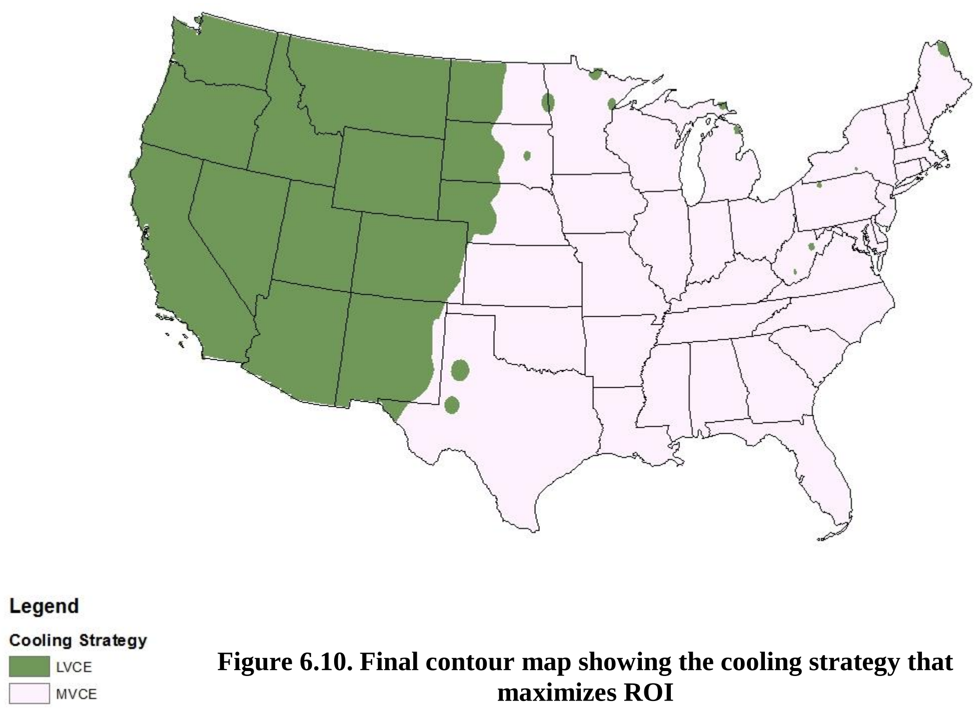


Figure 6.10. Final contour map showing the cooling strategy that maximizes ROI

The overall ROI map just shows one scenario with fixed prices. Keep in mind that the economic cost does not include potential loss of a fetus or lower pregnancy rates due to heat stress, equipment costs, labor, or maintenance costs. Equation 6.2 also doesn't include a water conservation factor, which could impact certain regions during a drought.

6.2 Conclusions

A thermal regulation barn and cow model was developed in order to assess a cow's thermal environment to determine if she is in heat stress and what cooling strategies have the greatest cooling potential as well as being economical for the producer. The main conclusions of this project are as follows:

- The developed cow model's results were comparable to published field data including accurately predicting the cow's core and skin temperatures as well as the sensible and latent heat production
    - The respiration rates predicted by this model were generally higher than the field data respiration rates
    - The cow's sweating rates were comparable to field data as well except at low airspeeds or in very humid environments due to the model not including sweat that is being pooled on the cow's skin surface, but rather only accounts for the amount of sweat that the environment can handle

- CDH developed for this project proved to be a useful thermal indicator of assessing heat stress and the magnitude of heat stress on the cow, especially useful for comparing various industry recommended thermal indices

- Using CDH, it was shown that CCI is a better indicator of the thermal environment, compared to THI, since it includes a correction factor for the airspeed, but improvements with CCI can still be made
- Multiple maps were created along with an equation that producers can use to determine which cooling strategy is the most economical for their area

The next section looks at possible improvements to the current model and suggests adding other cooling strategies as well as future work that could be done as a result from the development of this barn and cow thermoregulation model.

6.3 Future Work

All research that has been done can always be improved or shows possible improvement for other research in order to better the science community. For this project, a couple improvements could have been made in order to improve the model. First of all, this model only considered the energy lost through the potential to sweat and did not consider the energy lost through sweat that the environment could not evaporate. In other words, this model did not allow the cow to sweat no matter what, but just allowed her to sweat only what the environment could handle. Future research could look at how much energy is associated with a drop of sweat, which could then be applied to this model. Equation 6.3 would need to be used to find the sensible enthalpy associated with pooled sweat:

$$H_{sweat} = \dot{m}_{sweat} * c_{p,l} * (\Delta T) \tag{6.3}$$

where,

H_{sweat} = enthalpy of sweat

$\dot{m}_{sweat}$ = mass flow of swat ($kg_w\ s^{-1}$)

ΔT = a representative temperature difference (°C).

This model also took into consideration the roof insulation, but further research could be done on the influence of that roof insulation. One could look at increasing or decreasing the roof insulation resistance and the impact it has on the heat dissipation of the cow, since radiation had a major impact on the cow's sensible heat loss (Chapter 4).

Further research can be done from the development of this model. First, a new heat stress index could be created by adjusting THI to account for airspeed and/or radiation. Also, another cooling strategy could be incorporated into this model. One could look at ventilating the barn at the MVC rate (1 m s^{-1}) and then implementing stir fans throughout the barn to achieve the HVC or VHVC rate just over the cows to cut down on fan energy cost while increasing milk production. The stir fans could be monitored to turn on/off when evaporation potential can be utilized and then determine the overall barn ROI. Finally, with slight improvements to the model, an app could be created for producers that allows them to enter their location, milk price, water cost, and fan cost and outputs the most economical cooling strategy for their facility.

Finally, this model can be improved even further if other research is done and/or updated. There is a need in the research community for updating the sensible and latent heat loss which accounts for the updated genetics in today's dairy industry. The cow model applied a factor of 1.3 to the current best available published sensible and latent heat loss, but updated values would be more of an improvement. Also, there needs to be more research done on the impact on milk production from feed and water intake or even a correlation between milk production and a thermal index or a correlation of feed and water intake and a thermal index.

Overall, this model proved the need for a new thermal index that the dairy industry should be using to asses heat stress besides THI; one that takes into consideration more of the cow's environment such as airspeed over the cow and long-wave radiation typical for housed cows.

This developed model can help producers implement the best cooling strategy into their facility in order to reduce heat stress for their dairy cattle and overall increase milk production and their ROI. This model could also lead to better sprinkler sensor development to analyze the environment's potential to evaporate sweat, and then turn on/off those sprinklers to prevent water wastage.

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APPENDIX A. PSYCHROMETRIC CALCULATIONS

Water Vapor Partial Pressure

Saturated Air (P_{ws}; Pa)

$$P_{ws} = e^{(\frac{a_1}{T_{db}}+a_2+a_3*T_{db}+a_4*T_{db}^2+a_5*T_{db}^3+a_6*T_{db}^4+a_7*\ln(T_{db}))}$$

where,

T_{db} = dry bulb temperature (K).

If the temperature is between -100°C to 0°C, then

$a_1 = -5.6745359\ E+\ 03$

$a_2 = 6.3925247\ E+\ 00$

$a_3 = -9.677843\ E-\ 03$

$a_4 = 0.622157\ E-\ 06$

$a_5 = 2.0747825\ E-\ 09$

$a_6 = -9.484024\ E-\ 13$

$a_7 = 4.1635019\ E+\ 00.$

If the temperature if between 0°C to 200°C then

$a_1 = -5.8002206\ E+03$

$a_2 = 1.3914993\ E+00$

$a_3 = -0.4864023\ E-01$

$a_4 = 4.1764768\ E-\ 05$

$a_5 = -1.4452093\ E-08$

$a_6 = 0.0$

$a_7 = 6.5459673\ E+00.$

Unsaturated Air (P_w; Pa)

$$P_w = P_{ws} * \omega$$

where,

ω = relative humidity (decimal form).

Relative Humidity (ω; %)

$$\omega = \frac{p_w}{p_{ws}} * 100$$

Humidity Ratio

Saturated Air (W_s; $kg_w\ kg_a^{-1}$)

$$W_s = 0.62198 * \frac{P_{ws}}{(P - P_{ws})}$$

Unsaturated Air (W; $kg_w\ kg_a^{-1}$)

$$W = 0.62198 * \frac{P_w}{(P - P_w)}$$

Specific Volume (ν; $m^3\ kg_a^{-1}$)

Dry Air

$$\nu_a = \frac{1}{P} * R_a * T_{db} * (1 + 1.6087 * W)$$

Moist Air

$$\nu_w = \frac{\frac{1}{P} * R_a * T_{db} * (1 + 1.6087 * W)}{1 + W}$$

where,

T = dry bulb temperature (K).

Density (ρ; kg_a m^{-3})

$$\rho = \frac{1}{\nu}$$

Dew Point Temperature (T_{dp}; °C)

If the temperature is between -60°C to 0°C, then

$$T_{dp} = -60.45 + 7.0322 * \ln(P_w) + 0.3700 * (\ln(P_w))^2.$$

If the temperature is between 0°C to 70°C, then

$$T_{dp} = -35.957 - 1.8726 * \ln(P_w) + 1.1689 * (\ln(P_w))^2.$$

Enthalpy (h; kJ kg^{-1} K^{-1})

$$h = cp_a * T_{db} + W * \left(h_{fg} + cp_w * T_{db}\right)$$

where,

T_{db} = dry bulb temperature (°C).

Wet Bulb Temperature (T_{wb}; °C)

The wet bulb temperature was calculated iteratively using the process of adiabatic saturation (Albright, 1990).

APPENDIX B. FREESTALL BARN DESIGN

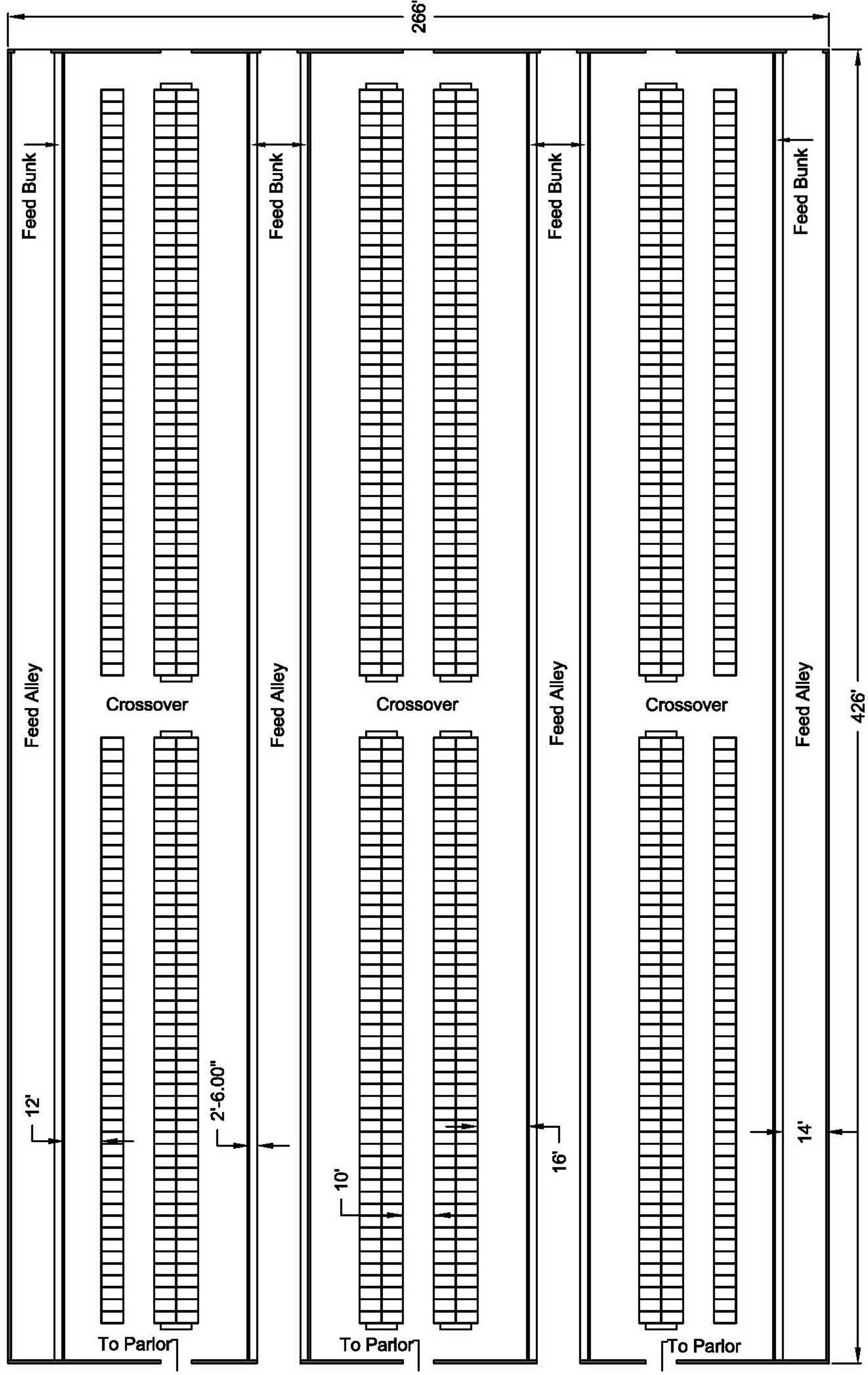

Figure B. 1. Freestall barn layout

APPENDIX C. BUILDING THERMAL ANALYSIS

Each building component has a thermal resistance to heat flow. Table C.1 outlines all of the resistances of each building material that was used.

