A MODEL FOR PREDICTING NON-EVAPORATIVE CONVECTIVE HEAT LOSS FROM THE SURFACE OF A BOVINE

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CHAPTER I

INTRODUCTION

In the first century before Christ, Marcus Vitruvious Pollio, a celebrated Roman architect and engineer, wrote: "The great hall is to be placed in the warmest part of the court; united to this are the ox stalls, with cribs towards the fire and the east, for oxen with their faces to the light and the fire do not become rough coated."

Man, furless and at the mercy of the elements has ever endeavored to improve his environment. Engineers, scientists and physicians have studied man's physiological and psychological reactions to various environments and have identified optimum conditions which can be synthesized for him by designers of heating and cooling equipment. Man has the knowledge and ability to cover himself with clothing suitable for his immediate environment.

On the other hand, birds and wild animals rely on instinct and physiological changes. Birds migrate to comfortable environments. The bear grows fat during comfortable summers and hibernates during uncomfortable winters. Animals stay on the lee side of cliffs and in the forests when winds are cold, and seek shade when the sun is hot. They are provided with a coating to suit their climate. Their covering thickens and becomes rough in preparation for winter, and achieves a cooler sheen when spring brings out the warmth of the sun.

Domestication of wild animals restricted their movement and kept them from selecting their own environment. Man soon realized that an animal's acclimatization was something to be considered when changing its natural habitat.

Many factors contribute to the interest in climatic physiology of cattle. The population shift to the hotter southwestern regions of the United States has been accompanied by an increased demand for milk and beef. Increased operating costs without a similar increase in milk and beef prices requires peak production for profitable operation. Development of artificial insemination and semen storage techniques permit selection and breeding of cattle on a global scale, subjecting cattle of a particular ecology to a wide variation of climates.

Milk can effectively supplement the diet in countries where many malnourished people live mostly on foods of plant origin. Most of these people inhabit high temperature regions where the higher producing cattle of European origin suffer heat stress. Cattle shelter specifications are needed to permit climate modification in areas where milk and beef production involve cattle poorly adapted to the climate. Consequently, considerable effort is being devoted toward defining comfort conditions for livestock and to finding practical means by which these conditions can be provided. This study is intended to make a contribution toward that goal.

CHAPTER II

OBJECTIVES

Early attempts at cooling cattle by convection utilized large fans blowing ambient air horizontally against and over a group of animals. Later methods included blowing air vertically from above and blowing evaporatively cooled air at an angle from above. A current project at Oklahoma State University will study cooling by blowing air up through slotted floors.

Measuring the cooling benefits of the various sources of moving air other than on a group production basis is very difficult because of the need for numerous measurements on uncooperative animals. Immediate response of an animal to micro-climatic changes could be studied in detail and with greater control if an inanimate model possessing heat transfer characteristics similar to those of a given type animal were used. Research was conducted to define the characteristics to be included in an engineering model for convective cooling response studies.

Specific objectives of the study were to:

1. Establish requirements for a thermal model of a bovine for predicting non-evaporative convective heat loss from the surface.

2. Produce, from experimental data, equations for predicting convective heat loss from the dry surface of an animal as related to:

   A. Reynolds Number.

   B. Grashof Number.

   C. Animal orientation in free air stream.

   D. Group arrangement of animals.

3. Evaluate the model as a representation of actual heat transfer effects in live animals by comparing predicted heat transfer indices with corresponding values determined by laboratory measurements on cattle.

CHAPTER III

THEORETICAL CONSIDERATIONS IN DETERMINING PREDICTION EQUATIONS FOR CONVECTIVE HEAT TRANSFER

A complete analytical solution to convection heat transfer from irregular surfaces or objects in cross-flow is difficult, and perhaps impossible to obtain. It is necessary to rely on experiments and to generalize the results by dimensional analysis. However, examination of the basic relationships has provided insight into some of the problems and permits a more intelligent approach to the experimentation. The basic conservation equations can be non-dimensionalized to gain this insight without actually solving the equations.

The application of dimensional analysis to heat transfer enabled W. Nusselt, in a fundamental paper published in 1915, systematically to coordinate for the first time earlier experimental results and to plan new experiments (26). In his papers on the application of velocity profile similarity to convection heat transfer from a body surface, he began with the following equations (42):

Conduction equation and Newton's law of cooling

$$dq = -k \frac{\partial T}{\partial n}\, dA = h\,(T_\infty - T_w)\, dA \qquad (3-1)$$

Momentum equations

$$\eta_\epsilon \rho\left(\frac{\partial u}{\partial t}+u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}\right)=-\frac{\partial P}{\partial x}+\mu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}\right)+G\beta\rho\Delta T \qquad (3\text{-}2a)$$

$$\eta_\epsilon \rho\left(\frac{\partial v}{\partial t}+u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial v}{\partial z}\right)=-\frac{\partial P}{\partial y}+\mu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2}\right) \qquad (3\text{-}2b)$$

$$\eta_\epsilon \rho\left(\frac{\partial w}{\partial t}+u\frac{\partial w}{\partial x}+v\frac{\partial w}{\partial y}+w\frac{\partial w}{\partial t}\right)=-\frac{\partial P}{\partial z}+\mu\left(\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\frac{\partial^2 w}{\partial z^2}\right) \qquad (3\text{-}2c)$$

Continuity equation (incompressible flow)

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}=0 \qquad (3\text{-}3)$$

Energy equation (no viscous dissipation)

$$\rho C\left(\frac{\partial T}{\partial t}+u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z}\right)=k\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}\right) \qquad (3\text{-}4)$$

where, expressed in familiar units,

q = heat flux, BTU/hr

k = fluid thermal conductivity, BTU/ft-hr-F

T = temperature at a point, F

T_w = temperature at body surface, F

T_∞ = temperature in free stream, F

h = heat transfer coefficient, BTU/hr-ft^2-F

t = time, hours

A = surface area, ft^2

n = distance in direction normal to surface, ft

Ne = Newton's Second Law Coefficient, lbf-hr^2/lbm-ft

ρ = fluid density, lbm/ft^3

x, y, and z = cartesian coordinates, ft

u, v, and w = velocities in the x, y, and z directions, ft/hr

P = pressure, lbf/ft^2

μ = fluid dynamic viscosity, lbf-hr/ft^2

G = Gravity field force, lbf/lbm

β = coefficient of thermal expansion, 1/F

C = fluid specific heat, BTU/lbm-F

These equations can be simplified further without loss of applicability when existing conditions are defined. In convective cooling of cattle, the following assumptions can be made:

1. Steady state
2. Constant fluid properties
3. Low velocities
4. Uniform surface temperature
5. Incompressible flow

The fluid flow process characteristics are defined by the momentum and continuity equations. With the listed assumptions and no body forces, the momentum equation (3-2a) reduces to

$$\eta_c \rho \left(u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} + w \frac{\partial u}{\partial z} \right) = - \frac{\partial P}{\partial x} + \mu \left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2} \right) \tag{3-5}$$

Body, or bouyant, forces are present but will be considered later. The continuity equation (3-3) remains the same. These equations with momentum equations for the y and z directions provide four equations for the unknowns u, v, w, and P.

Boundary conditions for an object at rest can be specified. Surface velocity is zero. Uniform velocity is assumed to exist at sufficient distance from the object and is constant in magnitude and direction with respect to position on the outer boundary of the flow field. With the coordinate system oriented such that the undisturbed fluid movement is in the x direction, the uniform velocity on the outer boundary is designated u_∞. All other velocities, i. e., v_∞ and w_∞, are therefore zero. The fluid pressure and density are also constant at all positions on the outer boundary of the flow field.

The boundary conditions and the differential equations can be made dimensionless by dividing all length parameters by a characteristic length λ and all velocity parameters by u_∞. Pressure can be made dimensionless by the term $Ne\rho u_\infty^2$ based on the constant density and prescribed velocity on the boundary. The dimensionless quantities, identified by (*), are

$$x^* = x/\lambda \qquad v^* = v/u_\infty$$

$$y^* = y/\lambda \qquad w^* = w/u_\infty$$

$$z^* = z/\lambda \qquad P^* = P/Ne\rho u_\infty^2$$

$$u^* = u/u_\infty$$

Substituting the dimensionless quantities in the continuity and momentum equations constitutes valid arithmetic operations and results in the following:

$$\frac{\partial u^*}{\partial x^*} + \frac{\partial v^*}{\partial y^*} + \frac{\partial w^*}{\partial z^*} = 0 \tag{3-6a}$$

and

$$\frac{N_e \rho u_\infty^2}{\lambda}\left(u^* \frac{\partial u^*}{\partial x^*} + v^* \frac{\partial u^*}{\partial y^*} + w^* \frac{\partial u^*}{\partial z^*}\right) = -\frac{N_e \rho u_\infty^2}{\lambda}\frac{\partial P^*}{\partial x^*} + \mu \frac{u_\infty}{\lambda^2}\left(\frac{\partial^2 u^*}{\partial x^{*2}} + \frac{\partial^2 u^*}{\partial y^{*2}} + \frac{\partial^2 u^*}{\partial z^{*2}}\right) \tag{3-6b}$$

This equation is still not dimensionless but is dimensionally homogeneous and can be reduced to one with dimensionless terms simply by dividing it by $N_e \rho u_\infty^2/\lambda$, resulting in

$$u^* \frac{\partial u^*}{\partial x^*} + v^* \frac{\partial u^*}{\partial y^*} + w^* \frac{\partial u^*}{\partial z^*} = -\frac{\partial P^*}{\partial x^*} + \frac{\mu}{N_e \rho \lambda u_\infty}\left(\frac{\partial^2 u^*}{\partial x^{*2}} + \frac{\partial^2 u^*}{\partial y^{*2}} + \frac{\partial^2 u^*}{\partial z^{*2}}\right) \tag{3-7}$$

The dimensionless boundary conditions are

$u^* = u_\infty/u_\infty = 1$ at sufficient distance from the object

$u^* = v^* = w^* = 0$ at the body surface

The solution to the differential equations will have the dimensionless dependent variables u^*, v^*, w^* and P^* as functions of the independent variables x^*, y^*, z^*, and the constant parameter $\mu/N_e \rho u_\infty \lambda$, which is the reciprocal of the Reynolds Number, designated Re. The solution therefore has the form

$$u^* = f_u(x^*, y^*, z^*, Re) \tag{3-8a}$$

$$v^* = f_v(x^*, y^*, z^*, Re) \tag{3-8b}$$

$$w^* = f_w(x^*, y^*, z^*, Re) \tag{3-8c}$$

$$p^* = f_p(x^*, y^*, z^*, Re) \tag{3-8d}$$

The significance of these equations may not at first seem apparent. However, when considering flow past two geometrically similar objects of different size, the above equations indicate the dimensionless velocity components and the dimensionless pressures are for both objects the same functions of the dimensionless coordinates and of Re.

If the object is kept at a temperature T_w while the upstream fluid has a temperature T_∞, a temperature field is established around the object. The field characteristics can be defined by the energy equation (3-4) which, based on the assumed conditions, reduces to

$$\rho C \left(u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} + w \frac{\partial T}{\partial z} \right) = k \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right) \tag{3-9}$$

The equation is independent of temperature level since only temperature differentials are involved. The temperatures can therefore be measured from an arbitrary reference value. The constant temperature T_∞ on the outer boundary of the flow field is commonly used as this reference.

If temperature differentials are based on T_∞ as a reference temperature, a dimensionless excess temperature ratio $\theta^* = \theta/\theta_w$ can be formed where

$$\theta = T - T_\infty \tag{3-10a}$$

$$\theta = T_w - T_\infty \tag{3-10b}$$

Multiplying the left side of Equation 3-9 by $u_\infty \theta_w \lambda / u_\infty \theta_w \lambda$ and the right side by $\lambda^2 \theta_w / \lambda^2 \theta_w$ is equivalent to substituting the dimensionless parameters in the equation resulting in

$$\frac{\rho C u_\infty \theta_w}{\lambda}\left(u^* \frac{\partial \theta^*}{\partial x^*} + v^* \frac{\partial \theta^*}{\partial y^*} + w^* \frac{\partial \theta^*}{\partial z^*}\right) = \frac{k \theta_w}{\lambda^2}\left(\frac{\partial^2 \theta^*}{\partial x^{*2}} + \frac{\partial^2 \theta^*}{\partial y^{*2}} + \frac{\partial^2 \theta^*}{\partial z^{*2}}\right) \tag{3-11}$$

Multiplying by $\lambda / \rho C u_\infty \theta_w$

$$u^* \frac{\partial \theta^*}{\partial x^*} + v^* \frac{\partial \theta^*}{\partial y^*} + w^* \frac{\partial \theta^*}{\partial z^*} = \frac{k}{C \rho \lambda u_\infty}\left(\frac{\partial^2 \theta^*}{\partial x^{*2}} + \frac{\partial^2 \theta^*}{\partial y^{*2}} + \frac{\partial^2 \theta^*}{\partial z^{*2}}\right) \tag{3-11a}$$

Multiplying the coefficient on the right hand side of the equation by $\mu Ne / \mu Ne$ modifies its form to

$$\frac{\mu}{Ne \rho \lambda u_\infty} \frac{k Ne}{C \mu} = \frac{1}{Re} \cdot \frac{k Ne}{C \mu} \tag{3-11b}$$

The dimensionless parameter $C\mu / kNe$ is the Prandtl Number, designated Pr, so the coefficient can be written 1/RePr.

Boundary conditions are

$\theta^* = 0$	in the free stream
$\theta^* = 1$	at the surface

To obtain a solution, the velocity components defined in equations 3-8 must be introduced into the energy equation. The solution, if it could be determined, would then be of the form

$$\theta^* = f_1(x^*, y^*, z^*, Re, Pr) \tag{3-12}$$

The heat exchange between a fluid and a body can be calculated with the heat transfer coefficient (h) defined by the equation

$$-k\left(\frac{\partial T}{\partial n}\right)_{surface} = h(T_\infty - T_w) = -h\theta_w \qquad (3\text{-}13)$$

in which n indicates distance in a direction normal to body surface. A dimensionless form of n can be written $n^* = n/\lambda$. Introducing the dimensionless parameter θ^*,

$$h\,\theta_w = k\left(\frac{\partial \theta}{\partial n}\right)_{surface} = \frac{k\,\theta_w}{\lambda}\left(\frac{\partial \theta^*}{\partial n^*}\right)_{surface}$$

or

$$\frac{h\lambda}{k} = \left(\frac{\partial \theta^*}{\partial n^*}\right)_{surface}$$

The dimensionless parameter $h\lambda/k$ is therefore the dimensionless temperature gradient at the surface and is named the Nusselt Number after W. Nusselt and designated Nu. By differentiating equation (3-12) with respect to the dimensionless parameter n^* and introducing the expression for Nu, for the assumed conditions

$$Nu = f_2(x^*, y^*, z^*, Re, Pr) \qquad (3\text{-}14)$$

The terms x^*, y^*, and z^* identify a specific point where Nu is a local Nusselt number at a point. The average Nu of the entire surface of the object is given by

$$\overline{Nu} = f_3(Re, Pr) \qquad (3\text{-}15)$$

Equation 3-15 assumes no bouyant forces and is valid only where free stream velocities are of such magnitude that velocities generated by bouyancy are negligible. In convection cooling of cattle, significant bouyant forces may result from a change in density of the fluid with temperature differences. In a gravity field, bouyancy forces arise when temperature differences exist in the fluid, and these forces produce free convection currents.

The bouyancy force per unit volume of fluid element is $G(\rho - \rho')$ where G is the gravity field force, ρ is the density of the fluid at a local point and ρ' is the density that the fluid would have at that point if it were not heated by the heat transfer from the object to the fluid. Since this difference will be small compared to the density itself, the fluid properties are still assumed constant. The term $\rho - \rho' = \Delta\rho$ can be expressed as a temperature difference by introducing a coefficient of expansion β defined, where V is specific volume, as

$$\beta = \frac{1}{V}\left(\frac{\partial V}{\partial T}\right) = \frac{1}{V}\,\frac{dV}{d\rho}\,\frac{\partial\rho}{\partial T}$$

Since air follows the laws of an ideal gas reasonably well,

$$\rho = \frac{P}{RT}$$

Therefore

$$\frac{\partial\rho}{\partial T} = -\frac{P}{R}\,\frac{1}{T^2} \quad and \quad \frac{\partial V}{\partial\rho} = \frac{dV}{d\rho} = -\frac{1}{\rho^2}$$

β then becomes

$$\beta = -\frac{1}{\rho}\left(\frac{\partial\rho}{\partial T}\right)$$

For small temperature variations, sufficient accuracy is retained in the equation

$$\Delta \rho = -\rho\beta\Delta T \tag{3-16}$$

and bouyancy force per unit volume becomes $-G\rho\beta\Delta T$. When this force acts in the x direction, it can be added to the momentum equation which then becomes

$$\eta_c \rho\left(u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} + w\frac{\partial u}{\partial z}\right) = -\frac{\partial P}{\partial x} + \mu\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2}\right) - G\beta\rho\Delta T \tag{3-17}$$

In a constant property fluid the temperature changes only in the neighborhood of surfaces with temperatures different from the free stream temperature. In this case, $\Delta T = \theta_w$ as defined in equation 3-10. When the equation is made dimensionless as in equation 3-7, but including bouyant forces, the equation becomes

$$u^*\frac{\partial u^*}{\partial x^*} + v^*\frac{\partial u^*}{\partial y^*} + w^*\frac{\partial u^*}{\partial z^*} = -\frac{\partial P^*}{\partial x^*} + \frac{1}{Re}\left(\frac{\partial^2 u^*}{\partial x^{*2}} + \frac{\partial^2 u^*}{\partial y^{*2}} + \frac{\partial^2 u^*}{\partial z^{*2}}\right) - \left(\frac{G\beta\lambda\theta_w}{\eta_c u_\infty^2}\right)\theta^* \tag{3-18}$$

This dimensionless equation contains an additional parameter $G\beta\lambda\theta_w/Ne\ u_\infty^2$. This parameter is inconvenient because it contains two values, u_∞ and θ_w, prescribed on the boundaries. The velocity u_∞ can be removed by multiplying by Re^2, resulting in

$$\frac{G\beta\lambda\theta_w}{\eta_c u_\infty^2} \cdot \frac{\eta_c^2\rho^2\lambda^2 u_\infty^2}{\mu^2} = \frac{G\beta\theta_w\lambda^3\rho^2}{\mu^2}$$

The dimensionless parameter $G\beta\theta_w\lambda^3\rho^2/\mu^2$ was named by Graetz as the Grashof Number designated Gr(42). The dimensionless momentum equation is then

$$u^* \frac{\partial u^*}{\partial x^*} + v^* \frac{\partial u^*}{\partial y^*} + w^* \frac{\partial u^*}{\partial z^*} = - \frac{\partial P^*}{\partial x^*} + \frac{1}{Re} \left(\frac{\partial^2 u^*}{\partial x^{*2}} + \frac{\partial^2 u^*}{\partial y^{*2}} + \frac{\partial^2 u^*}{\partial z^{*2}} \right) + \frac{Gr}{R^2} \theta^* \quad (3\text{-}19)$$

The general prediction equation for forced and free convection resulting from this development becomes

$$\overline{Nu} = f_4 (Re, Pr, Gr) \quad (3\text{-}20)$$

resulting in a system of only four variables. For a chosen value of Pr, for instance, Pr = 0.74, which applies to all diatomic gases at atmospheric pressure and 212 F, and for a given Reynolds number, $\overline{Nu}$ is a unique function of Gr. This has been substantiated by experiments with different geometrical dimensions, substances and temperatures (42).

CHAPTER IV

PREVIOUS RESEARCH IN ENVIRONMENTAL PHYSIOLOGY OF ANIMALS

An insight into the mechanism of heat control and dissipation is prerequisite to establishing the requirements for a valid thermal model. The literature review deals principally with studies of the physiology of heat regulation. The bovine species is a homeotherm, and although the transfer of heat from internal sources to the surface follows all the physical laws of heat transfer, it is effectively controlled by the homeothermic mechanism and influences the conditions associated with heat transfer from the surface to ambient air. The mechanism of heat flow from source to skin surface, from skin through hair coat, and from hair coat to ambient air is discussed. Attention is also devoted to temperature distribution patterns within and on the animal. Present methods of cooling cattle are reported. Reviews are made of literature on animal geometry and determination of surface area. Body shape, however, is included in Chapter VI with the description of model representation of the bovine. No previous applicable work on the influence of surface texture or roughness on convective cooling was located.

Evolution and Body-Temperature Regulation

Cattle breeds evolved in different climatic regions have adaptive characteristics harmonizing with the respective climates. Some characteristics, such as hair thickness and size of peripherae (ears, dewlap, navel flap, etc.) are recognized on sight. Others, such as neuroendocrine peculiarities are not visually apparent. Brody (16) summarizes the four rules governing climatic adaptations named after pioneer investigators.

1. Bergman Rule (1847) relates body size to climate. The larger breeds are found in colder parts of the geographical range, the smaller in the warmer parts. Pygmy races of elephant, hippopotamus, buffalo, and also pygmy man, exist in the hotter regions; larger breeds of cattle (Holsteins) are found in the colder American states with small breeds (Jerseys) in warmer states (16). Missouri Climatic Laboratory studies substantiate this conclusion: Small (Jersey) cows were more heat tolerant and less cold tolerant than large (Holstein) cows (15).

   Extra large skin folds give a larger ratio of surface to weight and more rapid heat loss per unit weight. The ears, dewlap, navel flap and vulva are much larger and more corrugated in loosely built heat-tolerant and cold-sensitive Indian cattle than in compactly built cold-tolerant and heat-sensitive European cattle. Merino sheep, with large skin folds and more surface per unit weight have a similar heat tolerance.

Perhaps larger surface area per unit weight should not affect climatic tolerance because the "surface law" postulates that heat production, like heat dissipation, is proportional to surface area (14). This may be true when bodies are geometrically similar and environmental temperatures are equivalent. Extra skin area in Indian cattle seems to be an evolutionary adaptation to the difficulty of heat dissipation in the tropics; in the same way extra fur is an adaptation for heat conservation in the arctic.

2. Wilson Rule (1854) relates the insulating cover to climates. Warmer-coated breeds of a given species are located in colder parts of the range, the less heavily coated in warmer parts. The downy, long, woolly, felting hair of European-evolved cattle, particularly Scottish Highland, is contrasted with the short, smooth, glossy, straight, stiff hair of Indian cattle. Seasonal hair growth complies to this rule. The thick sub-cutaneous fat of the arctic species similarly contrasts with the lean skin of tropical species. There are no differences in skin thickness (independent of sub-cutaneous fat) between Indian and European cattle (16).

3. Gloger Rule (1883) relates color to climate. Pigments in bare skin protect against ultraviolet radiation. Hair, on the

contrary, tends to be light-colored in the tropical animals to better reflect solar radiation, thus keeping them from becoming over-heated. Summer lightening and winter darkening of hair on cattle illustrate this rule although in some species the greater survival value of protective coloration against predators may mask the tendency of protection from radiation. Increasing sebum secretion with increasing temperature, which gives hair a reflective and protective sheen against solar radiations, also may be a factor.

4. Claude Bernard Rule (1876) more applicable to climatic conditions necessitating heat conservation, relates climate to changes in reactions not visually apparent. For instance, the temperature of more exposed parts (ears, face, legs, feet) of arctic species is adjusted for heat conservation by vascular feedback, whereby heat from the warmer arterial blood going to the periphery is used to warm the returning cooler venous blood. In this manner, heat loss from warm arteries to the surface is reduced, the surface temperature is lowered and the thermal gradient and heat losses from the body are minimized.

Methods of Cattle Cooling

Although intricate instrumentation and methods have been used in comparing comfort levels of various environments for cattle, all researchers recognize the ability of cattle to make their own selection when a choice exists. On occasions a cow would rather accept some hunger than leave the comfort of a shady spot. Consequently, temperature regulation in cattle will be reflected in their behavior in the field, i.e., their grazing habits.

In high thermal environments, loss of appetite in cattle tends to reduce heat production (27). Reduction in food intake to decrease the heat load influences their grazing habits.

A diagram compiled by Findlay (27) shows the grazing behavior of a variety of breeds of cattle exposed to different weather conditions. Under hot sunny conditions, grazing time of cattle is related to the proportion of tropical ancestry. Clipping the coat slightly alleviates the heat burden on the animals as judged by their grazing time. The grazing time of a temperate breed, Aberdeen-Angus, in overcast weather was much longer than in sunny conditions with no wind. Wind tended to lessen the effects of high temperature for all animals.

Seath and Miller (63) observed grazing habits of Jersey and Holstein cattle at air temperatures from 70 to 90 F and found that animals sought shade at an air temperature of about 80 F when their body temperatures had reached 102.4 F, about 0.7 F above normal. On days when shade temperature was about 85 F, animals spent only 11 percent of their time grazing, though at night, when air temperature was 81 F, 35 percent of their time was spent grazing.

Grazing behavior studies led many workers to investigate measures to increase food intake of cattle exposed to hot conditions. Water sprinkling, shelter, and increased air movement have been used singly or in combination as described by Seath and Miller (64, 65, 66). After two hours in an ambient air temperature of 88 F, the rectal temperature of Jersey cows was 104 F. Sprinkling them with water at 85 F reduced the rectal temperature by 0.9 F; sprinkling with water plus air movement reduced the rectal temperature by 1.2 F, while sprinkling with water in shade plus air movement reduced the rectal temperature by 1.8 F. Frye, Miller and Burch (29) showed that in air temperatures of 89 F cows of European breeds with access to sprinklers grazed 42 percent of the time while those with access to shade but no sprinklers grazed about 34 percent of the time.

Suitable shelters for hot semi-arid conditions have been investigated by Kelly, Bond and Ittner (44) and Kelly and Ittner (45). They showed that the highest rate of weight gains were made by cattle under a desert cooler which consisted of a three-sided shelter open to the north with an upper roof of aluminum, a sub-roof of three layers of hay and an evaporative cooler. Given a choice between a galvanized-iron roof shelter, a louvered shelter and a hay covered shelter with an iron sub-roof and no cooler, the animals chose the last of these and never chose the galvanized-iron roofed shelter.

According to Nelson, Berousek and Mahoney (58), in a somewhat less arid climate no gains in production from dairy cattle confined in an evaporative cooled enclosure were noted over those given simple shade.