Table C. 1. Thermal resistances of building materials

Material	Resistance (m^2 K W^{-1})	Notes:
Roof	1.22	without air films
Wall	1.21	with air films
End wall	2.65	with air films
Curtain	0.47	with air films
Fan	1.21	with air films
Concrete	0.57	with air films
Still air	0.12	--

These resistances were used to determine the UA-value of the building and were determined by adding the individual components that make up the modeled barn. Multiple input variables (Table C.2) were used to determine the total area of each component, and Table C.3 outlines each resistance value of those components.

Table C. 2. Input variables to determine component areas

Input Variable	Value	Unit
Building width	129.8	m
Building length	81.1	m
Side-wall height	4.27	m
Concrete end-wall height	2.44	m
Concrete side-wall height	0.3	m
Curtain height	3.66	m
Inside air temperature	20	°C
Roof pitch	1√12	--

Table C. 3. Component resistance values

Component	Thickness (m)	Thermal conductivity ($W\ m^{-1}\ °C^{-1}$)	Resistance value ($m^2\ °C\ W^{-1}$)
Outside air-film	--	--	0.03
24 gauge steel	0.0065	50.20	0.000129
2x4 Wood – Douglas Fir	0.09	0.14	0.64
2x6 Wood – Douglas Fir	0.14	0.14	1.00
2x12 Wood – Douglas Fir	0.29	0.14	2.04
Acrolite Insulation	--	--	1.06
Curtain Material	--	--	0.26
Concrete	0.30	0.72	0.42
Fan Shutters*	--	--	0.30
Inside air-film	--	--	0.12

*R-value for house siding,

Finally, these tables were used to produce Table C.4. The total resistance for each heat transfer path includes air-films. These paths were used to calculate the overall building UA-value, 10415 W $°C^{-1}$. This overall UA-value was the sum of the section UA-values.

Table C. 4. UA-value determination

Section	Heat transfer paths	Total resistance ($m^2\ °C\ W^{-1}$)	Area (m^2)	Section UA-value ($W\ °C^{-1}$)
End Walls (x2)	Concrete	0.57	197.75	1120
	Wood	2.20	57.05	
	Insulation	1.21	228.21	
Curtain Wall	Curtain	0.41	427.54	1187
	Curtain (wood)	1.05	47.50	
	Wall (wood)	2.20	7.89	
	Wall (insulation)	1.21	31.56	
	Concrete	0.57	39.59	
Fan Wall	Concrete	0.57	39.59	456.7
	Wall (wood)	2.20	102.90	
	Wall (insulation)	1.21	411.59	
	Fans*	0.45	0	
Roof	Wood	3.25	2112.78	7651.4
	Insulation	1.21	8451.11	

*Assumed area for the fans is 0 since a fan number was not determined, and it is also conservative

APPENDIX D. VENTILATION CURVE DEVELOPMENT

Ventilation curves were determined using mass and energy balances. The following figures (D.1 – D.4) show the resulting graphs for each ventilation rate chosen for this project, which include the 2°C Rule (LVC), 1 m s^{-1} (MVC) 2 m s^{-1} (HVC), and 3 m s^{-1} (VHVC). Since the curves for each ventilation rate where not uniform, best fit lines were used between selected outdoor temperature ranges. The ranges for each curve and each ventilation rate are: -22°C to -5°C, -5°C to 12°C, 12°C to 18°C, 18°C to 20°C, and 20°C to 31°C.

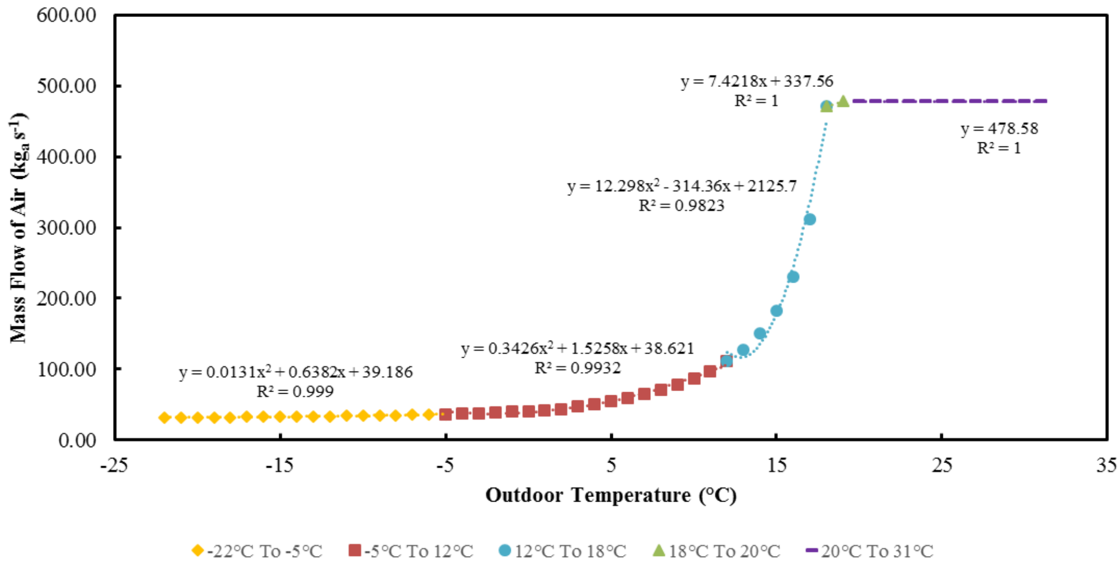

Figure D. 1. LVC curve

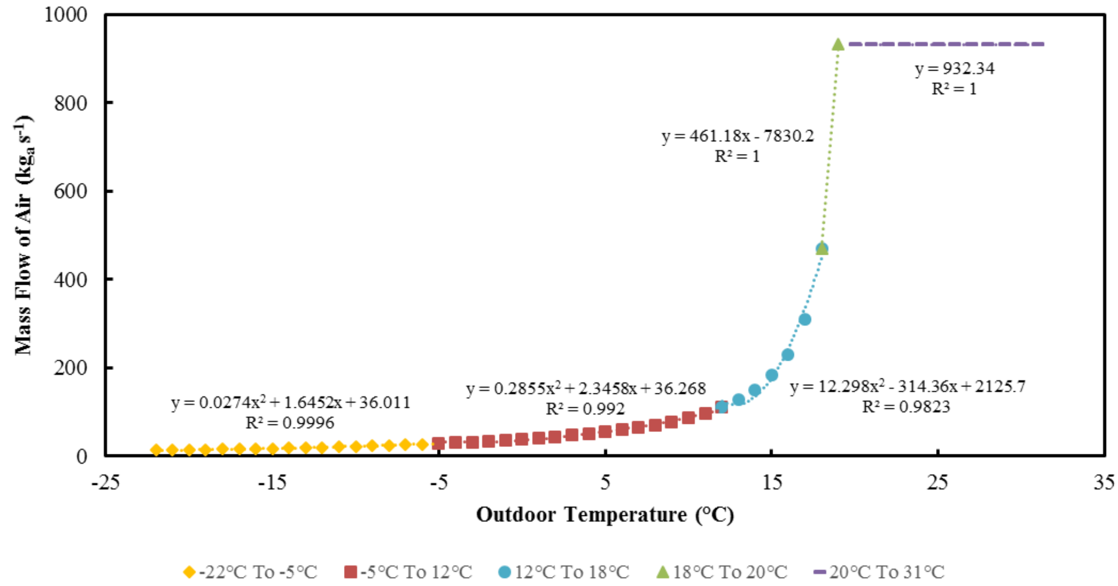

Figure D. 3. MVC curve

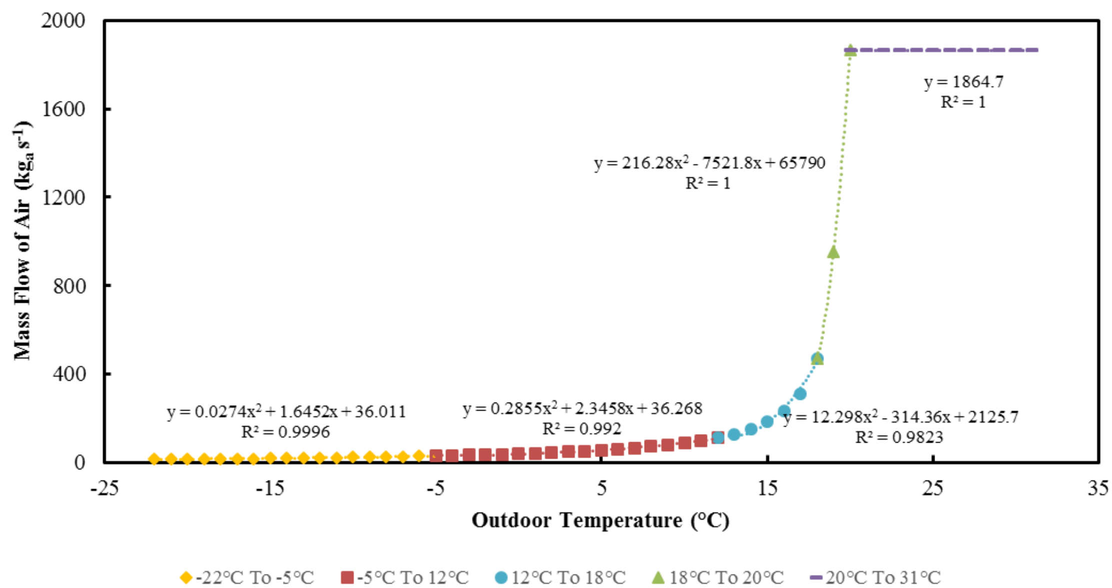

Figure D. 5. HVC curve

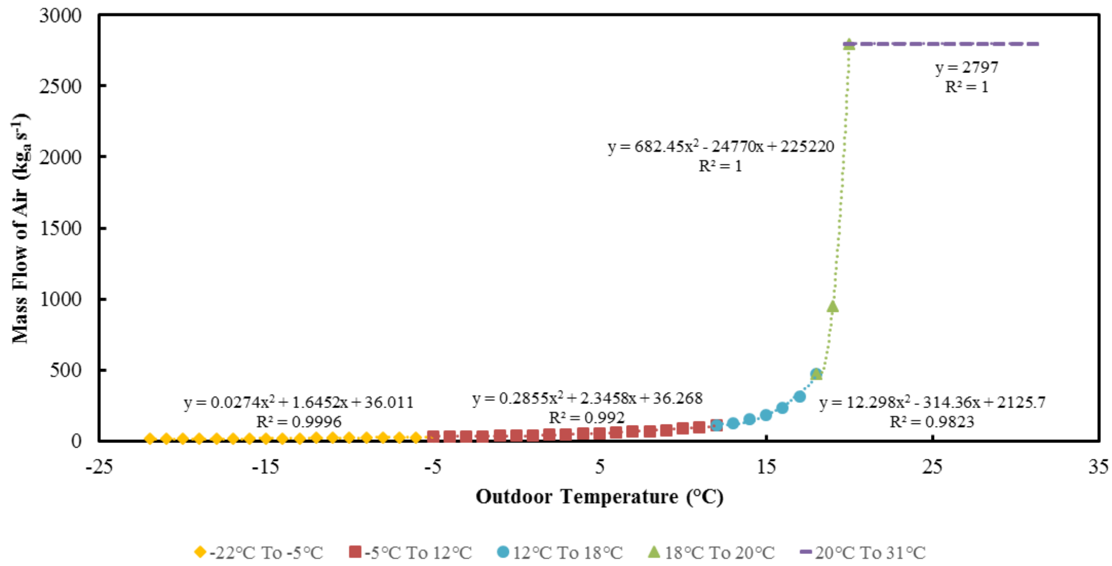

Figure D. 6. VHVC curve

These curves were then used to determine the velocity in the barn by using Equations D.1 and D.2.

$$\dot{V} = \dot{m} * v_{evap} \tag{D.1}$$

where,

$\dot{m}$ = mass flow rate ($kg_a\ s^{-1}$)

$\dot{V}$ = ventilation rate ($m^3\ s^{-1}$)

v_{evap} = specific volume of air in barn ($m^3\ kg_a^{-1}$)

$$\vec{V} = \frac{\dot{V}}{A_{barn}} \tag{D.2}$$

where,

$\vec{V}$ = velocity ($m\ s^{-1}$)

A_{barn} = average cross-sectional area of barn (773.2 m^2).