Minett (54) investigated the effect of artificial showers, rain and wallowing on body temperatures of water buffalo, Zebu cattle, small hill cattle and sheep. His studies on the effect of natural rain under air temperatures of about 59 F show decreases of 1.6 to 3.8 F in rectal temperatures of small hill bulls and 1.4 to 3.7 F in the rectal temperature of sheep.

Four methods of cooling cattle with water were tested at El Centro, California, by Kelly, Bond, and Ittner (43). Spraying the animals, cooling drinking water, cooling air by evaporation, and cooling shade surfaces by evaporation were all of some benefit in increasing weight gains of beef cattle during hot weather. In all cases where weight gains were increased, feed use per pound gain was decreased.

Results of increased air movement are reported by Bond, Kelly and Ittner (11). With a 42-inch fan providing an average air velocity of 3.7 miles per hour during 70-day feeding periods, seven Hereford cattle made additional daily gains of 1.03 pounds in 1955 and 0.53 pounds in 1956, compared with check animals in a normal wind that averaged 0.6 miles per hour. The gains were assessed to improved cattle comfort by convective cooling, but cooling by convection ceases when moving air is at a temperature equal to or greater than the animal surface. At high air temperatures, wind may still be potentially effective for evaporating moisture from the animal, but studies indicate that under the conditions of increased air movement, convective cooling is probably primary (77).

Experiments have been conducted on the effect of ambient temperature on drinking habits and water consumption of cattle and buffalo (41, 57). Results suggest that provision of cool drinking water would be of value in mitigating the effects of high environmental temperatures on cattle.

Thompson, Worstell and Brody (73) point out that increasing water consumption by 24 gallons per day and changing the temperature of the water from 60 F to body temperature of about 106 F removes about 2850 kilogram-calories of sensible heat per day from the body.

Heat Sources and Heat Production

Qualitative sources of heat in cattle are defined and diagrammed by Findlay (27) (Figure 1). The heat produced by animals and its relationship to environmental temperature has been extensively investigated.

Herrington (38) showed the effect of environmental temperature on heat production in rats, mice and guinea pigs. For temperature increases from 15 to 30 C he determined the following relationships:

Rats	$H = 1879 - 43.2\ T$
Mice	$H = 2093 - 46.3\ T$
Guinea Pigs	$H = 1426 - 26.1\ T$

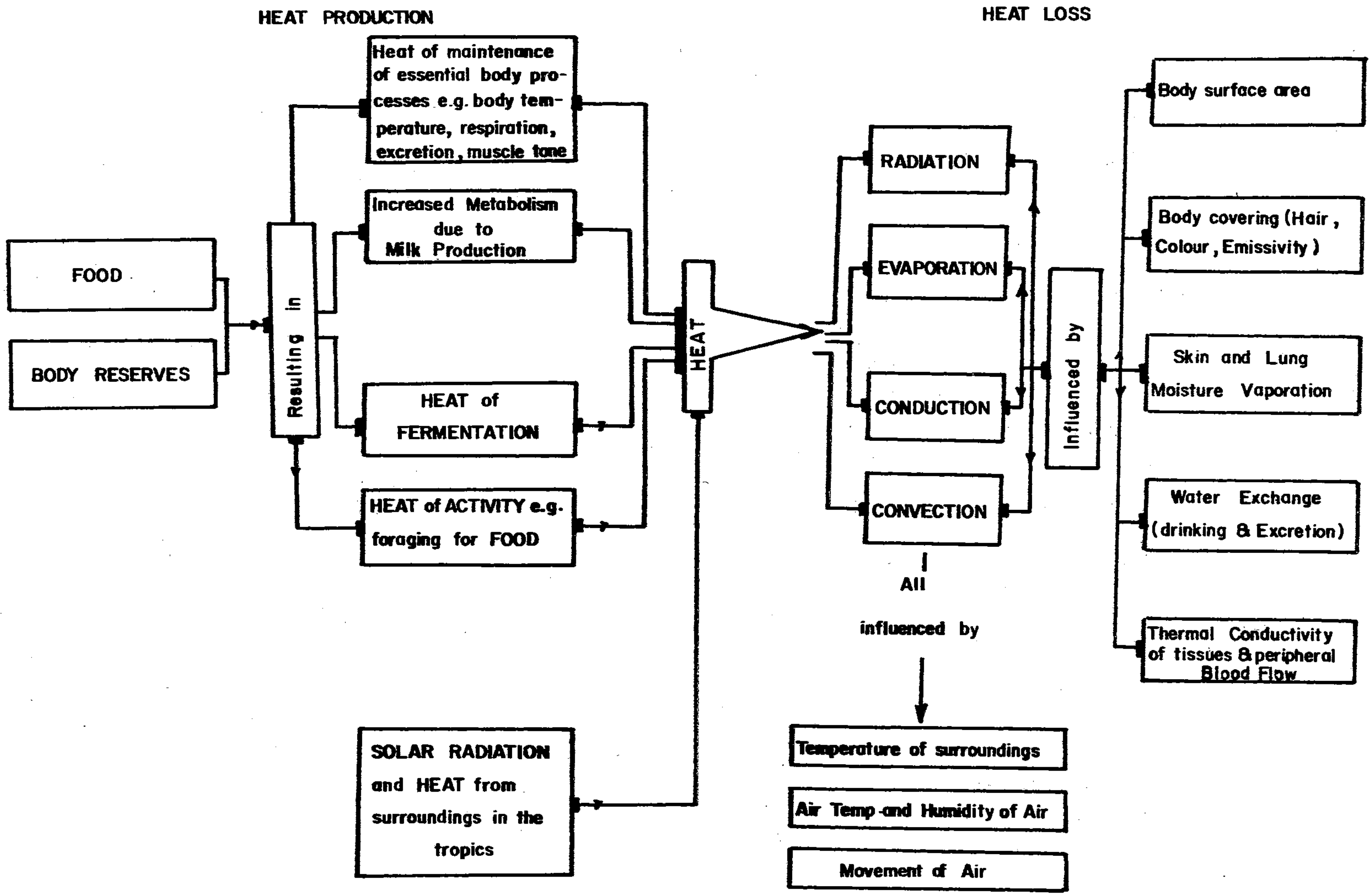


Figure 1. Diagram Showing the Principal Methods by Which Heat is Produced, Gained, and Lost by Cattle. Reproduced from Findlay (27).

where

H = heat produced, k-cal/square meter - 24 hours

T = environmental temperature, C

In the range from 30 to 35 C, data were gained from rats only, resulting in the equation

$$H = -503 + 42.8\ T$$

Brody (15) points out that the heat increment of feeding at its peak can account for up to 2600 k-cal/square meter-day. Kibler, Brody and Worstell (46) showed that total heat production (Btu/1000 pound body weight per hour) decreased with increasing temperature from about 3800 at 10 F to about 3000 at 80 F and dropped off more rapidly thereafter to slightly less than 2500 at 95 F. The latent heat portion, however, increased with temperature from about 600 BTU per 1000 pound per hour at 10 F to about 1300 at 65 F and thence more rapidly, increasing to about 2400 at 90 F. Increasing relative humidity from 40 percent to 90 percent increased total heat dissipation but decreased the latent heat portion. Increasing air velocities from 0.5 to 10 miles per hour increased total heat production at temperatures below 65 F but had no effect above 65 F.

Temperature Distribution in the Bovine

In using the general equation for convective heat transfer, temperature of the body surface must be characterized by a single number. Hardy and DuBois (36) pointed out the inconsistency of this, especially at low air temperatures when surface temperatures of the extremities may be substantially different than temperature of the trunk surface. Variations will lessen as air temperatures more nearly coincide with normal body temperature. Thompson and others (75) measured temperatures at six locations on the animal body in studying skin and hair temperatures. In subsequent analysis, however, they used the average temperature because they found little variation between hip, neck, back, sides and belly.

Blaxter (10) reports on the temperature of different parts of the body of a calf at environmental temperatures of 35, 20 and 5 C. Measurements of rectal temperature and skin surface temperatures of the tail, foot, shin, thigh, withers, chest and ear were made. With almost constant rectal temperature, the ear temperature varied from 38.5 C, the highest of all surface temperatures, at 35 C air, to 7 C, the lowest of all surfaces at 5 C air. The chest maintained the most constant temperature, varying only from 38.0 to 31.2 C. At 35 C air, temperatures of the different body points were relatively constant, the lowest temperature, 36.5 C, existing at the foot.

Internal temperatures of an animal body are more uniform than those of the surface, especially at lower air temperatures. Burton (19), in calorimeter studies on the human body reports the surface of the human body is 4 or 5 C lower than the interior temperature at a depth of several centimeters. He also shows that about 50 percent of the body is within one inch of the surface.

Rectal temperature is commonly used as an indication of body temperature in cattle although it has never been construed to represent the average. When under no stress, the normal rectal temperature of a cow, though variable for different species and different animals within a species, is recognized as 101 to 101.5 F. Brody, Dale and Stewart (18) report the lower rumen to have a temperature about equal to that of the rectum. Upper rumen temperature (6 inches from the top) was about 2 F higher and middle rumen about 1 F higher than lower rumen (18 inches from the top). In the same tests, a thermocouple was inserted into the jugular vein at a point nine inches below the ramus (the posterior branch) of the lower jaw and 28 inches above the olecranon (point or projection at the knee) of the right ulna. The right jugular blood temperature was constant at about 99 F down to 17 inches; then remained at 100.3 F to 36 inches from the point of insertion.

Heat Flow From the Animal Core to the Skin

All farm animals are homeotherms and attempt to maintain body temperatures within the range for optimum biological activity. To do this, a thermal balance must be maintained between heat produced or gained from the environment and heat lost from the environment. Figure 1 illustrates the main routes of heat production and loss. The heat produced must be dissipated by the four channels; radiation, conduction, convection and evaporation. Feed digestion and utilization of body reserves result in heat production for maintaining body temperature, cardiorespiratory activities and muscle tone. Milk production involves increased heat production, i.e., a non-lactating, resting dairy cow might produce about 12,000 calories per day, but the same cow yielding four gallons of milk per day might produce twice that amount (16). The heat of fermentation in the rumen also adds to heat produced.

To preserve homeothermy, farm animals must conserve heat in cold conditions, and dissipate it in hot conditions. Heat is generated in the core of the animal and transferred to the surface through tissues by conduction and through the blood stream by convection. The transport of heat from the core of an animal depends on the difference in temperature between the surface and deep core and, where peripheral blood flow is minimal, to some extent on thermal conductivity of tissues. Animals control this heat loss by alterations in peripheral blood supply.

Much more work on transfer of heat from the deep core to the periphery has been done with respect to the physiology of humans than of farm mammals. The most noticeable change in the human body on exposure to cold conditions is a fall in skin temperature (79). This is passively influenced by changes in total heat storage in the tissues, but much more by alteration of the distribution of blood between superficial and deeper tissues. Mean skin temperatures of clothed subjects may drop about one degree for every two degree decrease in external temperature. Wide variations occur on various body areas; the trunk surface varies least and the extremities most. Stimulation in one area may cause changes in other areas. In one experiment, application of a small ice bag on the back of the neck for 15 minutes caused a drop in skin temperature on the finger of 10 C and this low finger temperature continued for one hour after the ice bag was removed (79).

The construction of peripheral blood vessels in the zone of body cooling is a physiological phenomenon of major importance. Its effects have been studied by Kleiber (49), Bazett and McGlone (3), Burton and Bazett (20) and Hardy and DuBois (35).

Experiments on an anesthetized dog by Cooper, Randall and Hertzman (22) provide evidence of vascular heat convection to the skin from rhythmically contracting leg muscles under electrical stimulation. Increases in skin temperature were similar to those observed in man and limited to the area overlying active muscles. Temperature gradients in the leg were measured by inserting thermocouples subcutaneously and intramuscularly. With the onset of contractions, both muscle and cutaneous surface temperatures rose promptly, while subcutaneous temperature increased only slightly, so that during much of the work period the subcutaneous tissue was actually cooler than either the muscle or the skin surface. Only vascular heat convection to the superficial skin vessels would account for this temperature gradient. This interpretation was strengthened by a modification in the experiment. After completion of the work period, the overlying skin was detached, so that all vascular connections with the deeper skin were broken, then reattached. On repetition of the exercise, muscle temperature increased as rapidly as before, but skin temperature rose only slightly during the entire work period, while the course of the subcutaneous temperature was essentially the same as during the control work period. Certainly, these observations demonstrate the slight magnitude of direct heat conduction and emphasize the dominant role of vascular convection in temperature increases in skin overlying active muscle.

The vascular supply to the skin of cattle has been investigated by Goodall and Yang (31) who showed that there are three vascular plexuses (network of vessels) in cattle skin comparable to those in human skin. There is, however, one essential difference. Whereas arteries of the cutaneous arteriolar plexuses and veins of the venous plexuses run independently of each other in human skin, arteries and veins of the corresponding plexuses always run together in cattle skin. Many of the larger arteries in bovine skin are accompanied by two veins. This is apparently of significant importance in heat regulation in cattle (27).

Bazett (4) reports that as early as 1876 Claude Bernard recognized the importance of interchange of heat between an artery and its accompanying veins in body heat conservation. Warm blood in the artery moving to the periphery is cooled by the returning venous blood, reducing the temperature gradient and heat loss at the skin surface. Whereas the accompanying veins seem to be present only in conjunction with the large arteries in the limbs in the human body, they appear to be almost universally distributed throughout the skin in cattle.

Goodall and Yang (31) found that injection of India ink into intercostal (between the ribs) arteries of cattle perfused a large area of skin on the back and flanks. Most of the blood reaching those skin areas had passed the intercostal muscles, and the venous return to the heart and lungs must have been by a similar path.

Bazett (2), considering dissipation of heat during heavy exercise, points out that skin over active muscles shows an increase in temperature long before any increase in rectal temperature. Interchange of heat between intercostal muscles and blood in arteries and veins must therefore play a prominent part in heat regulation. The importance of this finding needs no emphasis because most active muscles in stressed cattle are the intercostals, since heavy breathing is one of the first defense mechanisms against heat stress.

Assessing the contribution of the extremities such as limbs and ears to heat dissipation is difficult. Rapid blood flow is often used to combat extreme cold, for example, in birds' legs or the ear of a rabbit. Goodall (30) demonstrated that the ear of an Ayrshire calf contains many anastomoses (connections between blood vessels), peripheral blood flow varies with the need of heat loss or conservation, and this vascular control is a significant heat regulation mechanism. A study by Beakley (5) shows ratios of thermal conductances from ear to environment and from body to ear before and after an environmental change from 35 to 18 C. Thermal conductances were defined as heat transfer coefficients based on ear to environment and rectum to ear temperature differentials. The ratio changed from 1/4 to 5. Since the ear to environment conductance remained relatively unchanged, the change in rectal to ear conductance is equivalent to a 20-fold increase in blood flow to the ear.

Heat is thus very effectively transferred from the centers of heat production in the core of an animal to the periphery by the blood, and this transference is modified by various anatomical arrangements.

Heat Transfer Through the Hair Coat

Heat transfer from skin to hair surface is measurably affected by hair coat characteristics such as thickness and length. Blaxter (10) found that in cool weather, as the fleece of a sheep grew, heat production fell to a minimum by the time the sheep had 30 to 40 millimeters of wool. The effectiveness of wool as an insulating layer is shown but quantitative evaluation is more difficult.

The coefficient of thermal conductance, K_c, was related to fleece thickness by Armstrong, and others (1) by the equation

$$K_c = 80.1 - 29.7 \log_{10} t$$

where t is average depth of covering over sheep in centimeters. They were less successful in characterizing the insulating value of a cattle coat. The numerical insulation value for a cattle coat, erected by piloerector muscles is lower than the value for sheep. They attribute this to the sparser hair coat, permitting some convection between hairs. Thermal insulation of 0.004 centigrade degrees - square meter - 24 hr/kcal provided by the eight millimeter hair coat of a steer was determined experimentally by measuring the total cooling resistance of the animal, and subtracting experimentally gained values for insulation of air and of tissues corrected for basal evaporative losses.

Berry, Shanklin and Johnson (6) studied four physical factors thought to affect thermal insulation of livestock hair coats. Multiple correlation of 38 readings taken at 90 F air temperature resulted in the following equation:

$$k = 1.125 + 0.0232\,S + 0.0179\,N - 0.0634\,D_b - 0.00783\,d$$

where

k = conductivity, BTU-in/hour-ft^2

S = depth-over-length ratio, percent

N = number of hairs, 1000 per in^2

D_b = bulk density of hair coat, lb/ft^3

d = hair diameter, microns

Partitioning of Heat Dissipating Mechanisms

Animals lose latent or insensible heat by evaporation. They lose sensible heat by radiation, conduction and convection. Heat dissipation by evaporation can be further divided into heat lost by vaporization from respiratory surfaces and that lost through vaporization from the skin surface.

Heat Loss Through Respiration

Heat loss by vaporization from the lungs is governed by the respiratory volume rate and humidity of inspired and expired air. Under influence of heat stress, cattle exhibit polypnoea (rapid breathing) to a degree dependent on the magnitude and duration of the heat stress (27). Increasing the rate of respiration causes an increased dissipation of heat by warming the inspired air and by increasing evaporation from respiratory passages and lungs. Heat exchange of warming inspired air is proportional to the mass in inspired air, its specific heat, and the temperature difference between inspired and expired air. Heat exchange, by evaporation is the product of the mass of vapor added to the air and the latent heat of evaporation of water. A higher convection coefficient would also result from the increased velocities associated with rapid breathing.

Although qualitative aspects of heat loss by cattle are similar to those of man, the quantitative relations are quite different. There are two fundamental ways in which cattle and other farm animals differ from man in heat loss mechanisms. First, cattle do not possess such a highly developed perspiration mechanism as man. Second, below a temperature of 80 F, cattle have a pulmonary ventilation rate per unit body weight about twice as great as man giving cattle a greater respiratory advantage (14). However, Brody (16) points out that the contribution of increased respiration to evaporative cooling in cattle may be offset partially by increased heat production from the work of panting.

Thompson, McCrosky, and Brody (75) show the percentage of total heat production lost by vaporization of water from the cow's body. At 0 F, only about eight percent of total heat production was dissipated through vaporization of water. The percentage increased to twenty percent at 50 F and rose rapidly to about 100 percent at 100 F.

Heat Loss by Surface Vaporization

Kibler and Brody (48) have shown the partitioning of evaporative cooling between outer body surfaces and the respiratory tract. Figure 2 indicates noticeable increase in cutaneous evaporative heat loss above about 65 F. This might be analogous to an outbreak of sweating in man. Quantitatively, however, the difference in sweating or skin surface evaporation rate between the cow and man given by Brody

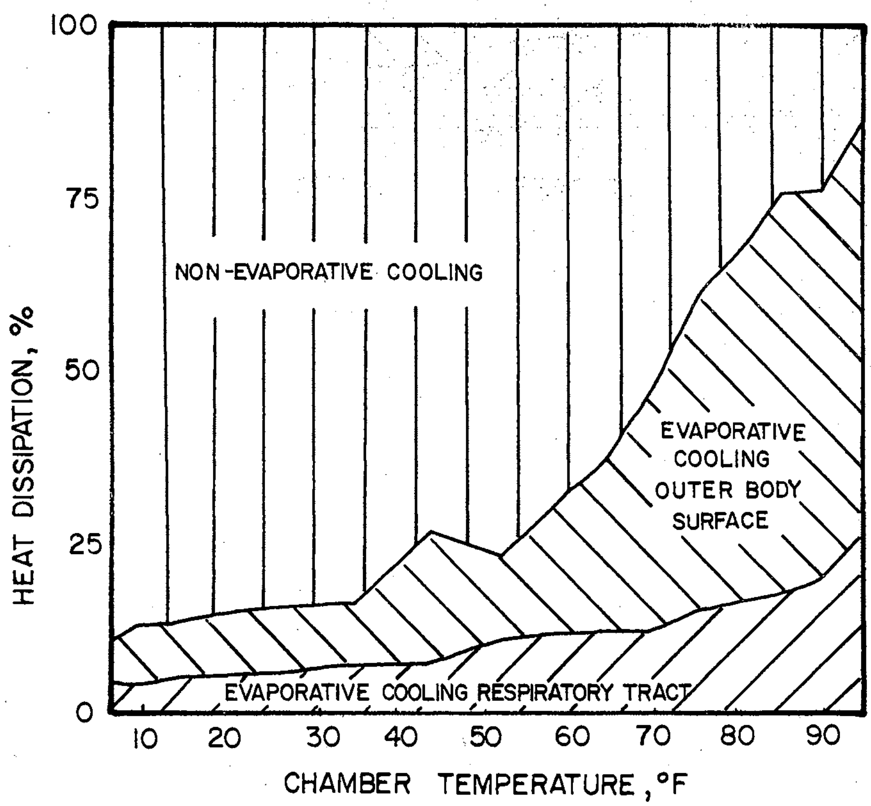


Figure 2. Partition of Total Heat Loss (or Heat Production) in Holstein and Jersey Cows. Reproduced from Kibler and Brody (48).

and others is large. A man in dry air can perspire at the rate of approximately 24 grams per kilogram body weight per hour (28) whereas cows could attain a maximum vaporization rate of only 1.5 grams per kilogram body weight per hour.

Although cattle are categorized as a non-sweating animal, heat loss by vaporization of body moisture on the surface plays a very significant role in dissipating heat. Findlay, Goodall, and Yang (28) have shown that cattle of temperate and tropical breeds possess sweat glands distributed over the entire surface of the body. One sweat gland is associated with each hair follicle. Glands have a poor blood supply and are of the apocrine type in which the secretory cells pinch off their product as it is formed. They point out that, anatomically sweat glands of cattle are like those of cats and dogs and not like man's highly active eccrine (excretory) glands, whose principle function is heat dissipation. In eccrine glands, sweat is formed and discharged from the glandular cell without disruption of the cell wall. Water may, nevertheless, be transpired through the skin of cattle by simple osmosis, in which sweat glands play no part.

Kibler and Brody (47) and Thompson, McCroskey and Brody (74) showed that for European evolved cattle, insensible weight losses, which are composed of metabolic weight loss and vaporized moisture, vary with surface area rather than body weight. For both European and Indian evolved cattle the ratio of evaporative loss to total heat production is the same for all animals and increases exponentially with environmental temperature up to 85 F regardless of breed, weight and productive level. Vaporized moisture loss per animal is lower for European than for Indian evolved cattle. Thompson and his co-workers produced prediction curves for the variation in total moisture loss with different environmental temperatures for Jersey and Holstein cattle. Moisture evaporated in pounds per hour per cow was 0.30, 0.61, and 1.92 at temperatures of 0, 50 and 105 F respectively. Yeck and Stewart (82) indicate that slightly more than 50 percent of stable moisture load was vaporized directly from the animal's body in work at the psychroenergetic laboratory in Missouri.

Thompson and others (77) show the effects of low, medium and high air velocities over a temperature range from 18 to 93 F on total evaporation cooling (Figure 3) and on surface temperature of lactating Brown Swiss, Holstein, Jersey and on non-lactating Jersey and Brahman cows. Vaporization at low air velocity gradually increases with increasing environmental temperature from 18 to 65 F, then more rapidly to 80 F when near maximum vaporization is reached. When vaporization at high velocity is similarly plotted, the rapid increase in vaporization begins nearer 80 F and continues up to 95 F.

Taneja (71), using a capsule technique developed for measurement of cutaneous evaporation, reports evaporation from the shoulder of Zebu-cross is significantly higher than that of Shorthorn. There was, however, no difference between the two breeds in the cutaneous evaporation from the belly area. In Zebu-cross, cutaneous water

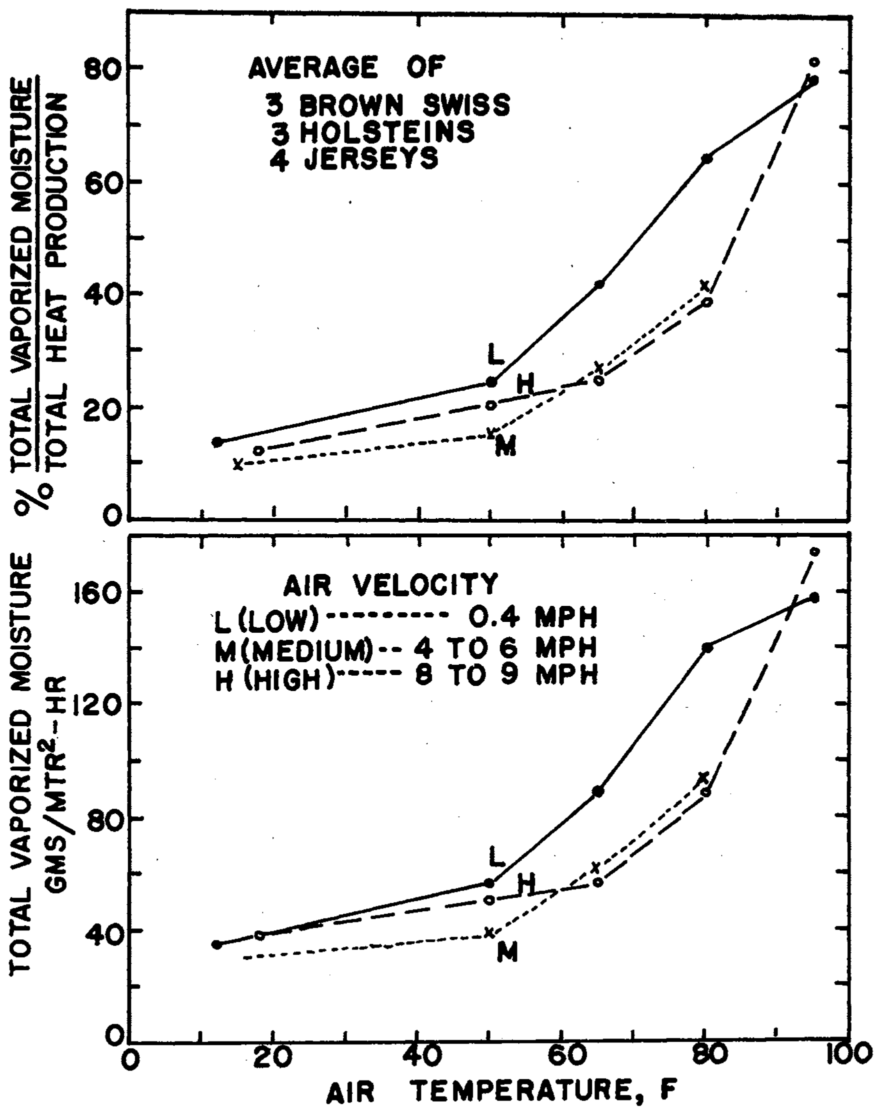


Figure 3. A Summary Chart Showing Average Values at Each of the Air Velocities for Jersey, Holstein and Brown Swiss. Reproduced from Thompson and Others (77).

losses from the shoulder area increased linearly with increase in skin temperature. In shorthorn, there was no important increase in cutaneous evaporation from the shoulder area, although skin temperature increased by about two to three degrees F. Zebu-cross had lower skin temperatures of the shoulder area when compared with that of Shorthorn. The lower skin temperatures were associated with higher cutaneous evaporation. Taneja (72), in a subsequent study, concluded that sweat glands in cattle are indeed functional and prevent body temperatures from rising. He based his conclusions on the skin vapor production response to the introduction of various drugs known to activate or suppress sweat gland activity.