APPENDIX E. MODEL CONSTANTS AND EQUATIONS

Table E. 1. Table of model's constants and equations

Equation*	Units	Description
$A_{cow} = 0.14 * m_{cow}^{0.57}$	m^2	Surface area of cow [¥ ¶]
$D_{cow} = 0.06 * m_{cow}^{0.39}$	m	Diameter of cow [†]
$h_{coat} = 11.6$	$W\ m^{-2}\ K^{-1}$	Coat heat transfer coefficient [†]
$L_{coat} = 0.01$	m	Length of coat [†]
$L_{cow} = \frac{(A_{cow} - D_{cow}^2 * \frac{\pi}{2})}{D_{cow} * \pi}$	m	Length of cow [†]
$m_{cow} = 600$	kg	mass of cow [† ¶ »]
$RR_{min} = 12$	BPM	Minimum respiration rate [¥]
$RR_{max} = 120$	BPM	Maximum respiration rate
$RR = 6.25 * T_{db} - 80.47$	BPM	Respiration rate [₤]
$R_{t,min} = 0.0156$	$m^2\ K\ W^{-1}$	Minimum tissue resistance [† ϫ]
$R_{t,max} = \frac{(2.4 + 0.0054 * m_{cow})}{48}$	$m^2\ K\ W^{-1}$	Maximum tissue resistance [‡]
$R_t = 1.6212 * RR^{-1.029}$	$m^2\ K\ W^{-1}$	Tissue resistance [₤]
$SR_{min} = 14.4$	$g\ m^{-2}\ h^{-1}$	Minimum sweat rate [₴]
$SR_{max} = 288$	$g\ m^{-2}\ h^{-1}$	Maximum sweat rate [₴]
$THI = 0.72 * (T_{db} + T_{wb}) + 40.6$	dimensionless	Temperature Humidity Index [+]
$TV_{min} = 1.9$	$L\ breath^{-1}$	Minimum tidal volume [»]
$TV_{max} = 0.4 + 0.0064 * m_{cow}$	$L\ breath^{-1}$	Maximum tidal volume [»]
$TV = -9.4 * 10^{-5} * RR^2 - 1.8 * 10^{-4}$	$L\ breath^{-1}$	Tidal volume [₤]
$\alpha = 0.39$	dimensionless	Absorptivity of roof [ʅ]
$\varepsilon_{cow} = 0.98$	dimensionless	Emissivity of cow [¥]
$\varepsilon_{barn} = 0.92$	dimensionless	Emissivity of barn surfaces
$\varepsilon_{roof} = 0.90$	dimensionless	Emissivity of exterior roof [ʅ]
$\sigma = 5.669 * 10^{-8}$	$W\ m^{-2}\ K^{-1}$	Stefan-Boltzmann Constant

* All symbols and variables are described in the Nomenclature

[¶] Berman 2003
[‡] Berman 2004
[»] Berman 2005
[+] Hahn 2003
[ꭗ] McArthur 1981
[ƻ] McArthur 1987
[†] McGovern and Bruce 2000
[ↄ] Suehrcke 2008
[₤] Thompson 2011
[¥] Thompson 2014