McLean (53) studied anatomical distribution of moisture loss on the body of a calf. Maximum evaporation rates occurred behind the shoulder and were three times the minimum rates which were on the belly.

Heat Loss by Radiation

Thermal radiation emission from a body is characterized by the equation

$$q = \sigma \epsilon A T^4$$

where

q = heat emission, BTU/hr

σ = Stefan-Boltzman Constant, 0.174×10^{-8} BTU/hr-ft^2-R^4

ϵ = emittance of radiating surface, dimensionless

A = area of radiating surface, ft^2

T = temperature of radiating surface, R

Coincident with radiant heat dissipation from a body to its surrounding is a transfer of heat from the surroundings to the body. Net heat exchange therefore depends not only upon characteristics and temperature of the body, but of its surroundings as well. In addition, the shape factor, defined as the fraction of total energy emitted or reflected from the source impinging on the receiver, must be known. Perhaps the most difficult factors to evaluate are the emittance, the reflectance and the shape factor.

Stewart, Pickett and Brody (68) reported increases in reflectance from hair surfaces of Brown Swiss and Brahman cattle during three and four months confinement in a climatic chamber with environmental temperature slowly increasing from 65 to 95 F. The authors suggest the increase was apparently caused by temperature only since no solar radiation was present in the chamber. Solar radiation and other environmental variables, may modify the reaction considerably. The noted increases also could have been produced by time dependent changes in physical characteristics of hair from natural causes other than temperature.

Riemerschmid (61) indicated a difference in absorption of solar radiation by white and red hairy coats. In a subsequent study, Riemerschmid and Elder (62) evaluated the absorptance of hides with different colored hair coats. Their findings are summarized in Table I.

TABLE I

THE MEAN ABSORPTANCE OF HIDES OF DIFFERENT COLORS

White Zulu	Cream Simmenthaler	Red Afrikaner	Dark Red Sussex	Black Aberdeen Angus
49 percent	50 percent	78 percent	83 percent	89 percent

A companion study of absorptance by hair coats with various shades of red showed a range from 78 to 83 percent. For coats of like colors, they found no difference in absorption between smoother summer coats and hairy winter coats. Clipping the hair on a curly Sussex winter coat caused only two percent reduction in total absorptance. Part of this difference may have been due to a slight change in color caused by clipping.

Stewart and Brody (71) report the results of exposing Holstein, Jersey, and Brahman cattle to varying radiation intensities for a one week period. Data are presented on measured reflectance of different colored hairs from cattle and rates in the wave length range 290 to 1200 millimicrons. These data are summarized in Figure 4.

Shape factors for thermal radiation exchange between cows and their surroundings were studied by Perry and Speck (62). They present

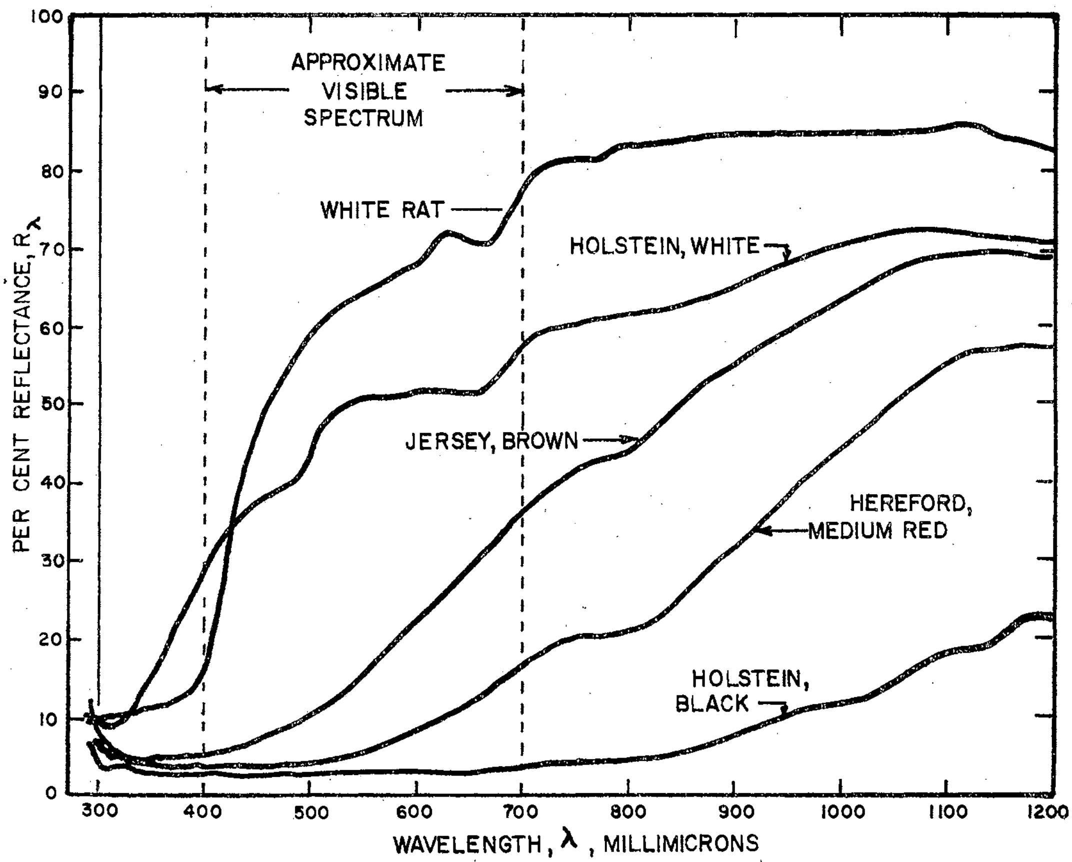


Figure 4. Typical Wavelength-Reflectance Curves for Hair of Animal Indicated. Reproduced from Stewart and Brody (68).

a method for evaluating the exchange, representing the cow by equivalent spheres. The radii of the equivalent spheres are 2.13 feet for exchange with floor and ceiling, 2.38 feet for sidewalls, and 2.02 feet for front and back walls. These values are 1.8, 2.08 and 1.78 times the heart girth.

Heat Loss by Conduction

An animal loses heat by conduction through contact with surrounding surfaces. The magnitude of loss depends on the thermal conductivities of the contacting surfaces, their areas and temperatures. Conduction heat loss from an animal surface to the surroundings would hardly seem significant for a standing animal because of the small contact surface (12).

Heat Loss by Convection

The loss of heat from surface to air (exclusive of evaporative losses) is not controlled by the homeothermic mechanism of the body but follows physical laws. In the absence of temperature differences between a standing animal and surrounding surfaces, Newton's Law provided the basis for the overall non-evaporative heat transfer. Newton's Law of Cooling states that the non-evaporative heat loss, Q, in time t, from a body of surface area A, is directly proportional to the temperature difference, $t_1 - t_2$, between the surface and surrounding fluid as given by the equation

$$Q/t = hA(t_1 - t_2)$$

in which h is a coefficient of heat transfer defined by the equation, and determined by experiment.

Thompson, Worstell and Brody (75) in investigations regarding the influences of surface temperatures on heat dissipation concluded that Newton's Law of Cooling is applicable to the non-evaporative cooling of cows under conditions where radiation and conduction are essentially eliminated.

Thompson and others (76) presented skin and hair temperature data for Holstein, Jersey, and Brahman cows exposed to environmental temperatures from 0 to 105 F, and from Brown Swiss cows and Brahman and Brown Swiss yearlings exposed from 40 to 105 F temperatures (Figure 5).

The data were analyzed and rationalized with Newton's Law of Cooling and estimates were made of the heat transfer coefficient in Newton's Law equation for several conductive or insulative layers. They found in comparing insulative characteristics from skin to hair with hair to air that the portion of total resistance attributed to hair to air increased from 60 percent to about 90 percent as chamber air temperatures were increased from 0 to 100 F. Figure 5 shows the skin temperatures increased from about 80 F at 5 F air temperature to about 102 at 100 F air temperature. Generally, hair temperature increased from about 50 at 5 F air to about 101 at 100 F air temperatures.

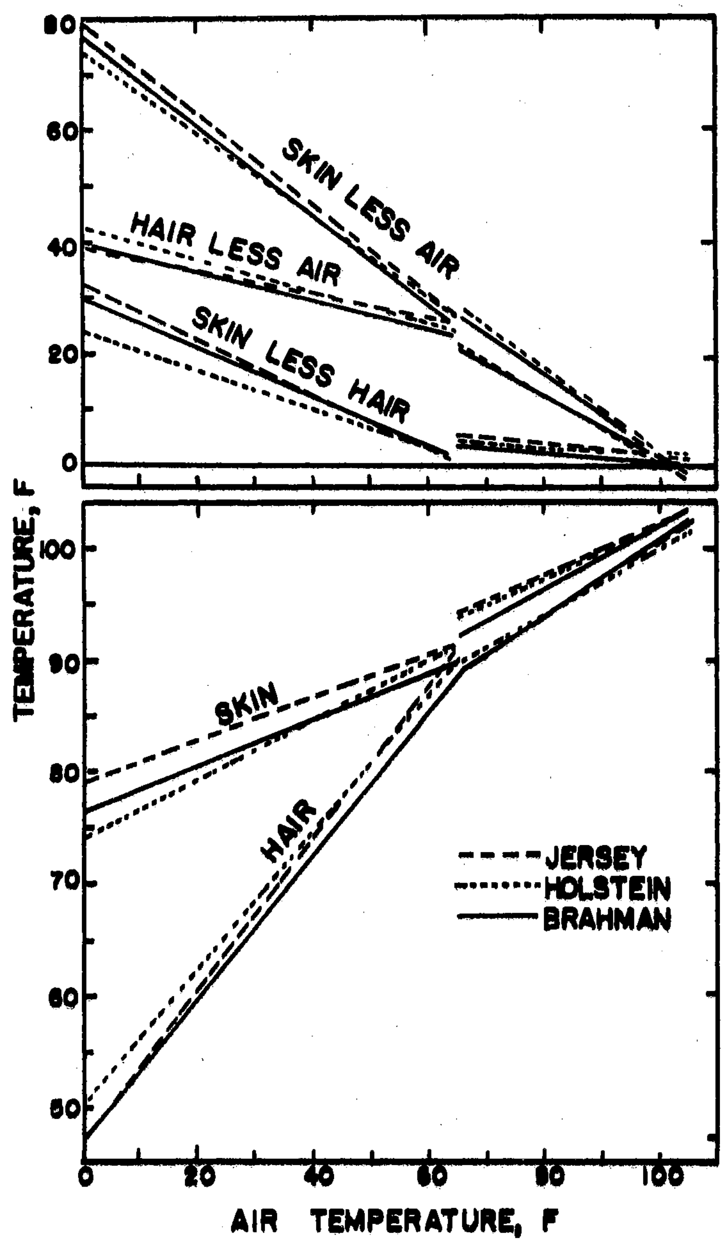


Figure 5. Skin and Hair Temperatures for Different Air Temperatures. Reproduced from Thompson and Others (76).

Winslow (79) determined convective interchanges from surfaces of humans by empirically deriving constants for a theoretical equation. He based his reasoning on the equation

$$C = K_c V^{1/2} (T_s - T_a)$$

where

C = convective loss, kg-cal/hr

Kc = a constant depending on the physical processes involved and on the posture and shape of the individual subject.

V = air velocity in cm/sec

T_s = mean temperature of body surface in centigrade degrees

T_a = air temperature, C

The proposition that moving air increases convective heat loss from the body in proportion to the square root of velocity began as a mere assumption, but the validity of the relationship for velocities up to 264 centimeters per second was shown by Winslow, Gagge and Herrington (81).

The hypothesis that heat transfer from surface to air in animals follows Newton's Law of Cooling is strengthened by a summarization of thermal conductances reported by Blaxter (10) and shown in Table II.

TABLE II

THERMAL CONDUCTANCE OF THE INTERFACE BETWEEN BODY SURFACE AND AIR IN DIFFERENT SPECIES

Species	Hair Coat	Thermal Conductance of Air Interface k-cal/m^2-24 hr-C
Sheep (7)*	Closely Clipped	130
Sheep (8)	With Fleece	140
Calf (10)	Normal Coat	174
Steer (9)	Normal Coat	167
Man (21)	Naked	167
Adult Pig (40)	Virtually None	125
Baby Pig (55)	Virtually None	172
Rat (37)	Normal Coat	151
Guinea Pig (37)	Normal Coat	142
Mouse (37)	Normal Coat	162

*Numbers in parenthesis refer to original references. Compilation of references credited to Blaxter (10).

In each instance the gradient of temperature is that from the surface in contact with the environment to the air; that is, in furred animals, surface temperature is that of the hair surface. No air velocities are given so the values are assumed to represent free convection coefficients. Any small differences might well be due to slight differences in air currents from experiment to experiment. The low value for the adult pig probably reflects the existence of a sparse hair coat which was neglected in the surface temperature measurements. Thermal conductance values listed are based on all non-evaporative heat transfer including radiation. The uniformity of conductance values suggests that surface to air heat transfer is relatively independent of fleece or hair characteristics.

Thompson, and others (79) studied the effect of wind on surface temperatures of cattle by varying air velocities and temperature. Increasing air velocity reduced skin and hair temperature roughly in proportion to the decline in ambient temperature. Lowering of surface temperature with increasing air velocity was apparently caused by increased convective, rather than evaporative, cooling since increased air velocity did not increase the vaporization rate except at 95 F.

In studying the influence of changing wind over a wide range of temperatures on respiration rate, pulse rate, pulmonary ventilation rate, rectal temperature, heat production, respiratory cooling and related values on cattle, Kibler and Brody (47) found definite cooling effects from increased air movement. Increasing air velocity from 0.5 to 10 miles per hour increased heat dissipation in cattle at 17 F air temperature. Similar increases in air velocity at 50 and 65 F had little effect on total heat dissipation. Non-evaporative losses were greater but were counteracted by a decrease in heat loss by vaporization from skin and respiratory tract. At 80 F, heat dissipation was depressed, but less so at high air velocities than at low air velocities. At 95 F, suppressed metabolism which developed under conditions of low air velocity was alleviated by introducing air velocities of eight to nine miles per hour.

Determination of Surface Area

The direct relationship between heat dissipation from the body and surface area prompted an investigation of the quantitative determination of this area. Hogan and Skouby (37) report the Meek Formula, developed in 1879, as the first attempt to relate surface area with some more easily determined parameter. Meek's formula for the human body is

$$S = K\,W^{2/3}$$

where

S = surface area, square centimeters

W = weight, grams

K = constant, 12.312 for adults

Brody (14) rationalized a similar equation relating surface to body weight based on the fact that in geometrically similar bodies, surface S, is proportional to the square of linear size L, and volume or weight W, is proportional to cube of linear size. Since

$$S \propto L^2 \text{ and } L \propto W^{1/3}$$

Therefore,

$$S \propto (W^{1/3})^2 = aW^{2/3}$$

He cautions that a growing animal does not always remain geometrically similar.

Brody and Elting (17), in describing a method of measuring surface area and its relation between growth in weight and skeletal growth in dairy cattle provide the equation

$$S = 0.15\ W^{0.56}$$

where

S = surface area, square meters

W = weight, kilograms

Blaxter and Wainman (9), in their report on the heat emission and energy metabolism of steers use the formula

$$S = 0.9\ W^{0.667}$$

where unit designations are the same as for Brody and Elting's equation. They do not, however, indicate the source of the equation.

Trowbridge, Moulton and Haigh (73) measured cattle and attempted to establish a constant K, for the Meek formula. Values ranged from $K \doteq 7.319$ for older, fat cattle to 10.474 in younger, thinner cattle. After DuBois and DuBois (23) developed an equation for man based on both height and weight, Hogan and Skouby (39) produced a more accurate equation for cattle. This equation was

$$S = W^{0.4} \times L^{0.6} \times K$$

where

S = surface, square centimeters

W = weight, kilograms

L = body length, measured from withers, centimeters

K = 217

The presence of a length factor in the equation of Hogan and Skouby allows for differences in body conformation. In designing a model to represent the shape and surface area of a bovine, the presence of a length factor in the equation for surface area provides a relationship not present in the other equations. Therefore, Hogan and Skouby's equation was selected for surface determination in the model design.

Summary

The physiology of heat regulation and dissipation in cattle is not only complex but also very effective. In cold surroundings, animals conserve heat and remain in relative comfort at temperatures as low as 0 F. As air temperatures rise and heat must be dissipated to prevent a rise in body temperature, the homeothermic mechanism responds with physiological changes to enhance removal of excess heat. Control is excellent for heat transport from internal sources to the outer surfaces of the skin but less effective between the skin surface and outer hair surface. Beyond the hair an animal has no control over heat exchange other than to seek the most favorable ambient conditions.

When air temperatures are low, animals conserve energy by reducing blood flow and heat transport to the surface and surface temperatures drop below internal body temperature. A wide variation of temperatures occurs over the body surface. In subzero climates, surface temperatures are often only slightly above freezing. In contrast, when air temperatures rise above comfort levels, blood flow is increased, inducing uniformly high surface temperatures to maximize the surface to air temperature gradient.

Equations are available for determination of surface area of cattle by relating it to more easily measured quantities. Surface area can be approximated from animal weight, and determined with somewhat better accuracy by relating it to both weight and length.

CHAPTER V

INVESTIGATIONS ON MIXED CONVECTION

In the absence of all outside sources of air movement such as blowers or natural wind currents, the only fluid motion present is that generated by body forces within the fluid. Body forces created by density variations in the fluid can cause significant fluid motion and therefore affect the energy transport within the fluid. Heat transfer by fluid motion of this type is called free or natural convection.

When air currents from other sources are present and are substantially greater than fluid motion generated by body forces, thermal transport from a surface to the fluid is designated as forced convection. Obviously, no heat loss by forced convection can occur unless a temperature differential exists, in which case body forces are also present. Therefore forced convection heat transfer is always accompanied by some, even though sometimes relatively insignificant, free convection fluid flow. Whether the heat loss caused by free convection is significant or not depends on the relative magnitude of the body forces compared to other forces acting on the fluid. When the relative magnitudes are such that both types of forces are significant, the heat loss is regarded as mixed convection.

Although considerable attention has been given to analytical and experimental solutions of the boundary layer equations for pure free and pure forced convection, only in recent years has an attack been made on mixed flow. In 1953, Eckert, Diaguila, and Curren (24) studied mixed convective heat transfer in a short tube with free convection siding and opposing the forced flow. They included a fifth dimensionless parameter, length/diameter, to the equation Nu = f(Re, Gr, Pr), but indicated this addition to be peculiar to tube flow. In the mixed flow region with aiding flows, the heat transfer coefficients were always larger than the larger of the calculated forced flow or free flow coefficients.

In 1955, Eckert, Diaguila and Livingood (25) reported on free convection effects for turbulent flow through a vertical tube with gravity flow opposite in direction to forced flow. In addition to determining limits between forced, mixed and free flow regions, they reported reduction in heat transfer rates for mixed opposing as compared to either free or forced flow. Hallman (33) also analyzed mixed convection heat transfer in vertical tubes with uniform heat generation. In 1960, Tao (70) used an analytical approach to heat transfer problems of combined free and forced convection by a fully developed laminar flow in a vertical channel of constant axial wall temperature gradient.

The aforementioned studies and others have confined their investigations to mixed convection in tubes, channels and other enclosures so none of the equations derived are directly applicable to the problem of cooling cattle.

Yuge (83) conducted experiments on heat transfer from small spheres including combined natural and forced convection with air as the cooling fluid. He related the mean of the measured values for Nusselt number for mixed flow with the Nusselt number for forced flow using the Nusselt number for natural flow as a parameter. This is consistent with the equation Nu = f(Pr, Re, Gr) (Eq. 3-20 of Chapter III). Klyachko (50) followed with a very critical analysis of Yuge's work and submitted what he considered preferable formulae for heat transfer with combined free and forced action. His proposed formulae, however, also show Nu to be a function of Re, Gr, and Pr. Both sets of equations relate to small spheres where the ratio of object diameter to boundary layer thickness is not necessarily large.

An analytical solution to the boundary layer equations for combined external flow was made by Sparrow, Eichhorn, and Greg (67). A study was made of similar solutions which constitute a set of exact solutions of the laminar boundary layer equations. They began with the boundary layer form of the equations of the basic conservation laws; mass, momentum and energy with steady, non-dissipative, constant property flows. Guided by previous experience with boundary layer problems, they proposed a new independent variable, η , and constructed two new dependent variables, both functions of η

to reduce the partial differential equations to a corresponding pair of ordinary ones. The resultant pair of equations were then solved for aiding flow using numerical techniques and 4.0 $Nu/Re^{0.5}$ was plotted versus $Gr^{0.5}/Re$ (Figure 6). The interesting feature of the relationship is that the heat transfer results for mixed flows show a surprisingly small deviation from the envelope formed by the asymptotic limiting lines for free and forced flow. The maximum error that could occur in heat transfer predictions based on the envelope lines rather than the mixed convection line would be 23 percent. Heat transfer from mixed aiding flow is thus somewhere between 0 and 23 percent higher than pure free or pure forced flow.

In Chapter III, the relative importance of pure free or pure forced convection in a mixed flow situation is shown as a function of the magnitude of the Gr/Re^2 ratio. However, no values are given for the ratio which could be used to identify a given condition as pure free, pure forced or mixed flow. The solutions of Sparrow, Eichhorn, and Greg provide a readily available criterion for defining the heat transfer based on this ratio for aiding flows. The authors suggest fluid flow might be considered effectively pure (either forced or free) if the heat transfer deviates by no more than five percent from the value associated with completely pure flow. Application of this five-percent criterion to their results identified mixed flow as occurring in the range $0.3 < Gr/Re^2 < 16$. For opposing flows, the

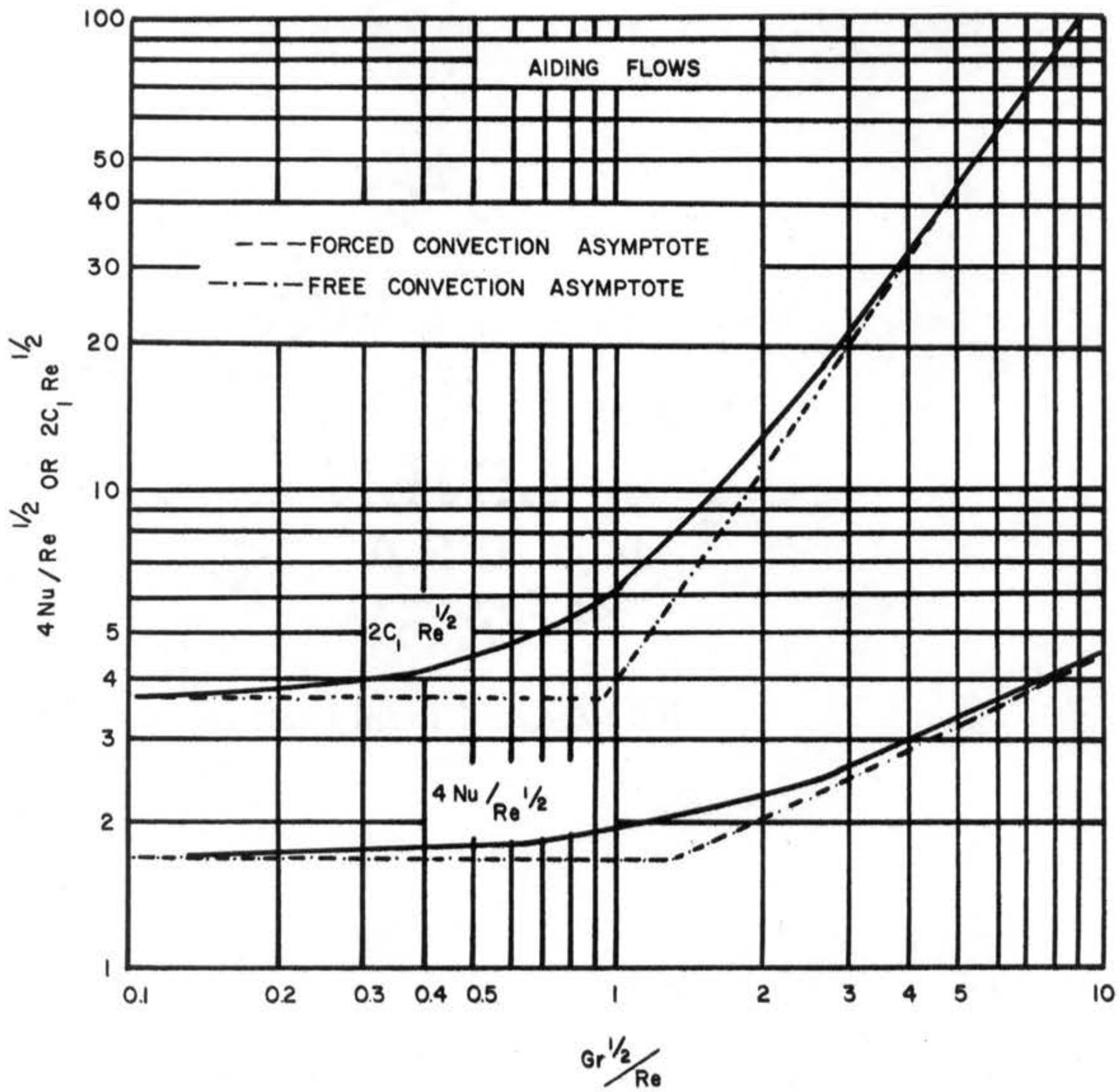


Figure 6. Heat Transfer and Friction Factor Results for Aiding Flows (Uniform Wall Temperature, Pr = 0.72). Reproduced from Sparrow and Others (67).

lower limit still applies but no criterion for purely free convection flow is available since solutions were not obtained beyond the separation point. These recent studies verify, to a degree, the insight possessed by McAdams (52) who suggested, for the mixed flow region, the use of the larger heat transfer coefficient obtained from forced and free convection correlations. His rule, however, was primarily intended for flow in tubes.