APPENDIX F. VBA CODE

```
Private Sub CommandButton1_Click()
Dim st, model, q As String
Dim ad$(10000), at$(10000), solar(10000), TotalCloud(10000), DryBulb(10000), RelativeHumidity(10000),
Pressure(10000), Wind(10000)
model = "Model"

'This file will contain summary data for each run that will be used for contour plotting
ContourData$ = "c:\TMY3Data\ContourData.csv"

Open "c:\TMY3Data\TMY3Stations.csv" For Input As #10

Do Until EOF(10)
STrun = STrun + 1
Worksheets(model).Range("m1") = STrun

''''''''''''''Between Runs we Need to Zero Summations''''''''''''''''''''''''''''''''''''''
THI_max = 0#: CCI_max = 0#: EvapCooler_Hours = 0#: THI_PreviousHour = 0#
THI_Events8 = 0: THI_Events10 = 0: THI_Events12 = 0: THI_Events14 = 0
CCI_Events8 = 0: CCI_Events10 = 0: CCI_Events12 = 0: CCI_Events14 = 0

TDBsum = 0#: WTDBsum = 0#: FeedSum = 0#
THI_1 = 0#: THI_2 = 0#: THI_3 = 0#: THI_4 = 0#
CCI_1 = 0#: CCI_2 = 0#: CCI_3 = 0#: CCI_4 = 0#
RR_1 = 0#: RR_2 = 0#: RR_3 = 0#: RR_4 = 0#
RRnew_1 = 0#: RRnew_2 = 0#: RRnew_3 = 0#: RRnew_4 = 0#
CDH_1 = 0#: CDH_2 = 0#: CDH_3 = 0#: CDH_4 = 0#
THI_Events8 = 0#: THI_Events10 = 0#: THI_Events12 = 0#: THI_Events14 = 0#
CCI_Events8 = 0#: CCI_Events10 = 0#: CCI_Events12 = 0#: CCI_Events14 = 0#
HTRRnew = 0#: HTRR = 0#: EITSum = 0#
Total_WaterUse = 0#
Sprinkler_Gallons_TotalPerHead = 0#: AvailableSprinklingHours = 0#
EvapCooler_Gallons_TotalPerHead = 0#
Win_TotalPerHead = 0#
FanEnergy = 0#
FeedSum_new = 0#
''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''

Input #10, SN, Location$, State$, Latitude, Longitude, TgroundF, Tground, EVAPeff, VelocityCase$
EVAPeff = EVAPeff / 100#
TDB_Evap_ON = 23#

TMY3Data$ = "c:\TMY3Data\" & SN & "TYA.csv"

Open TMY3Data$ For Input As #1
Input #1, SN_a, Location_a$, State_a$, junk, Latitude_a, Longitude_a, Elevation_a
Line Input #1, q$
```

```
'Line Input #1, q$

OutputData$ = "c:\TMY3Data\" & SN & "_" & EVAPeff & "_" & VelocityCase$ & ".csv"

''''''''''''''''''''''''''Getting All 71 Columns of TMY3 Data''''''''''''''''''''''''''''''''''''''''
For i = 1 To 8760
Input #1, ad$(i), at$(i), aa, a, solar(i), a, a, a, a, a, a, a, a, a, a, a, a, a, a, a, a, a, a, a, TotalCloud(i), a, a, a, a, a, DryBulb(i), a, a, a, a, a, RelativeHumidity(i), a, a, Pressure(i), a, a, a, a, a, Wind(i), a, a, a, a, a, a, a, a, a, a, a, a, a, a, a, a, a, a, a, a, a, a, a
Next i

a = 0#

Close #1
''''''''''''''''''''''''''''Now We Have Required TMY3 Data for the Year'''''''''''''''''''''''''''''''''''

'''''Set the Header for the Output Data File''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''
Open OutputData$ For Append As #2
Write #2, "SN", "Loc", "St", "Lat", "Lon", "Date", "Time", "JD", "Vdot", "Vel", "W", "WEVAP", "W_center", "TDB", "TDBEVAP", "TDB_center", "Tsa", "T_roof", "TMRT", "Tcore", "Tskin", "Rt_actual", "h_ov", "Fi", "Fi_CCI", "Win", "THP_Diet", "THP_Opt", "LHP_Opt", "SHP_Opt", "LHP_Model", "SHP_Model", "Qdot_store", "Q_sensible_breathing", "Q_latent_breathing", "Qdot_respiration", "Qdot_feedandwater", "Qdot_heat_to_skin", "Qcv", "Qrad", "Qdot_remaining_for_sweating", "Q_latent_evaporation_potential", "SR_Published", "SR_Reqd_for_Balance", "SR_Actual_Modeled", "Qdot_sweating_actual", "Heat_Difference", "Excess_Evap_Potential", "SprPotential", "AvailableSprHrs", "Spr_Gal_TotalPerHead", "EvapUse", "Evap_Gal_TotalPerHead", "FE", "TV", "RR", "RR_new", "EIT", "HTRR", "HTRRnew", "THI_1", "THI_2", "THI_3", "THI_4", "THI", "THI_Adjusted", "CCI_noRAD", "CCI_withRAD", "CCI_withMRT", "CDH", "CDH<1 Hours", "CDH<0.9 Hours", "CDH<0.8 Hours", "CDH<0.7 Hours", "THI8", "THI10", "THI12", "THI14", "CCI8", "CCI10", "CCI12", "CCI"
Close #2
''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''

  Tcore = 38.5 '°C
  Tref = 38.5 '°C
  Tskin = 35  '°C
  TDB_center = 20# 'Initial condition for cow heat production calculations in barn center scenario

'For i = 3 To 100
For i = 3 To 8762
Worksheets(model).Range("n1") = i
  JulianDay = 1# + (i - 2) / 24#
  TDB = DryBulb(i - 2)          '°C
  rh = RelativeHumidity(i - 2)   '%
  ATM = Pressure(i - 2) * 100#   'Pa
  ghi = solar(i - 2)            'W/m2
  Totcld = TotalCloud(i - 2)     'in tenths
  windspeed = Wind(i - 2)        'm/s
```

```
  p = ATM 'Pressure in Pa
  T = TDB 'Temperature in °C

'''' set initial parameters for calculation mode (constants)
   cpa = 1.006: cpw = 1.85: cpl = 4.186: hfg = 2501: ra = 287.1: rw = 461.5
    'kJ/kga-K    kJ/kgw-K     kJ/kgw-K      kJ/kg      J/kga-K     J/kgw-K

   rh = rh / 100!: T = T + 273.2  'Conversions
   w = 0: v = 0: h = 0: tdp = 0: twb = 0

'''ALL TAKEN FROM ALBRIGHT (psychrometric calculations)''''
'''' calculate the saturation vapor pressure

    If ((T - 273.2) < 0!) Then
     a1 = -5674.5359
     a2 = 6.3925247
     a3 = -0.009677843
     a4 = 0.000000622157
     a5 = 2.0747825E-09
     a6 = -9.484024E-13
     a7 = 4.1635019
    Else
     a1 = -5800.2206
     a2 = 1.3914993
     a3 = -0.04864023
     a4 = 0.000041764768
     a5 = -0.000000014452093
     a6 = 0!
     a7 = 6.5459673
    End If

    t1 = T
    b1 = a1 / t1 + a2 + a3 * t1 + a4 * t1 ^ 2 + a5 * t1 ^ 3 + a6 * t1 ^ 4 + a7 * Log(t1)
    Pws = Exp(b1)
    Pw = Pws * rh

'dew point temperature (tdp, °C)
    If ((T - 273.2) < 0) Then
      tdp = -60.45 + 7.03221 * Log(Pw) + 0.37 * (Log(Pw)) ^ 2
    Else
      tdp = -35.957 - 1.87261 * Log(Pw) + 1.1689 * (Log(Pw)) ^ 2
    End If

'humidity ratios (W, kgw/kga)
    w = 0.62198 * Pw / (p - Pw)
    WS = 0.62198 * Pws / (p - Pws)

'loop that determines wet bulb temperature (twb, °C)
```

```
    T = T - 273.2

    For tw1 = T + 273.2 To T + 273.2 - 30 Step -0.01

    If ((tw1 - 273.2) < 0!) Then
      a1 = -5674.5359
      a2 = 6.3925247
      a3 = -0.009677843
      a4 = 0.000000622157
      a5 = 2.0747825E-09
      a6 = -9.484024E-13
      a7 = 4.1635019
    Else
      a1 = -5800.2206
      a2 = 1.3914993
      a3 = -0.04864023
      a4 = 0.000041764768
      a5 = -0.000000014452093
      a6 = 0!
      a7 = 6.5459673
    End If
    b1 = a1 / tw1 + a2 + a3 * tw1 + a4 * tw1 ^ 2 + a5 * tw1 ^ 3 + a6 * tw1 ^ 4 + a7 * Log(tw1)
    pwstw = Exp(b1)
    wstw = 0.62198 * pwstw / (p - pwstw)
    h = cpa * T + w * (hfg + cpw * T)
    hs = cpa * (tw1 - 273.2) + wstw * (hfg + cpw * (tw1 - 273.2))
    hw = cpl * (tw1 - 273.2)
    rs = hs
    ls = h + hw * (wstw - w)
    Delta = rs - ls

    If (Delta <= 0.5) Then
    twb = (tw1 - 273.2)
    GoTo 10   'To get out of loop
    End If

    Next tw1

10

'specific volume (v, m3/kg)
      v = ((1 / p) * ra * (T + 273.2) * (1 + 1.6078 * w))

'enthalpy (h, kJ/kg)
    h = cpa * T + w * (hfg + cpw * T)

'Conditions Exiting Evaporative Cooler''''''''''''''''''''''''''''''''''
   TDBEVAP = TDB - EVAPeff * (TDB - twb)
```

```
    TWBEVAP = twb
    HEVAP = h
''''''''''''''We now know TDB and TWB leaving evaporative cooler. Need to find remaining properties''''''
    t1 = TDBEVAP + 273.15
     If ((t1 - 273.2) < 0!) Then
       a1 = -5674.5359
       a2 = 6.3925247
       a3 = -0.009677843
       a4 = 0.000000622157
       a5 = 2.0747825E-09
       a6 = -9.484024E-13
       a7 = 4.1635019
     Else
       a1 = -5800.2206
       a2 = 1.3914993
       a3 = -0.04864023
       a4 = 0.000041764768
       a5 = -0.000000014452093
       a6 = 0!
       a7 = 6.5459673
     End If
     b1 = a1 / t1 + a2 + a3 * t1 + a4 * t1 ^ 2 + a5 * t1 ^ 3 + a6 * t1 ^ 4 + a7 * Log(t1)
    PwsEVAP = Exp(b1)
    WsEVAP = 0.6221 * PwsEVAP / (ATM - PwsEVAP)

    t1 = TWBEVAP + 273.15
     If ((t1 - 273.2) < 0!) Then
       a1 = -5674.5359
       a2 = 6.3925247
       a3 = -0.009677843
       a4 = 0.000000622157
       a5 = 2.0747825E-09
       a6 = -9.484024E-13
       a7 = 4.1635019
     Else
       a1 = -5800.2206
       a2 = 1.3914993
       a3 = -0.04864023
       a4 = 0.000041764768
       a5 = -0.000000014452093
       a6 = 0!
       a7 = 6.5459673
     End If
     b1 = a1 / t1 + a2 + a3 * t1 + a4 * t1 ^ 2 + a5 * t1 ^ 3 + a6 * t1 ^ 4 + a7 * Log(t1)
    PwsStarEVAP = Exp(b1)
    WsStarEVAP = 0.6221 * PwsStarEVAP / (ATM - PwsStarEVAP)
    WEVAP_Step1 = (cpa * TWBEVAP + WsStarEVAP * (2501 + cpw * TWBEVAP) - cpa * TDBEVAP) -
cpl * WsStarEVAP * TWBEVAP
```

```
   WEVAP = WEVAP_Step1 / (2501 + cpw * TDBEVAP - cpl * TWBEVAP)
   pwEVAP = WEVAP * ATM / (0.6221 + WEVAP)
   RHEVAP = 100 * pwEVAP / PwsEVAP

   vEVAP = ((1 / ATM) * ra * (TDBEVAP + 273.2) * (1 + 1.6078 * WEVAP))
   HEVAP = cpa * TDBEVAP + WEVAP * (2501 + cpw * TDBEVAP)

'dew point temperature from evap (TDPEVAP, °C)
     If ((TDBEVAP - 273.2) < 0) Then
       TDPEVAP = -60.45 + 7.03221 * Log(pwEVAP) + 0.37 * (Log(pwEVAP)) ^ 2
     Else
       TDPEVAP = -35.957 - 1.87261 * Log(pwEVAP) + 1.1689 * (Log(pwEVAP)) ^ 2
     End If

''''''''''''''This section turns off the Evap Cooler if it is not needed''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''
Delta_W = (WEVAP - w) 'kgw/kga difference from evaporative cooler
If (EVAPeff = 0#) Then Delta_W = 0# 'Needed to do this due to small round-off errors that yielded evap cooler water use when eff=0

'This is where we check to see if evap cooling should be turned on. We are using TDB as our criteria since barns are not controlled on THI
If (TDB < TDB_Evap_ON Or EVAPeff = 0#) Then
TDBEVAP = TDB
WEVAP = w
Delta_W = 0#
TDPEVAP = tdp
RHEVAP = rh * 100
PwsEVAP = Pws
pwEVAP = Pw
vEVAP = v
HEVAP = h
End If

EvapUseFlag = 0#
If (TDB >= TDB_Evap_ON And EVAPeff > 0) Then EvapUseFlag = 1#
'
''''''''''''''''''''''''''''''''Now we are to determine the psychrometric conditions at the center of the barn''''''''''''''''''''''''''''''
'
L_barn = 266# / 3.28 'Need exact dimensions here
W_barn = 426# / 3.28
H_barn = 19.5 / 3.28
N_cows = 1000#
R_roof = 1.22  'Without Airflims
R_stillair = 0.12 'This is 1/hcv for the roof interior (ASHRAE 1985)
R_conc = 0.57: R_end = 2.65: R_curt = 0.47: R_fan = 1.21: R_walls = 1.21 'Through calulations in Building Thermal Analysis tab (m2-K/W) with airfilms
```

```
'use TDBEVAP as a good temperature to base each cows heat and moisture input to the barn

If (TDB_center < 10#) Then TDB_for_THPcenter = 10#
If (TDB_center >= 10#) Then TDB_for_THPcenter = TDB_center
THP = (-0.0168 * TDB_for_THPcenter + 2.3666) * ((600 / 500) ^ 0.734) * 500# 'Albright Table
THP = 1.3 * THP
LHP = (0.0263 * TDB_for_THPcenter + 0.4629) * ((600 / 500) ^ 0.734) * 500  'Albright Table
If (LHP < 100#) Then LHP = 100#
LHP = 1.3 * LHP
SHP = THP - LHP 'Published SHP and LHP used for comparing with modelled proportions

SHP_Per_m = SHP * N_cows / L_barn
MP = LHP / 2410000#
MP_Per_m = MP * N_cows / L_barn

'Vent rates for finding velocity for hcv.
''''''''''Ventilation rates determined through mass and energy balance, see Vent section''''''''
'Vent rates finding hcv for 2 m/s (kga/s)
If TDB <= -5 Then Mdot = 0.0274 * TDB ^ 2 + 1.6452 * TDB + 36.011
If -5 < TDB And TDB <= 12 Then Mdot = 0.2855 * TDB ^ 2 + 2.3458 * TDB + 36.268
If 12 < TDB And TDB <= 18 Then Mdot = 12.298 * TDB ^ 2 - 314.36 * TDB + 2125.7
If 18 < TDB And TDB <= 20 Then Mdot = 216.28 * TDB ^ 2 - 7521.8 * TDB + 65790
If TDB > 20 Then Mdot = 1865
'2C Rule (kga/s)
If (VelocityCase$ = "Low" And 12 < TDB And TDB <= 18) Then Mdot = 12.298 * TDB ^ 2 - 314.36 * TDB
+ 2125.7
If (VelocityCase$ = "Low" And 18 < TDB And TDB <= 19) Then Mdot = 7.4218 * TDB + 337.56
If (VelocityCase$ = "Low" And TDB > 19#) Then Mdot = 479
'Yields about 1 m/s average barn airspeed
If (VelocityCase$ = "Medium" And 12 < TDB And TDB <= 18) Then Mdot = 12.298 * TDB ^ 2 - 314.36 *
TDB + 2125.7
If (VelocityCase$ = "Medium" And 18 < TDB And TDB <= 19) Then Mdot = 461.18 * TDB - 7830.2
If (VelocityCase$ = "Medium" And TDB > 19#) Then Mdot = 932#
'Yields about 3 m/s average barn airspeed
If (VelocityCase$ = "SuperHigh" And 12 < TDB And TDB <= 18) Then Mdot = 12.298 * TDB ^ 2 - 314.36
* TDB + 2125.7
If (VelocityCase$ = "SuperHigh" And 18 < TDB And TDB <= 19) Then Mdot = 682.45 * TDB ^ 2 - 24771
* TDB + 225220
If (VelocityCase$ = "SuperHigh" And TDB > 19#) Then Mdot = 2797#

'Constants
alpha = 0.39 'absorptivity of roof for short-wave solar, SUEHRCKE 2008 P.2234 (Table A1)
em_roof = 0.9 'emissivity of roof for long-wave emission back to sky, SUEHRCKE 2008 P.2234 (Table A1)
em_barn = 0.92  'Typical for buildings (dimensionless)
F_cow_to_roof = 0.499 'via Monte Carlo simulation from MS degree code
F_cow_to_everythingelse = 1# - F_cow_to_roof
sigma = 0.00000005669 'Stephen Boltzman's Constant (W/m2-K4)
```

```
hcv_out = -0.1681 * (windspeed) ^ 2 + 4.8556 * windspeed + 8.33 'This is the exterior convective coefficient
required for Solair calculation
Tsa = TDB + (alpha * ghi - 6 * em_roof * Cos(4.76 * 3.14159 / 180) * (10 - Totcld)) / hcv_out

T_at_x = TDBEVAP
W_at_x = WEVAP
For x = 0 To L_barn / 2 Step 0.1
T_at_xplusdx = T_at_x + 0.1 * W_barn * (Tsa - T_at_x) / (Mdot * 1015# * R_roof) + 0.1 * H_barn * 2 *
(TDB - T_at_x) / (Mdot * 1015# * R_walls) + 0.1 * SHP_Per_m / (Mdot * 1015#)
T_at_x = T_at_xplusdx
W_at_xplusdx = W_at_x + 0.1 * MP_Per_m / Mdot
W_at_x = W_at_xplusdx
Next x

W_center = W_at_x
TDB_center = T_at_x
TDB_center_K = TDB_center + 273.15
v_center = ((1 / ATM) * ra * (TDB_center + 273.2) * (1 + 1.6078 * W_center))
Pw_center = ATM * W_center / (0.6221 + W_center)
    If ((TDB_center_K - 273.15) < 0!) Then
     a1 = -5674.5359
     a2 = 6.3925247
     a3 = -0.009677843
     a4 = 0.000000622157
     a5 = 2.0747825E-09
     a6 = -9.484024E-13
     a7 = 4.1635019
    Else
     a1 = -5800.2206
     a2 = 1.3914993
     a3 = -0.04864023
     a4 = 0.000041764768
     a5 = -0.000000014452093
     a6 = 0!
     a7 = 6.5459673
    End If
    b1 = a1 / TDB_center_K + a2 + a3 * TDB_center_K + a4 * TDB_center_K ^ 2 + a5 * TDB_center_K
^ 3 + a6 * TDB_center_K ^ 4 + a7 * Log(TDB_center_K)
Pws_center = Exp(b1)
Ws_center = 0.6221 * Pws_center / (ATM - Pws_center)
RH_center = 100# * Pw_center / Pws_center

    For tw1 = TDB_center_K To TDB_center_K - 30 Step -0.01

    If ((tw1 - 273.2) < 0!) Then
     a1 = -5674.5359
     a2 = 6.3925247
     a3 = -0.009677843
```

```
        a4 = 0.000000622157
        a5 = 2.0747825E-09
        a6 = -9.484024E-13
        a7 = 4.1635019
      Else
        a1 = -5800.2206
        a2 = 1.3914993
        a3 = -0.04864023
        a4 = 0.000041764768
        a5 = -0.000000014452093
        a6 = 0!
        a7 = 6.5459673
      End If
      b1 = a1 / tw1 + a2 + a3 * tw1 + a4 * tw1 ^ 2 + a5 * tw1 ^ 3 + a6 * tw1 ^ 4 + a7 * Log(tw1)
      pwstw = Exp(b1)
      wstw = 0.62198 * pwstw / (p - pwstw)
      h = cpa * TDB_center + W_center * (hfg + cpw * TDB_center)
      hs = cpa * (tw1 - 273.2) + wstw * (hfg + cpw * (tw1 - 273.2))
      hw = cpl * (tw1 - 273.2)
      rs = hs
      ls = h + hw * (wstw - w)
      Delta = rs - ls

      If (Delta <= 0.5) Then
      TWB_center = (tw1 - 273.2)
      GoTo 15
      End If

      Next tw1
15

THI = 0.72 * (TDB_center + TWB_center) + 40.6     'Chapter5: Thermal Indicies, Hahn
THI_tdb = 0.72 * (TDB + twb) + 40.6         'Chapter5: Thermal Indicies, Hahn

''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''
''''''''''General Model Inputs Needed''''''''''''''''''''''''''''''''''''''
''''''''The Modelled Barn''''''''''''''''

'Finding T_roof (Troof, °C) - ALBRIGHT
T_roof = (((Tsa - TDB_center) / ((1 / hcv_out) + R_roof + R_stillair)) * R_stillair) + TDB_center '°C
T_roof_K = T_roof + 273.15                                                      'K
'Finiding T_walls (Twalls, °C) - ALBRIGHT, HEAT TRANSFER
T_conc = TDB_center + ((R_stillair * (TDB - TDB_center)) / R_conc)                '°C
T_end = TDB_center + ((R_stillair * (TDB - TDB_center)) / R_end)                  '°C
T_curt = TDB_center + ((R_stillair * (TDB - TDB_center)) / R_curt)                '°C
T_fan = TDB_center + ((R_stillair * (TDB - TDB_center)) / R_fan)                  '°C
T_walls_total = 0.191 * T_conc + 0.275 * T_end + 0.267 * T_curt + 0.267 * T_fan   '°C
If TDB = TDB_center Then T_walls = TDB_center
```

```
If TDB <> TDB_center Then T_walls = T_walls_total
T_walls_K = T_walls + 273.15                                        'K

Abarn = (W_barn * H_barn) 'Cross-section of barn, 19.54 is mid height of barn (m2)
'Finding TMRT(°C)
TMRT = ((((T_roof_K) ^ 4) * F_cow_to_roof + ((T_walls_K) ^ 4) * F_cow_to_everythingelse) ^ 0.25) - 273.15 'Shape factor calculated from Dr. Hoff's Thesis
TMRT_K = TMRT + 273.15

'Cow Thermal Exchange Model Constants
em_cow = 0.98           'THOMPSON 2014, (DA SILVA 2000) (dimensionless)
kg_cow = 600#           'Weight used in McGovern 2000, Berman 2003, Berman 2005 (kg)
Ab = 0.14 * kg_cow ^ 0.57    'THOMPSON 2014 Eqn 3.18, (BRODY 1945), Berman's 2003 study confirms (m2)
D_cow = 0.06 * kg_cow ^ 0.39 'MCGOVERN, Bruce 2000, (Ehrlemark 1988) (m)
L_cow = (Ab - D_cow ^ 2 * 3.14159 / 2) / (D_cow * 3.14159)  'MCGOVERN, Bruce 2000 (m)
Coat_length = 0.01 'Need for the new routine on vapor transfer through coat MCGOVERN (2000), m
Rp = 0.086 'Thermal resistance through coat, MCGOVERN (2000)

'Tissue Resistance (m2-K/W)
Rt_min = 0.0156            'MCGOVERN and BRUCE p. 84, MCARTHUR 1981 (m2-K/W)
Rt_max = (2.4 + 0.0054 * kg_cow) / 48 'Berman 2004 (m2-K/W)

'Sweating Rates (g/m2-h)
SR_min = 14.4   'MCARTHUR 1987 p. 224, KIBLER & BRODY 1950 (g/m2-h)
SR_max = 288 'MCARTHUR 1987 p. 224, HALES 1974 (g/m2-h)
'SR_max = 220 'BEMAN 2005 p. 1378 for hot desert environment

'Respiration Rate (BPM - breaths per minute)
RR_min = 12#  'THOMPSON 2014 (BPM)
RR_max = 120# 'More realistic for model (Max in Kifle's study 2010) @35 °C

Tidal_max = 0.4 + 0.0064 * kg_cow 'BERMAN 2005 p. 1378 (L/breath)
Tidal_min = 1.9 'BERMAN 2005 p. 1378 (L/breath)

''''''Various Considerations for the THP we are Modelling to Assess the Environment’s Ability to Dissipate''''''
THP_Optimal = 1.3 * ((-0.0168 * 15# + 2.3666) * ((600 / 500) ^ 0.734) * 500#) 'Eventually this will be based on milk production we want to model

THP = (-0.0168 * TDB_center + 2.3666) * ((600 / 500) ^ 0.734) * 500# 'Albright Table
THP = 1.3 * THP
LHP = (0.0263 * TDB_center + 0.4629) * ((600 / 500) ^ 0.734) * 500   'Albright Table
LHP = 1.3 * LHP
SHP = THP - LHP 'Published SHP and LHP used for comparing with modelled proportions

LHP_ratio = LHP / THP
SHP_ratio = SHP / THP
```

```
THP = THP_Optimal     'Turning on or off the comparison with some baseline value
LHP = LHP_ratio * THP 'Keep the ratio the same as published data for comparison
SHP = SHP_ratio * THP

''''''''''''''''''''''''''''''''''''''''''''''''''''''''''
''''''''''Determine Convective hcv''''''''''''''''''''''''
''''''''''''''''''''''''''''''''''''''''''''''''''''''''''

Vdot = Mdot * vEVAP     '(m3/s)
Velocity = Vdot / Abarn '(m/s)

''''''''''All calculated from correlations in excel, HEAT TRANSFER BOOK''''''''''''''''''''''''''''

Tfilm = (TDB_center + Tskin) / 2 'The Tskin used is from the previous hour modelled 'Film temperature (°C)
'Air density (kg/m3)
air_density = 0.000012197 * TDB_center ^ 2 - 0.0050341 * TDB_center + 1.2963
'Kinematic Viscosity (m2/s)
kin_viscosity = 0.000000000105786 * TDB_center ^ 2 + 0.0000000884243 * TDB_center + 0.0000134661
'Prandtl number (dimensionless)
Pr = 0.00000055 * Tfilm ^ 2 - 0.00025818 * Tfilm + 0.71456
'Air conductivity (W/m-K)
kf = -0.000000030714 * Tfilm ^ 2 + 0.000079871 * Tfilm + 0.024132
'Specific heat (KJ/kg-K)
cp_KJ = 0.0000004 * TDB_center ^ 2 + 0.000013377 * TDB_center + 1.006
cp_air = cp_KJ * 1000  'J/kg-K
thermal_cond = kf / (air_density * cp_air)  'm2/s
grav = 9.81 'gravity (m/s2)

''''''Determining Natural and Forced Convection, h - HEAT TRANSFER BOOK''''''''''''
Re_D = (Velocity * D_cow) / kin_viscosity 'dimensionless
Re_L = (Velocity * L_cow) / kin_viscosity 'dimensionless
Nu_D = 0.0266 * (Re_D ^ 0.805) * (Pr ^ (1 / 3)) 'For cylinder (dimensionless)
Nu_L = 0.664 * (Re_L ^ 0.5) * (Pr ^ (1 / 3))    'For flat plate (dimensionless)

hcv = (Nu_D * kf) / D_cow        '(W/m2-K) cylinder in cross-flow case
'hcv = (Nu_L * kf) / L_cow        '(W/m2-K) flat plate case

'For natural convection calculations....This still applies for all cases since still air conditions and diameter is still the characteristic length