Although the practice of utilizing forced air currents to cool cattle was presumed to constitute a mixed convection situation, calculations were made to classify the flow according to the criterion of Sparrow. Assuming a bovine body could be represented by cylinders ranging from 0.25 feet to 2.0 feet in diameter and forced air velocities up to five feet per second would be prevalent, a value of Gr/Re^2 was computed and plotted versus free stream velocity with cylinder diameter as a parameter in Figure 7. For temperature differentials up to 15 F, mixed flow occurs on some member of the body for forced air velocities up to 2.4 feet per second. This velocity range presumably constitutes only the lower portion of the range that would be practical for cooling cattle. If the Gr/Re^2 criterion for defining flows were based on conditions more nearly identical to those expected in a cattle cooling system, the information in Figure 7 would justify limiting the study to forced convection. However, the criterion is based on the assumption of similarity

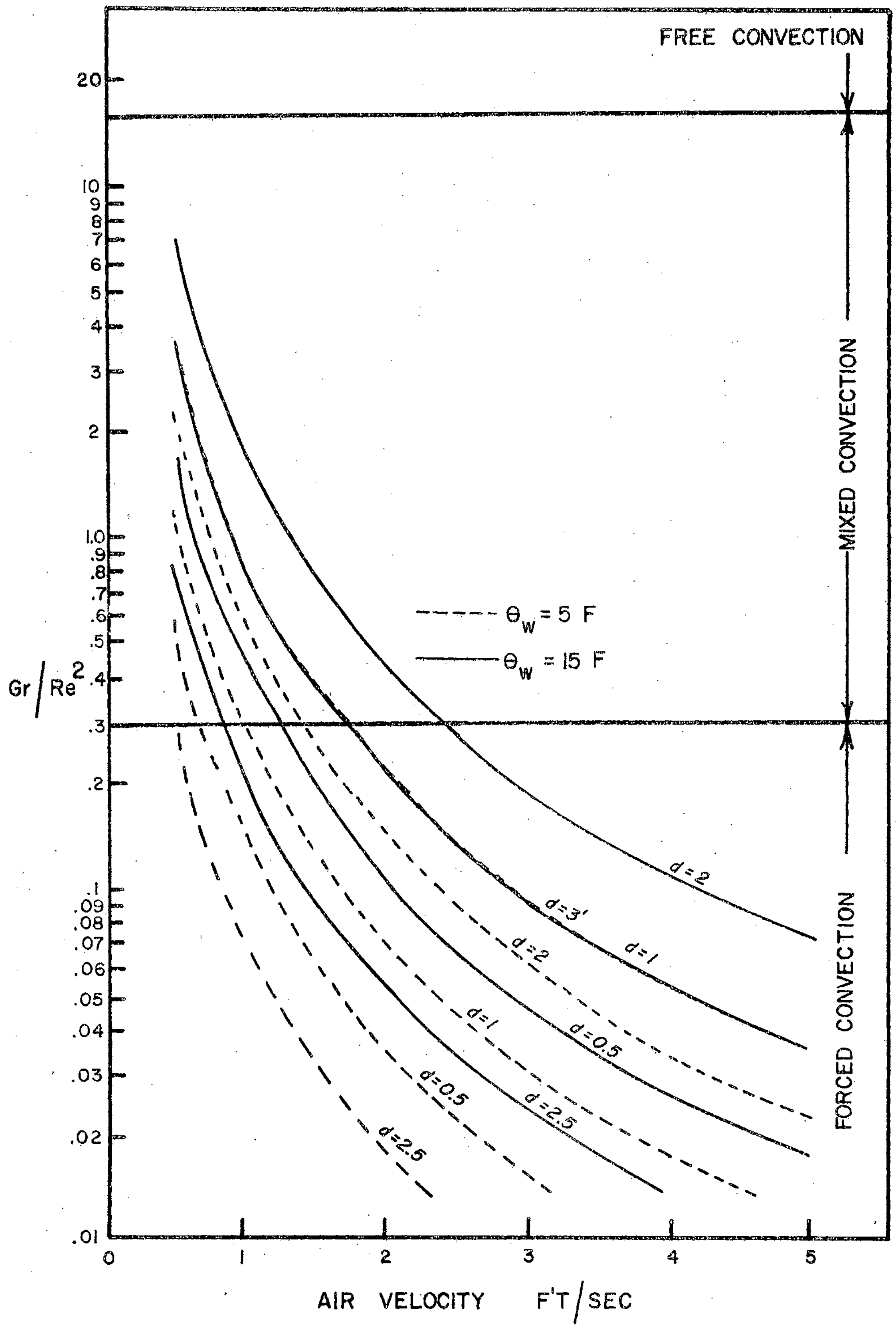


Figure 7. Relationship of Expected Values of Gr/Re^2 to Sparrow's Criterion for Convection Classification.

which would depart from actual conditions on the windward side of an object and would be irrelevant on the leeward side where separation occurs. Consequently, the curves may require shifting to be appropriate for cattle cooling, and the possible influence of body forces must be considered.

CHAPTER VI

EXPERIMENTAL DESIGN

Introduction

Heat exchange between fluids in motion and regular surfaces such as flat plates and cylindrical tubes with axial flow can be calculated. For irregular shapes and cylinders in cross-flow, where flow separates from the surface, present understanding of underlying processes is insufficient to permit direct solution. Therefore a model and dimensional analysis were used to develop prediction equations for heat transfer from a bovine body.

A knowledge of the variables influencing a physical system is a prerequisite to a meaningful investigation of the system. The variables must be inter-related and combined to form appropriate dimensionless parameters. The convection heat transfer equations developed in Chapter III provide insight on the variables and inter-dependence of the physical quantities for the formation of appropriate parameters.

Identification of Variables

The qualitative characteristics of the pertinent variables can be expressed by six independent dimensions, namely, Force (F), Mass (M), Length (L), Time (T), Temperature (θ), and Heat (H). Since the problem is one of heat flux (Q), which Newton's Law of Cooling identifies as a function of temperature differential (θ_W), coefficient of heat transfer (h), and surface area which can be characterized by a length (λ), the list of variables must include three of these four. Dissipated heat flow through and into the fluid is controlled by fluid thermal conductivity (k) and fluid specific heat (C). Fluid motion identifies the system as inertial, and flow can be characterized by fluid velocity (u_∞), density (ρ), viscosity (μ), and Newton's Second Law Coefficient (Ne). Body forces generate fluid motion as the result of volume change with temperature (β), and gravity forces (G) acting on the fluid. Thus, the pertinent variables and their dimensions are as listed in Table III.

TABLE III

PERTINENT VARIABLES FOR CONVECTION HEAT TRANSFER STUDY

No.	Symbol	Variable	Units	Dimensions
1.	Q	Heat flux	BTU/hr-ft^2	$HT^{-1}L^{-2}$
2.	θ_W	$T_W - T_\infty$	F	θ
3.	λ	Characteristic length	ft	L
4.	k	Fluid Conductivity	BTU/hr-ft-F	$HT^{-1}L^{-1}\theta^{-1}$
5.	C	Fluid specific heat	BTU/lbm-F	$HM^{-1}\theta^{-1}$
6.	u_∞	Free stream velocity	ft/hr	LT^{-1}
7.	ρ	Fluid density	lbm/ft^3	ML^{-3}
8.	μ	Fluid viscosity	lbf-hr/ft^2	FTL^{-2}
9.	Ne	Newton's 2nd Law Coef.	lbf-hr^2/lbm-ft	$FM^{-1}L^{-1}T^2$
10.	β	Coef. of thermal exp.	1/F	θ^{-1}
11.	G	Gravity field force	lbf/lbm	FM^{-1}

The rank of the dimensional matrix for the variables is six, so by Langhaar's Rule (51), which states that the number of independent, dimensionless parameters always equals the difference between the number of pertinent variables and the rank of the dimensional matrix, the variables can be combined in five parameters or Pi terms. The general prediction is of the form

$$\text{Pi} - 1 = f(\text{Pi-2, Pi-3, Pi-4, Pi-5})$$

or, using the dimensionless parameters, three of which were developed in Chapter III,

$$\frac{Q\lambda}{\theta_w k} = f\left(\frac{C\mu}{k\,Ne}, \frac{u_\infty \lambda \rho Ne}{\mu}, \frac{Ne\,G\lambda^3 \rho^2}{\mu^2}, \beta\theta_w \right)$$

Pi-1, $Q\lambda/\theta_w k$, a form of the Nusselt Number, describes the ratio of heat transfer through film to transfer into fluid. It is also an index of the ratio of characteristic physical dimension to boundary layer thickness. In this experiment it is the average Nusselt Number, $\overline{Nu}$.

Pi-2, $C\mu/kNe$, the Prandtl Number, is an index of the ratio of convected heat times viscous forces and conducted heat times inertial forces. Pr contains only thermo-physical properties of the fluid and is a composite property.

Pi-3, $u_\infty \lambda \rho Ne/\mu$ is the Reynolds Number, a measure of the ratio of inertial forces to viscous forces.

Pi-4, $NeG\lambda^3\rho^2/\mu^2$, and Pi-5, $\beta\theta_w$, are commonly combined in heat transfer problems to form the Grashof Number. Combining the two parameters is suggested in the theoretical development leading to the dimensionless equation (3-18) where the variable G never appears except with β in the form $G\beta$, implying both have the same exponent. Experiments with different geometric dimensions, substances and temperatures have verified the validity of combining the parameters (42). The meaning of Gr is obscure but can be clarified somewhat by considering its derivation from the differential equation. Gr resulted from the product of Re^2 and $G\beta\lambda\theta_w/Ne u_\infty^2$. Gr/Re^2 can be interpreted as an index of the ratio of bouyancy force per unit volume ($\rho G\beta\theta_w$) and the inertia force per unit volume (ρu_∞^2 Ne/L).

The general prediction equation was therefore simplified from one with five parameters to four, as shown in Chapter III (Eq. 3). The equation obtained was

$$\overline{Nu} = f(Pr, Re, Gr)$$

Selection of Characteristic Length

In the identification of pertinent variables, characteristic length, λ was included to represent all physical length dimensions. In geometrically defined bodies, all length and functions of length such as area and volume can be related to one characteristic length. In heat transfer studies on bodies of common shape, a standard dimension is used, for instance, distance from leading edge to point of interest in flat plates, and diameter for cylinders and spheres.

The characteristic length for the model bovine had to be one appropriate for heat transfer relationships and one which could be readily related to other physical dimensions of cattle. The most common length term used in describing cattle size is the heart girth, the circumference of the trunk behind the fore legs. By dividing by pi, this can easily be related to diameter of the trunk which is one of the cylinders in the model. Consequently, heart girth divided by pi, or trunk diameter was selected for characteristic length, .

Model Requirements and Construction

Air movement, forced or natural, about an animal body will cause separation of flow from the surface. Therefore the quantitative relations among the parameters in the general equation cannot be determined analytically; they must be determined by experiment.

Prediction equations were developed by conducting experiments on a model representing a bovine body. The requirements of the model were based on experimental information describing the size, shape, and heat generation and heat transfer characteristics of cattle as reviewed in Chapter IV.

The complexity and effectiveness of the homeothermic mechanism in developing most advantageous temperature distribution patters for heat conservation or dissipation have been described in Chapter IV. As ambient air temperatures approach those of the body, surface temperatures are maximized to maintain the most favorable surface-to-air gradient over the entire body. The circulatory system effectively transports heat to all extremities, resulting in a uniform temperature over the body surface. To model the internal system would be a formidable task, but its effectiveness in establishing a uniform surface temperature simplifies model representation for cooling studies. Thus, one model requirement was a uniform surface temperature.

Convection heat transfer depends on fluid flow characteristics which in turn are influenced by the shape of the object. Consequently, the model had to simulate the geometric configuration of a bovine body. An idealized prototype bovine shape was developed utilizing a combination of cylinders and tubes representing trunk, head and neck, and legs. The length-diameter ratio of each member equalled the length-diameter ratio of the body section represented. Brody (14) compiled dimensions for livestock which indicate the average values for an 800-pound cow listed in Table IV.

TABLE IV

DIMENSIONS FOR 800-POUND COW

Section	Length	Diameter
Trunk	4.8 ft.	2.0 ft.
Head and Neck	1.8 ft.	0.75 ft. horizontally 1.0 ft. vertically
Legs below trunk	2.0 ft.	0.33 ft.

Hogan and Skouby's equation (39) for surface area determinations was converted to foot pound units to read

$$S = KW^{0.4} L^{0.6}$$

where

S = surface area, ft^2

W = weight, pounds

L = length of trunk from point of withers to base of tail, ft.