Beta = 1 / (Tfilm + 273.15) '1/K
Gr = (grav * Beta * Abs(Tskin - TDB_center) * (D_cow) ^ 3) / (kin_viscosity ^ 2)  'dimensionless
Ra_d = (grav * Beta * Abs(Tskin - TDB_center) * (D_cow) ^ 3) / (kin_viscosity * thermal_cond) 'dimensionless
NU_natural = (0.6 + ((0.387 * Ra_d ^ (1 / 6)) / ((1 + (0.559 / Pr) ^ (9 / 16)) ^ (8 / 27)))) ^ 2   'dimensionless
h_natural = NU_natural * kf / D_cow   '(W/m2-K)

If h_natural > hcv Then h_overall = h_natural
```

```
If h_natural <= hcv Then h_overall = hcv

Rbl = 1 / h_overall '(m2-K/W)

''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''
'''''''''Now the Main Model is Below''''''''''''''''''''''''
''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''
RR = 6.25 * TDB_center - 80.47 'THOMPSON 2011, EQN. 12 (BPM) - The entire process starts with an estimated RR based on dry-bulb temperature surrounding cow. This is raised if required for a balance.
If (RR < RR_min) Then RR = RR_min
If (RR > RR_max) Then RR = RR_max

20

Tidal = -0.0000944 * RR ^ 2 - 0.000182 * RR + 4.3187 'Correlation from THOMPSON EQN 11 & 12 (L/breath)
'Tidal = -0.1 * TDB_center + 6.22 'THOMPSON 2011, EQN. 11 (L/breath)
If (Tidal > Tidal_max) Then Tidal = Tidal_max
Rt_actual = 1.6212 * RR ^ (-1.029) 'Original published relation Thompson 2011. Seems most reasonable.
'Rt_actual = (4.3 - 0.1 * TDB_center) / 48  'BERMAN 2004, corrected for units (m2-°C/W)
If (Rt_actual < Rt_min) Then Rt_actual = Rt_min
If (Rt_actual > Rt_max) Then Rt_actual = Rt_max

SR_published = 0.75 * Exp(0.15 * Tskin)    'THOMPSON and FADEL 2011 (g/m2-h)....Note I changed the variable name 02/26/2017
If (SR_published < SR_min) Then SR_published = SR_min
If (SR_published > SR_max) Then SR_published = SR_max


''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''
'Respiration in current conditions. Handled as a straight heating and humidifying process with virtual body and air temperatures as per published procedures.
''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''
Pws_body = Exp(77.345 + 0.0057 * (Tcore + 273.15) - 7235 / (Tcore + 273.15)) / (Tcore + 273.15) ^ 8.2 'ALBRIGHT (Pa)
Pws_air = Exp(77.345 + 0.0057 * (TDB_center + 273.15) - 7235 / (TDB_center + 273.15)) / (TDB_center + 273.15) ^ 8.2      'ALBRIGHT (Pa)
Tvb = (Tcore + 273.15) * (1 + 0.38 * Pws_body / ATM)      'MCARTHUR 1987 p. 217, Virtual body and air temperatures (K)
Tva = (TDB_center + 273.15) * (1 + 0.38 * Pws_air / ATM)    'MCARTHUR 1987 p. 217 (K)
cp_mix = 1006# + W_center * 1850#  '(J/kgK)
Q_sensible_breathing = (1000 / 100 ^ 3) * (RR / 60) * Tidal * (1 / v_center) * cp_mix * (Tvb - Tva) 'THOMPSON 2011, EQN 9 (W)
Pw_lungs = 0.85 * Pws_body 'Not assuming full saturation here. (Pa)
W_lungs = 0.6221 * (Pw_lungs / (ATM - Pw_lungs))    'ALBRIGHT (kgw/kga)
Q_latent_breathing = (1000 / 100 ^3) * (RR / 60) * Tidal * (1 / v_center) * 2410000# * (W_lungs - W_center) 'THOMPSON 2011, EQN 9 modified (W)
Qdot_respiration = Q_sensible_breathing + Q_latent_breathing 'Watts
```

```
''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''
'Energy for heating feed/water intake to core body temp.
''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''
Fi = -0.0003 * TDB_center ^ 3 + 0.0056 * TDB_center ^ 2 - 0.0103 * TDB_center + 22.919 'Correlation in excel from NRC 1989 (kg/day)
If Fi < 0.02 * kg_cow Then Fi = 0.02 * kg_cow   'Ensures she is always consuming at least 50% feed intake
Win = 0.0347 * TDB_center ^ 2 + 2.1338 * TDB_center + 78.37 'Correlation in excel from NRC 1989, (kg/day)
cpf = 3.25     'kJ/kg-°C Estimated from Thermal properties of haylage (600ENSD)
Tfd = TDB   'Assume temp of food = outdoor temp
Tw = (TDB_center + Tground) / 2
THP_Based_on_Diet = Fi * 1648# * 4184# / (24# * 3600#) ' assumes net ME of feed lost to heat is 1648 kcal/kg

Qdot_feedandwater = (Fi * cpf * (Tcore - Tfd) + Win * cpl * (Tcore - Tw)) / 86.4  'Internal Energy Calc. (86.4 is from 24 hours * 3600 seconds to KJ)
Qdot_heat_to_skin = THP - Qdot_respiration - Qdot_feedandwater   'Watts

''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''
''''''''At this point the energy that is left can be dissipated by convection, radiation, and evaporation of sweat
''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''
'Skin in current conditions
Tskin = Tcore - (Rt_actual / Ab) * Qdot_heat_to_skin  '°C
Rt_required_for_balance = (Tcore - Tskin) * Ab / Qdot_heat_to_skin 'Diagnostic stuff
Taverage_cv = 1# * Tskin + 0# * TDB_center     'Allows for averaging if required.
Taverage_rad = 0.8 * Tskin + 0.2 * TDB_center
Qcv = Ab * (Taverage_cv - TDB_center) / (Rbl + Rp)
Qrad = Ab * sigma * em_cow * ((Taverage_rad + 273.15) ^ 4 - (TMRT_K) ^ 4)
‘'
''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''
''''''At this point we need to determine if the environment is capable of evaporating enough sweat to balance energy''''''''''''''
''''''Updated Routine for Vapor Mass Transfer Through Coat and Outside Coat Boundary Layer....Directly from McGovern and Bruce''''''
''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''
'Evaporation potential in current conditions
Pws_skin = Exp(77.345 + 0.0057 * (Tskin + 273.15) - 7235 / (Tskin + 273.15)) / (Tskin + 273.15) ^ 8.2 'Pa
pw_skin = 0.95 * Pws_skin 'Not quite full vapor saturation at the skin (Pa)
W_skin = 0.6221 * (pw_skin / (ATM - pw_skin)) 'ALBRIGHT (kgw/kga)
Max_wettedness = 0.5534 'Based on SR_max and RR_max and 1.9W/kg*(1.3HF) maximum possible latent heat loss.
'actual % body area that is sweating at any given time, back calculated
P_mbar = ATM / 100#
Pw_mbar = Pw_center / 100#
Pa_mbar = (ATM - Pw_center) / 100#
Pw_skin_mbar = pw_skin / 100#
Pw_coat_mbar = 0.5 * (Pw_mbar + Pw_skin_mbar)
```

```
Tcoat = 0.5 * (Tskin + TDB_center)
Tcoat_K = Tcoat + 273.15
Tskin_K = Tskin + 273.15
D_vapor_in_air = 0.0000285 'm2/s
Delta_Coat_Length = 0.0025 * Velocity

Gr_coat = (grav * D_cow ^ 3 * P_mbar * Abs(Tcoat - TDB_center) + 0.61 * Abs(Pw_coat_mbar * Tcoat_K - Pw_mbar * TDB_center_K) / (273# * Pa_mbar * (kin_viscosity * air_density) ^ 2))
Nu_coat_natural = 0.48 * Gr_coat ^ 0.25      'Valid for all cases using D_cow as characteristic dimension
Sc_coat = kin_viscosity / D_vapor_in_air
Le_coat = Sc_coat / Pr
Sh_natural = Nu_coat_natural * Le_coat ^ 0.25
hvc = D_vapor_in_air / (Coat_length - Delta_Coat_Length)       'Vapor DIFFUSION through the coat, m/s
hvcbl = Sh_natural * D_vapor_in_air / D_cow                    'Vapor NATURAL CONVECTION through the coat, m/s
hvbl = h_overall * v_center / (cp_air * Le_coat ^ (2 / 3))     'Vapor transport through outer boundary layer
                                                               'Based purely on the analogy between heat and mass transfer, m/s.
'Holman p626, eq 11-31
Rvc = 1 / hvc      'Next four s/m
Rvcbl = 1 / hvcbl
Rvbl = 1 / hvbl
Rv_equivalent = Rvbl + (Rvc * Rvcbl) / (Rvc + Rvcbl) 'Through the coat, vapor is transferred in parallel. Then this is in series with the outer boundary layer.
hv_equivalent = 1 / Rv_equivalent
Q_latent_evaporation_potential = Max_wettedness * Ab * (2410000) * (W_skin - W_center) / (Rv_equivalent * v_center) 'This now replaces everything else we were using for vapor evaporation from skin/coat
If (Q_latent_evaporation_potential < 0#) Then Q_latent_evaporation_potential = 0#
Mdot_potential = (W_skin - W_center) * hv_equivalent * 3600# * 1000# / v_center
If (Mdot_potential < 0#) Then Mdot_potential = 0#
'
''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''
''''''''''''Quantify the Amount Released by Sweating if Possible''''''''
''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''
Qdot_remaining_for_sweating = Qdot_heat_to_skin - (Qcv + Qrad)  'Watts
Area_body_sweating = Qdot_remaining_for_sweating * ((3600# * 1000#) / (2410000# * SR_max)) 'm2
Percent_body_sweating = (Area_body_sweating / Ab) * 100

If (Percent_body_sweating > 90#) Then Percent_body_sweating = 90    'can't have 100% body sweating
Qdot_sweat_maxpossible = SR_max * Max_wettedness * Ab * 2410000# / (3600# * 1000#)

If (Qdot_remaining_for_sweating > Q_latent_evaporation_potential) Then Qdot_sweating_actual = Q_latent_evaporation_potential

If (Qdot_remaining_for_sweating <= Q_latent_evaporation_potential) Then Qdot_sweating_actual = Qdot_remaining_for_sweating
```

```
If (Qdot_sweating_actual > Qdot_sweat_maxpossible) Then Qdot_sweat_actual =
Qdot_sweat_maxpossible

Mdot_sweating_rate_required = Qdot_remaining_for_sweating * 1000# * 3600# / (2410000# * Ab)
Mdot_sweating_rate_actually_evaporated = Qdot_sweating_actual * 1000# * 3600# / (2410000# * Ab)

Excess_Evaporation_Potential = Q_latent_evaporation_potential - Qdot_remaining_for_sweating
Qdot_storage = Qdot_remaining_for_sweating - Qdot_sweating_actual
Qdot_storage_for_this_hour = Qdot_storage
''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''
'''''''If sweating cannot dissipate enough heat, then allow RR to increase to help the process
''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''
If (Qdot_storage > 0 And RR < RR_max) Then
RR = RR + 2
If (RR > RR_max) Then RR = RR_max
GoTo 20
End If

'''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''
'If Tcore is elevated but now we can dissipate extra stored energy, then do it
'''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''
If (Tcore > 38.5 And Excess_Evaporation_Potential > 0#) Then
Tcore = Tcore - Excess_Evaporation_Potential * 3600# / (kg_cow * 3400#)
If (Tcore < 38.5) Then Tcore = 38.5
End If

''''''''''''''''''''''''''''''''''''''''''''''''''''''
'If excess energy needs to be stored, then Tcore must rise
''''''''''''''''''''''''''''''''''''''''''''''''''''''
If (Qdot_storage > 0) Then
Tcore = Tcore + 0.25 * Qdot_storage * 3600# / (kg_cow * 3400#)
End If

Total_Modelled_Heat = Qdot_respiration + Qdot_feedandwater + Qdot_sweating_actual + Qcv + Qrad +
Qdot_storage
Heat_Difference = Abs(THP - Total_Modelled_Heat)
If (Heat_Difference < 1#) Then Heat_Difference = 0#

'''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''
'Now, if we can in fact take advantage of sprinkling, let's see how much extra we can actually achieve through
sprinkling AND what potential
'reduction in RR could be achieved with extra loss through sprinkling
'''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''
SprinklingPotential = 0# 'Needed to add this in based on a detailed line-by-line analysis
If (Q_latent_evaporation_potential > Qdot_remaining_for_sweating) Then
SprinklingPotential = Q_latent_evaporation_potential - Qdot_remaining_for_sweating
End If
```

```
If (SprinklingPotential > 50 And TDB > TDB_Evap_ON And EVAPeff = 0#) Then
AvailableSprinklingHours = AvailableSprinklingHours + 1
Sprinkler_Gallons = ((SprinklingPotential / 2410000#) / 997#) * 3.28 ^ 3 * 7.5 * 3600# 'gallons per hour per cow if the need is 100W or more
Sprinkler_Gallons_TotalPerHead = Sprinkler_Gallons_TotalPerHead + Sprinkler_Gallons
End If

RR_new = RR

''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''
'This routine does the following: If sprinkling can remove enough excess energy, then it is possible that the RR can be reduced
''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''
If (SprinklingPotential > 50 And TDB > TDB_Evap_ON And EVAPeff = 0#) Then
   For RR_reduction = 1 To 100
      Q_sensible_breathing_reduction = (1000 / 100 ^ 3) * (RR_reduction / 60#) * Tidal * (1 / v_center) * cp_mix * (Tvb - Tva)
      Q_latent_breathing_reduction = (1000 / 100 ^ 3) * (RR_reduction / 60#) * Tidal * (1 / v_center) * 2410000# * (W_lungs - W_center)
         If (Abs(SprinklingPotential - (Q_sensible_breathing_reduction + Q_latent_breathing_reduction)) < 50#) Then
            RR_new = RR - RR_reduction
            GoTo 30
         End If
   Next RR_reduction
End If
30
If (RR_new < 20#) Then RR_new = 20# 'We might have lots of excess evaporation potential but in the end she still needs to breath. Use resting RR from Merck vet manual
'http://www.merckvetmanual.com/appendixes/reference-guides/resting-respiratory-rates

''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''
Mdot_Water_EvapCooler = Mdot * Delta_W
EvapCooler_Gallons_Total = Mdot_Water_EvapCooler / 997# * 3.28 ^ 3 * 7.5 * 3600# 'Water density of 997 kg/m3

If (EvapCooler_Gallons_Total < 50# Or TDB < TDB_Evap_ON) Then EvapCooler_Gallons_Total = 0# 'Not worth operating evaporative cooler if the need is small or she is not stressed!
If (EvapCooler_Gallons_Total >= 50# And TDB >= TDB_Evap_ON) Then EvapCooler_Hours = EvapCooler_Hours + 1
EvapCooler_Gallons_PerHead = EvapCooler_Gallons_Total / N_cows
EvapCooler_Gallons_TotalPerHead = EvapCooler_Gallons_TotalPerHead +
EvapCooler_Gallons_PerHead

SHP_Model = Qcv + Qrad + Qdot_feedandwater + Qdot_storage
LHP_Model = THP - SHP_Model

''''''''''''''''''''''''''''''''''''''''''''''''''''
```

```
'''''''Heat Stress Indicators'''''''''''''''''''
''''''''''''''''''''''''''''''''''''''''''''''''''''
If (TDB_center >= 25#) Then TDBsum = TDBsum + 1
If (TDB_center >= 25# And W_center >= 0.01) Then WTDBsum = WTDBsum + 1

If (THI >= 68 And THI < 72) Then THI_1 = THI_1 + 1
If (THI >= 72 And THI < 79) Then THI_2 = THI_2 + 1
If (THI >= 79 And THI < 89) Then THI_3 = THI_3 + 1
If (THI >= 89) Then THI_4 = THI_4 + 1

If (THI > THI_max) Then THI_max = THI

Velocity_Correction = Velocity
If (Velocity < 0.5) Then Velocity_Correction = 0.5
THI_Airspeed_Correction = 5.2225 * Log(Velocity_Correction) + 3.3039
THI_Adjusted = THI - THI_Airspeed_Correction
''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''

'''''''CCI''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''
RH_correction = (Exp(0.00182 * RH_center + 0.000018 * TDB_center * RH_center)) * (0.000054 * TDB_center ^ 2 + 0.00192 * TDB_center - 0.0246) * (RH_center - 30)
Log_Factor = 2.26 * Velocity + 0.33
Base = 0.3
Log_Part = Log(Log_Factor) / Log(Base)
WS_correction_A = 0.45 * (2.9 + 0.00000114 * Velocity ^ 2.5 - Log_Part ^ -2)
If (WS_correction_A > 500#) Then WS_correction_A = 500#
If (WS_correction_A < -500#) Then WS_correction_A = -500#
WS_correction_B = (2.26 * Velocity + 0.23) ^ WS_correction_A
WS_correction_C = 1# / WS_correction_B
WS_correction = -6.56 / (Exp(WS_correction_C)) - 0.00566 * Velocity ^ 2 + 3.33
RAD_correction = 0.0076 * ghi - 0.00002 * ghi * TDB_center + 0.00005 * TDB_center ^ 2 * (ghi + 1) ^ 0.5 + 0.1 * TDB_center - 2#
CCI_withRAD = TDB_center + RH_correction + WS_correction + RAD_correction 'Straight inclusion based on CCI even though the cow is not directly exposed to solar
CCI_noRAD = TDB_center + RH_correction + WS_correction
CCI_withMRT = TDB_center + RH_correction + WS_correction + (TMRT - TDB_center) 'Just a quick proposal related to MRT

If (CCI_noRAD >= 25 And CCI_noRAD < 30) Then CCI_1 = CCI_1 + 1
If (CCI_noRAD >= 30 And CCI_noRAD < 35) Then CCI_2 = CCI_2 + 1
If (CCI_noRAD >= 35 And CCI_noRAD < 40) Then CCI_3 = CCI_3 + 1
If (CCI_noRAD >= 40) Then CCI_4 = CCI_4 + 1

If (CCI_noRAD > CCI_max) Then CCI_max = CCI_noRAD
''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''
'''''RR and RR_new Summary''''''''''''''''''''''''''''''''''''
If (RR >= 60 And RR < 75) Then RR_1 = RR_1 + 1
If (RR >= 75 And RR < 85) Then RR_2 = RR_2 + 1
```

```
If (RR >= 85 And RR < 110) Then RR_3 = RR_3 + 1
If (RR >= 110) Then RR_4 = RR_4 + 1

If (RR_new >= 60 And RR_new < 75) Then RRnew_1 = RRnew_1 + 1
If (RR_new >= 75 And RR_new < 85) Then RRnew_2 = RRnew_2 + 1
If (RR_new >= 85 And RR_new < 110) Then RRnew_3 = RRnew_3 + 1
If (RR_new >= 110) Then RRnew_4 = RRnew_4 + 1
''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''

''''CDH Summary Data''''''''''''''''''''''''''''''''''''''''
CDH = 1 - Qdot_storage / THP 'If she needs to store energy than the environment is incapable of dissipating the required heat and therefore CDH goes below 1.00.
If (CDH < 1#) Then CDH_1 = CDH_1 + 1
If (CDH < 0.9) Then CDH_2 = CDH_2 + 1
If (CDH < 0.8) Then CDH_3 = CDH_3 + 1
If (CDH < 0.7) Then CDH_4 = CDH_4 + 1
''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''

''''''Now Consecutive Hours at Various Heat Stress Levels is Handled Here''''''
If (THI > 68 And THI_PreviousHour > 68) Then
a = a + 1: b = b + 1: c = c + 1: d = d + 1
   If (a = 8) Then
   THI_Events8 = THI_Events8 + 1
   a = 0
   End If
   If (b = 10) Then
   THI_Events10 = THI_Events10 + 1
   b = 0
   End If
   If (c = 12) Then
   THI_Events12 = THI_Events12 + 1
   c = 0
   End If
   If (d = 14) Then
   THI_Events14 = THI_Events14 + 1
   d = 0
   End If
Else
a = 0: b = 0: c = 0: d = 0
End If
THI_PreviousHour = THI

If (CCI_noRAD > 25 And CCInoRAD_PreviousHour > 25) Then
e = e + 1: f = f + 1: g = g + 1: h = h + 1
   If (e = 8) Then
   CCI_Events8 = CCI_Events8 + 1
   e = 0
   End If
```

```
    If (f = 10) Then
    CCI_Events10 = CCI_Events10 + 1
    f = 0
    End If
    If (g = 12) Then
    CCI_Events12 = CCI_Events12 + 1
    g = 0
    End If
    If (h = 14) Then
    CCI_Events14 = CCI_Events14 + 1
    h = 0
    End If
Else
e = 0: f = 0: g = 0: h = 0
End If
CCInoRAD_PreviousHour = CCI_noRAD

'''''Additional heat stress indicators''''''''''
HT_RR = (Tcore / 38.33) + (RR / 23)
If (HT_RR > 4#) Then HTRR = HTRR + 1
HT_RRnew = (Tcore / 38.33) + (RR_new / 23)
If (HT_RRnew > 4#) Then HTRRnew = HTRRnew + 1

EIT = 27.88 - 0.456 * TDB_center + 0.010754 * TDB_center ^ 2 - 0.4905 * RH_center + 0.00088 *
RH_center ^ 2 + 1.15 * Velocity - 0.12644 * Velocity ^ 2 + 0.019876 * TDB_center * RH_center - 0.046313
* TDB_center * Velocity
If (EIT > 25#) Then EITSum = EITSum + 1
''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''
'''''''This is the end of calculations for this hour''''''''''''''''''''''''''''
''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''''
Win_TotalPerHead = Win_TotalPerHead + Win * 2.2 * 7.5 / (24# * 64#) 'Win in kg/day
FeedSum = FeedSum + Fi / 24# 'Fi in kg/day
FanEnergy = FanEnergy + (Vdot * 60# * 3.28 ^ 3 / 18#) / 1000# 'kW of fan power assuming 18 cfm/watt
average fan efficiency
Fi_CCI = -0.0002 * CCI_noRAD ^ 3 + 0.0069 * CCI_noRAD ^ 2 - 0.0487 * CCI_noRAD + 22.994
FeedSum_new = FeedSum_new + Fi_CCI / 24# 'Fi in kg/day

Open OutputData$ For Append As #2
Write #2, SN, Location$, State$, Latitude, Longitude, ad$(i - 2), at$(i - 2), JulianDay, Vdot, Velocity, w,
WEVAP, W_center, TDB, TDBEVAP, TDB_center, Tsa, T_roof, TMRT, Tcore, Tskin, Rt_actual, h_overall,
Fi, Fi_CCI, Win, THP_Based_on_Diet, THP, LHP, SHP, LHP_Model, SHP_Model, Qdot_storage,
Q_sensible_breathing, Q_latent_breathing, Qdot_respiration, Qdot_feedandwater, Qdot_heat_to_skin,
Qcv, Qrad, Qdot_remaining_for_sweating, Q_latent_evaporation_potential, SR_published,
Mdot_sweating_rate_required, Mdot_sweating_rate_actually_evaporated, Qdot_sweating_actual,
Heat_Difference, Excess_Evaporation_Potential, SprinklingPotential, AvailableSprinklingHours,
Sprinkler_Gallons_TotalPerHead, EvapUseFlag, EvapCooler_Gallons_TotalPerHead, FanEnergy, Tidal,
RR, RR_new, EIT, HT_RR, HT_RRnew, THI_1, THI_2, THI_3, THI_4, THI, THI_Adjusted, CCI_noRAD,
```

```
CCI_withRAD, CCI_withMRT, CDH, CDH_1, CDH_2, CDH_3, CDH_4, THI_Events8, THI_Events10,
THI_Events12, THI_Events14, CCI_Events8, CCI_Events10, CCI_Events12, CCI_Events14
Close #2

Next i

Total_WaterUse = Sprinkler_Gallons_TotalPerHead + EvapCooler_Gallons_TotalPerHead +
Win_TotalPerHead

'Now send summary data to the contour file for plotting later''''''''''''''''
Open ContourData$ For Append As #3
Write #3, SN, Location$, State$, Latitude, Longitude, EVAPeff, VelocityCase$, AvailableSprinklingHours,
Sprinkler_Gallons_TotalPerHead, EvapCooler_Hours, EvapCooler_Gallons_TotalPerHead, EITSum,
HTRR, HTRRnew, THI_1, THI_2, THI_3, THI_4, THI_max, CCI_1, CCI_2, CCI_3, CCI_4, CCI_max,
CDH_1, CDH_2, CDH_3, CDH_4, RR_1, RR_2, RR_3, RR_4, RRnew_1, RRnew_2, RRnew_3, RRnew_4,
THI_Events8, THI_Events10, THI_Events12, THI_Events14, CCI_Events8, CCI_Events10, CCI_Events12,
CCI_Events14, Win_TotalPerHead, Total_WaterUse, FeedSum, FeedSum_new, TDBsum, WTDBsum,
FanEnergy
Close #3
Close #1

Loop
Close #10
End Sub
```

APPENDIX G. FEED AND WATER INTAKE CORRELATIONS

Feed and water intake rates were more difficult to estimate since both are very dependent on temperature and milk production, but milk production is also dependent on temperature and feed and water intake. The National Research Council published several tables accounting for feed and water intake adjustments that are dependent on temperature. An assumption was made that the modeled cow produces 35 kg of milk per day in order to determine the standard dry matter intake so that adjustments could be made. At this milk production rate, Table G.1 shows that this modeled cow, which is 600 kg, has a DMI of 22.2 kg day^{-1}. Then, Tables G.2 and G.3 were used to adjust the feed and water intake accordingly. Table G.4 shows the results after adjustments were made.

Table G. 1. Dry matter intake in dairy cattle of different weights and milk production rates (NRC, 1989)

Milk Production	**Dry matter intake[2] (% of body weight):**				
4% FCM[1] kg cow^{-1} day^{-1}	400 kg cow	500 kg cow	600 kg cow	700 kg cow	800 kg cow
10	2.7 (10.8)	2.4 (12.0)	2.2 (13.2)	2.0 (14.0)	1.9 (15.2)
15	3.2 (12.8)	3.0 (15.0)	2.6 (15.6)	2.3 (16.1)	2.2 (17.6)
20	3.6 (14.4)	3.2 (16.0)	2.9 (17.4)	2.6 (18.2)	2.4 (19.2)
25	4.0 (16.0)	3.5 (17.5)	3.2 (19.2)	2. (20.3)	2.7 (21.6)
30	4.4 (17.6)	3.9 (19.5)	3.5 (21.0)	3.2 (22.4)	2.9 (23.2)
35	5.0 (20.0)	4.2 (21.0)	3.7 (22.2)	3.4 (23.8)	3.1 (24.8)
40	5.5 (22.0)	4.6 (23.0)	4.0 (24.0)	3.6 (25.2)	3.3 (26.4)
45	–	5.0 (25.0)	4.3 (25.8)	3.8 (26.5)	3.5 (28.0)
50	–	5.4 (27.0)	4.7 (28.2)	4.1 (28.7)	3.7 (29.6)
55	–	–	5.0 (30.0)	4.4 (30.8)	4.0 (32.0)
60	–	–	5.4 (32.4)	4.8 (33.6)	4.3 (34.4)

[1]FCM = fat-corrected milk
[2]kg cow^{-1} day^{-1}

Table G. 2. Feed intake adjustments for different environmental temperatures (NRC, 1987)

Temperature (°C)	Intake Adjustment (%)
> 35, no night cooling	- 35
> 35, with night cooling	- 10
25 to 35	-10
15 to 25	None
5 to 15	3
-5 to 5	5
-15 to -5	7
< -15	16

Table G. 3. Relationship between environmental temperature and water requirements of livestock (NRC, 1981)

Environmental temperature (°C)	Water requirements (kg per kg dry matter consumed)
> 35	8 to 15
25 to 35	4 to 10
15 to 25	3 to 5
-5 to 15	2 to 4
< -5	2 to 3[1]

[1]Increases of 50 to 100% occur when there is a rise in environmental temperature following a period of very cold temperature, e.g. a rise from -20 to 0 °C

Table G. 4. Final DMI and water intake rates after adjustment

Temperature (°C)	Intake Adjustment (%)	DMI (kg day^{-1})	Water Intake (kg day^{-1})
-16	16	25.8	51.5
-15	12	24.8	50.2
-6	7	23.8	69.9
-5	5	23.3	70.6
4	5	23.3	93.2
5	4	23.1	94.7
14	3	22.9	114.3
15	0	22.2	117.7
24	0	22.2	142.1
25	-5	21.1	143.4
34	-10	20.0	189.8
35	-10	20.0	199.8

Table G. 4 (cont). Final DMI and water intake rates after adjustment

36	-23	17.2	204.2
37	-35	14.4	209.8
40	-40	13.3	216.5

Using Table G.4, third and second order polynomial correlations were made to create feed and water intake equations (Figures G.1 and G.2).

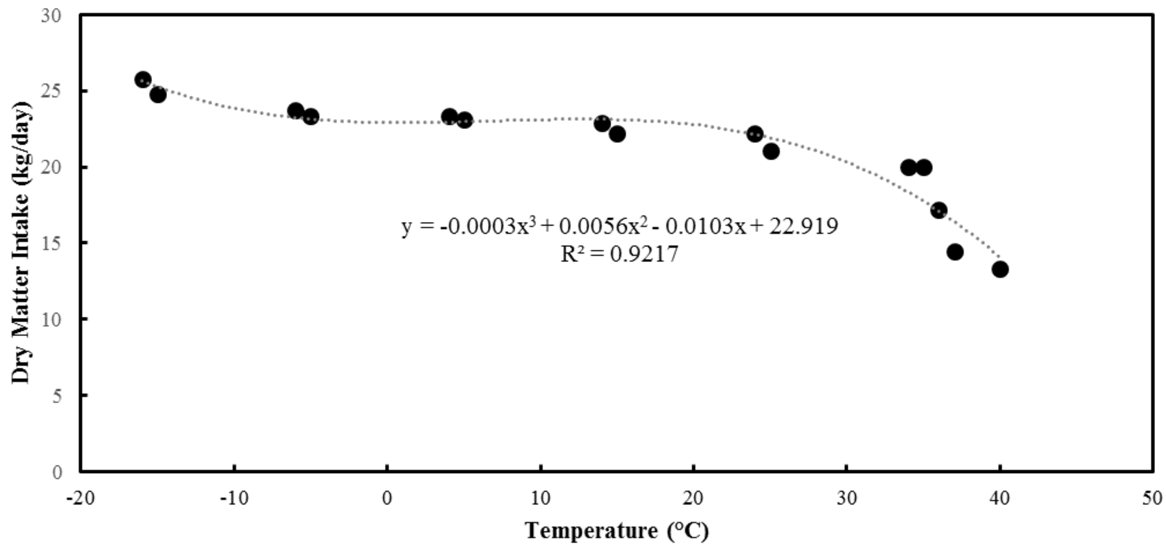

Figure G. 1. Correlation used to determine equation for DMI

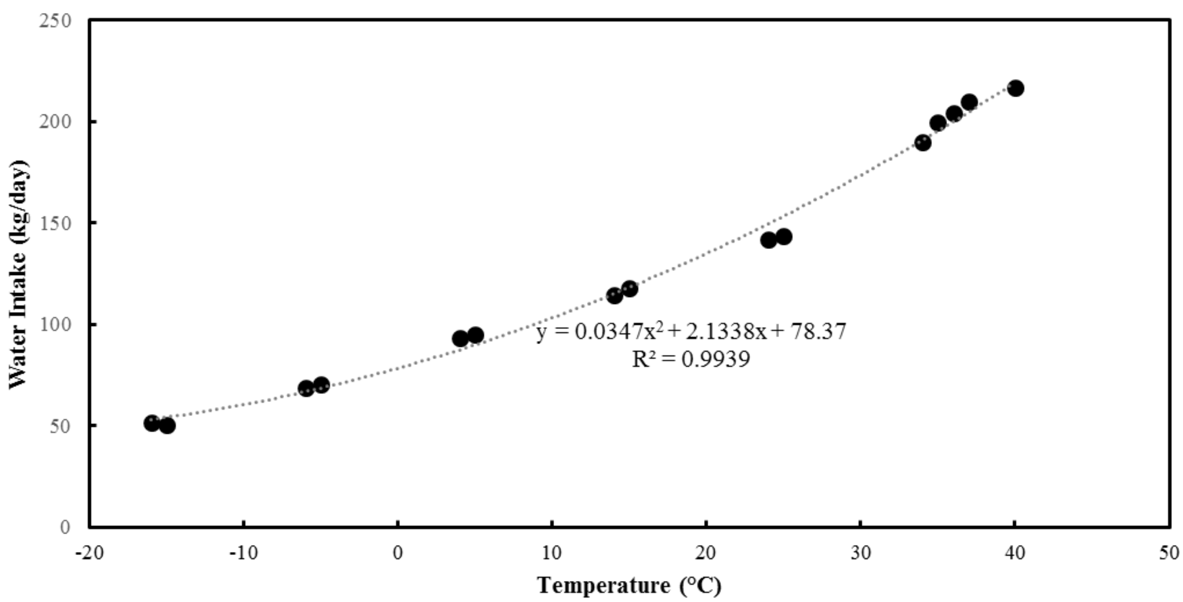

Figure G. 3. Correlation used to determine equation for water intake

APPENDIX H. NATURAL AND FORCED CONVECTIVE HEAT TRANSFER EQUATONS

The governing convective heat transfer coefficient is the larger of the natural or forced convective heat transfer coefficients. In order to determine these coefficients, the air properties need to be known. Table H.1 outlines different air properties at varied temperatures, which include kinematic viscosity (ν), Prandtl's number, air density (ρ), air conductivity (k_f), and specific heat of air (c_p). Correlations (Equations H.2 – H.6) were made to determine the air properties at all temperatures. The Prandtl number and air conductivity were evaluated at the film temperature (Equation H.1):

Table H. 1. Air properties (Bergman, Lavine, Incropera, & Dewitt, 2011)

Temperature (°C)	ν (m^2 s^{-1})	Pr (@ film Temp)	ρ (kg m^{-3})	k_f (W m^{-1} K^{-1}, @ film temp)	c_p (kJ kg^{-1} K^{-1})
-73	7.59 E-6	0.737	1.75	0.0181	1.007
-23	1.14 E-5	0.720	1.39	0.0223	1.006
27	1.59 E-5	0.707	1.16	0.0263	1.007
77	2.09 E-5	0.700	1.00	0.0300	1.009
127	2.64 E-5	0.690	0.87	0.0338	1.014
177	3.24 E-5	0.686	0.77	0.0373	1.021

Film Temperature (°C):

$$T_{film} = \frac{(T_{db,center} + T_{skin})}{2} \tag{H.1}$$

Kinematic Viscosity (m^2 s^{-1}):

$$\nu = 1.05786^{-10} * T_{db,center}^2 + 8.84243^{-8} * T_{db,center} + 1.34661^{-5} \tag{H.2}$$

Prandtl Number (dimensionless):

$$\mathrm{Pr} = 5.5^{-7} * T_{film}^2 - 2.5818^{-4} * T_{film} + 0.71456 \tag{H.3}$$

Air Density (kg m^{-3}):

$$\rho_{air} = 1.2197^{-5} * T_{db,center}^{2} - 0.0050341 * T_{db,center} + 1.2963 \quad \text{(H.4)}$$

Air Conductivity (W m^{-1} K^{-1}):

$$k_{f,air} = -3.0714^{-8} * T_{film}^{2} + 7.9871^{-5} * T_{film} + 0.024132 \quad \text{(H.5)}$$

Specific Heat (kJ kg^{-1} K^{-1}):

$$c_p = 4^{-7} * T_{db,center}^{2} + 1.3377^{-5} * T_{db,center} + 1.006. \quad \text{(H.6)}$$

With these correlations, more air properties can be calculated such as the diffusivity (m^2 s^{-1}) of air (Equation H.7):

$$\alpha = \frac{k_{f,air}}{\rho * c_p}. \quad \text{(H.7)}$$

Now, the Reynold's number (Equation H.8) can be calculated to determine which Nusselt number needs to be used. The cow is assumed to be positioning herself that would provide her the optimal cooling (in cross flow), which is why the diameter (See Appendix E) is used in the Reynold's number equation as the characteristic dimension:

$$Re_D = \frac{Velocity * D_{cow}}{\nu}. \quad \text{(H.8)}$$

Since the Reynold's number falls in between 0.4 and 400,000, the following Nusselt number equation was used, which if for a cylinder in cross flow:

$$Nu_D = C * Re_D^{n} * \mathrm{Pr}^{0.33}. \quad \text{(H.9)}$$

The constants C and n used in Equation H.9 depended on the range of Reynold's numbers and are outlined in Table H.2.

Table H. 2. Constants for use with Equation H.9

Re_D	**C**	**n**
0.4 – 4	0.989	0.330
4 – 40	0.911	0.385
40 – 4000	0.683	0.466

Table H. 2 (cont). Constants for use with Equation H.9

4000 – 40,000	0.193	0.618
40,000 – 400,000	0.0266	0.805

Since all Reynold's numbers calculated fall in between 40,000 and 400,000, Equation H.10 was used:

$$Nu_D = 0.0266 * Re_D^{0.805} * \Pr^{0.33}. \tag{H.10}$$

Forced convection (W m^{-2} K$^{-1)}$ is a function of the Nusselt number and is shown in Equation H.11:

$$h_{cv} = \frac{Nu_D * k_{f,air}}{D}. \tag{H.11}$$

For natural convection, two dimensionless numbers need to be calculated first, which include the Grashoff and Rayleigh numbers (Equations H.12 and H.13).

$$Gr = \frac{\left(g * \tau * \left|T_{skin} - T_{db,center}\right| * D^3\right)}{\nu^2} \tag{H.12}$$

where,

g = gravity (9.81 m s^{-2})

τ = inverse of the film temperature (K^{-1})

$$Ra_D = \frac{\left(g * \tau * \left|T_{skin} - T_{db,center}\right| * D^3\right)}{\nu * \alpha}. \tag{H.13}$$

The Nusselt number for a cylinder having a wide range of Rayleigh numbers is shown in Equation H.14.

$$Nu_{nat} = \left(0.6 + \frac{0.387 * Ra_D^{\frac{1}{6}}}{\left(1 + \frac{0.559}{Pr^{\frac{9}{16}}}\right)^{\frac{8}{27}}}\right)^2. \tag{H.14}$$

Finally, the natural convective heat transfer coefficient (Equation H.15) can be determined:

$$h_{nat} = \frac{Nu_{nat} * k_{f,air}}{D} \,. \tag{H.15}$$