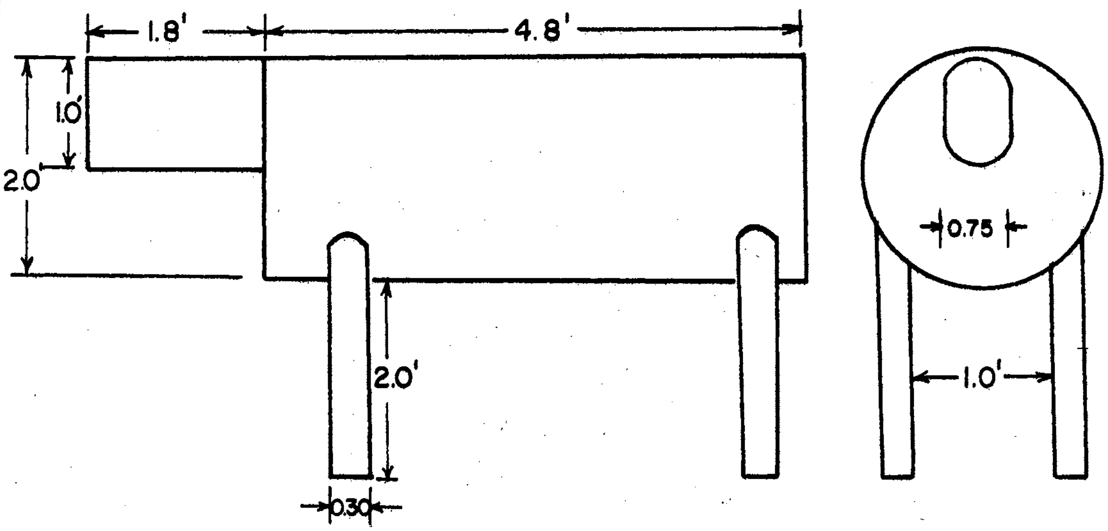
K = a constant equal to $1.32\ ft^{1.4}/lb^{0.4}$

Substituting the dimensions taken from Brody,

$$S = 1.32 \times (800)^{0.4} \times (4.8)^{0.6} = 49.2\ ft^2$$

This area would be approximated in an idealized prototype by the surface area of the sections assembled as shown in Figure 8.

A full size model was considered difficult to handle. However, a large length scale factor n, presented problems in establishing conditions where the Grashof Number would be within a realistic range for cattle cooling because of the appearance of λ^3 in Gr. That is, for Gr-prototype to equal Gr-model when λ-prototype equals $n\lambda$-model, some other quantity in Gr-model would have to be increased by a factor of n^3. The only other easily variable quantity in Gr is temperature differential. Therefore, maximum allowable temperature range controlled maximum length scale. Under conditions where artificial cooling by convection is practical, a maximum surface-air differential of 15 F for a cow was considered appropriate. Maximum allowable surface temperature for the model was arbitrarily set at 150 F. The experiment was conducted where air temperatures up to 100 F could be expected, permitting a differential of 50 F. A maximum length scale



Trunk perimeter surface:	30.20 ft^2
Trunk end surface:	6.28 ft^2
Head and neck surface:	4.55 ft^2
Leg surface:	8.30 ft^2
TOTAL	49.28 ft^2

Figure 8. Prototype Dimensions and Arrangement

of 1.5 was then selected based on the relationship.

$$n = \sqrt[3]{50/15} = 1.5$$

To avoid unnatural flow separation at the sharp corners on each end of the trunk, the ends were made of half oblate spheroids with minor radii of 2.5 inches. The 2/3 scale model surface area was then

Trunk	12.20 ft^2
Head and neck	2.30 ft^2
Legs	3.75 ft^2
Ends	3.15 ft^2
Total	21.40 ft^2

The model body sections were made of 26-gage sheet metal and assembled in the required shape (Figure 9). A heating system was enclosed in the model body. The system included a 375-cubic feet per minute squirrel-cage blower which circulated air through a plenum chamber and distribution system containing a 1000-watt strip heater connected to a variable transformer to change the internal heat. Baffles near all plenum openings permitted air flow adjustment for uniform surface temperature.

The surface or hair geometry could not easily be modeled to 2/3 scale. The literature search revealed little information regarding the influence of surface roughness on heat transfer characteristics. Preliminary tests were conducted to examine the hypothesis that the convective heat transfer from a body could be accurately predicted

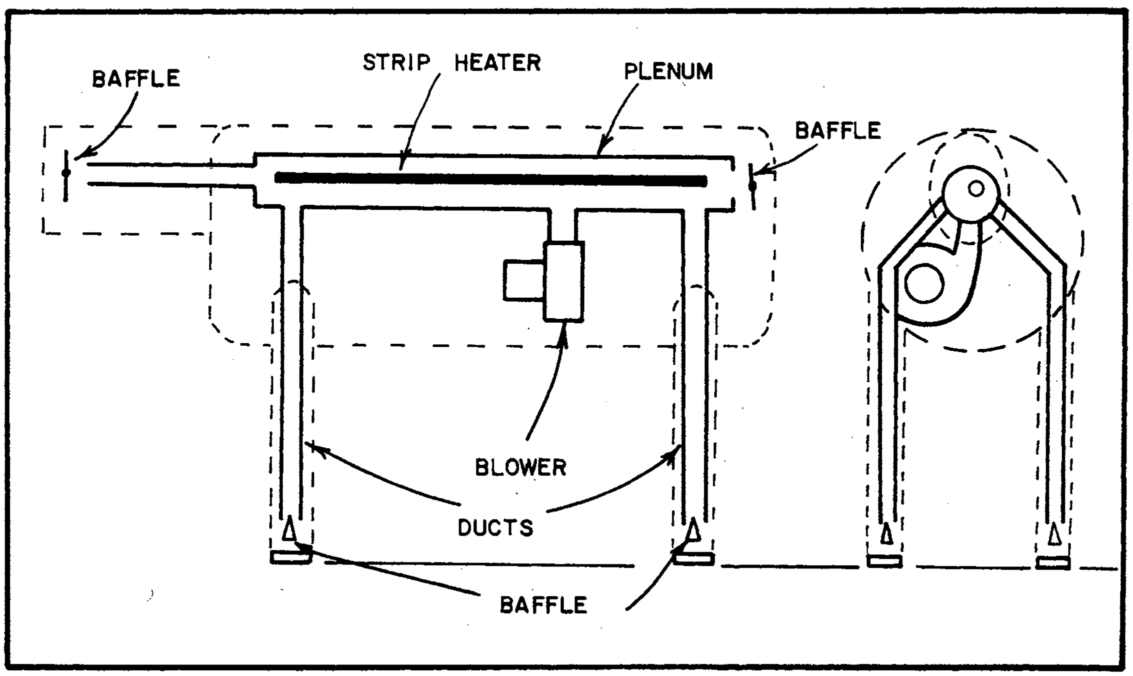


Figure 9. Model Cow With Heat Source and Internal Forced Air Heat Distribution System.

using a model with a length scale of 1.5 but full scale surface texture. In these preliminary tests, two cylinders, 1.33' and 2.0' in diameter and each 3.33' long were used. The perimeter surfaces of the two cylinders were covered with cow hide. The hide had been tanned and cured using a procedure designed to retain the hair in its natural condition.

A section of dry hide was cut to fit the cylinder and holes were drilled along the edges for stitching. The hide was then soaked to make it pliable, wrapped around the cylinder and stitched with nylon thread. As the hide dried, shrinkage pulled it into intimate contact with the metal surface. A heating unit and blower were placed in each cylinder and the ends were covered with two-inch styrofoam plugs.

An 11 by 24 feet test area with ceiling height of 8 feet was enclosed with plywood. The ends were open to allow passage of air through the tunnel shaped enclosure. A variable speed fan on one end provided forced air movement. Air velocities were measured with a vane anemometer.

A strip heater connected to a variable transformer controlled internal heat in the cylinders. The total heat energy was measured electrically. A probe similar to those described by Thompson, Worstell and Brody (75) was constructed for measuring hair surface temperatures (Figure 10). A thermocouple junction was made by butt welding 35-gage iron and constantan wires. The temperature sensing point was

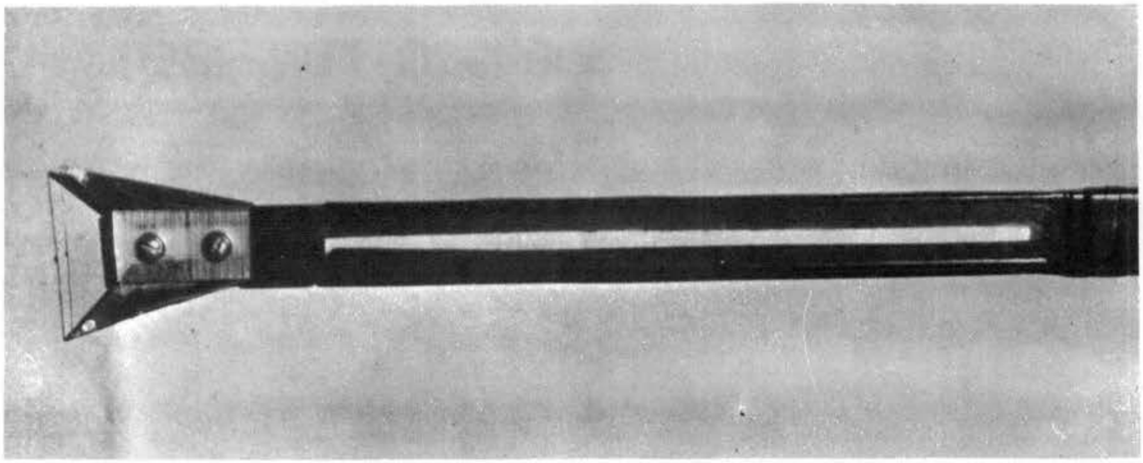


Figure 10. Hair Surface Temperature Sensing Probe.

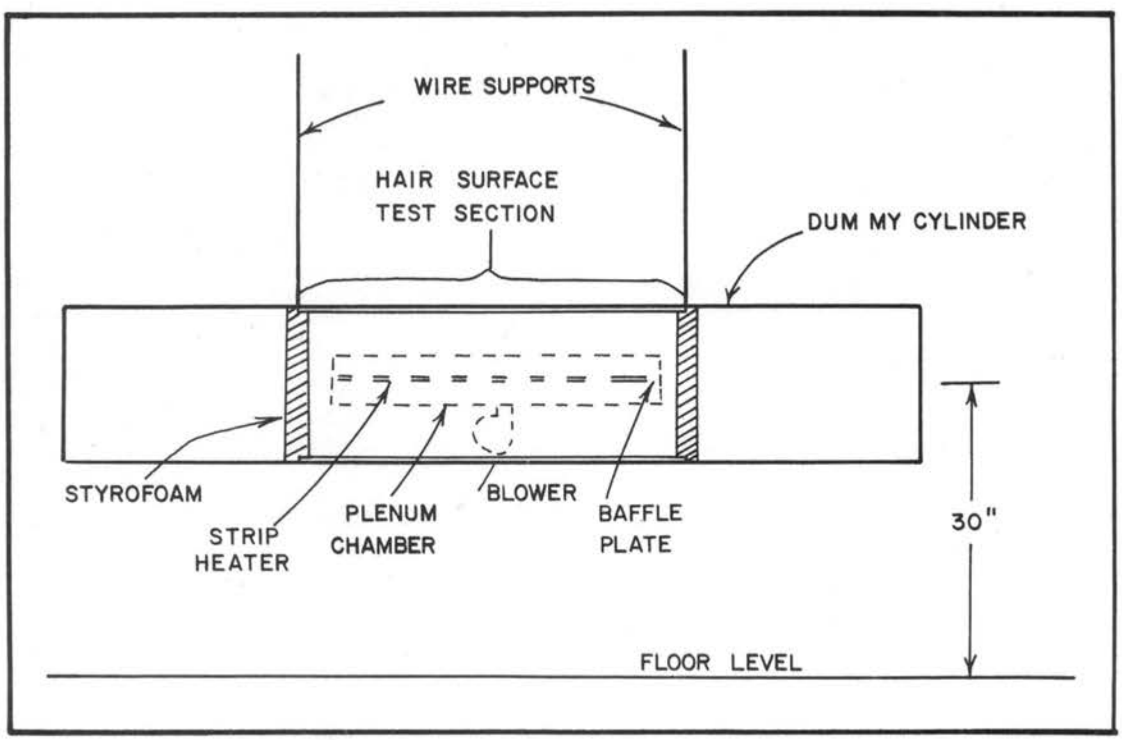


Figure 11. Test Cylinder for Determining Influence of Surface Texture Difference.

located at the weld, or bead, but in touching the hair surface, the entire length of wire between probe points was heated, thus minimizing error of conduction of heat through lead wires. Temperatures were read on a 0 to 300 F iron-constantan pyrometer.

In the preliminary test, each cylinder was suspended with its centerline 30 inches above floor surface (Figure 11). Dummy cylinders were placed on each end so that test cylinders represented sections of infinitely long cylinders in relation to air flow. Each cylinder was subjected to an experimental series wherein Re was varied by changing air velocity while holding Gr constant and then, while holding Re constant, Gr was varied by changing heating energy input to control surface to air temperature differentials. Heat input was observed to determine the average Nusselt number ($\overline{Nu}$). The results, corrected for heat loss by conduction through styrofoam ends and by radiation to the surroundings are plotted in Figure 12 with least squares fit regressions. The difference in cylinder diameter caused no significant difference in slopes of the regression lines. There was a difference in elevation between regressions with varying Re, but this resulted from the difference in values used for constant Gr. Equations for the straight lines form two pairs of component equations. Prediction equations, generated from each pair of component equations by multiplication as described by Murphy (56) were

$$Nu = 0.0631\,Re^{0.532}\,Gr^{0.137} \quad \text{for data from 2-foot cylinder}$$

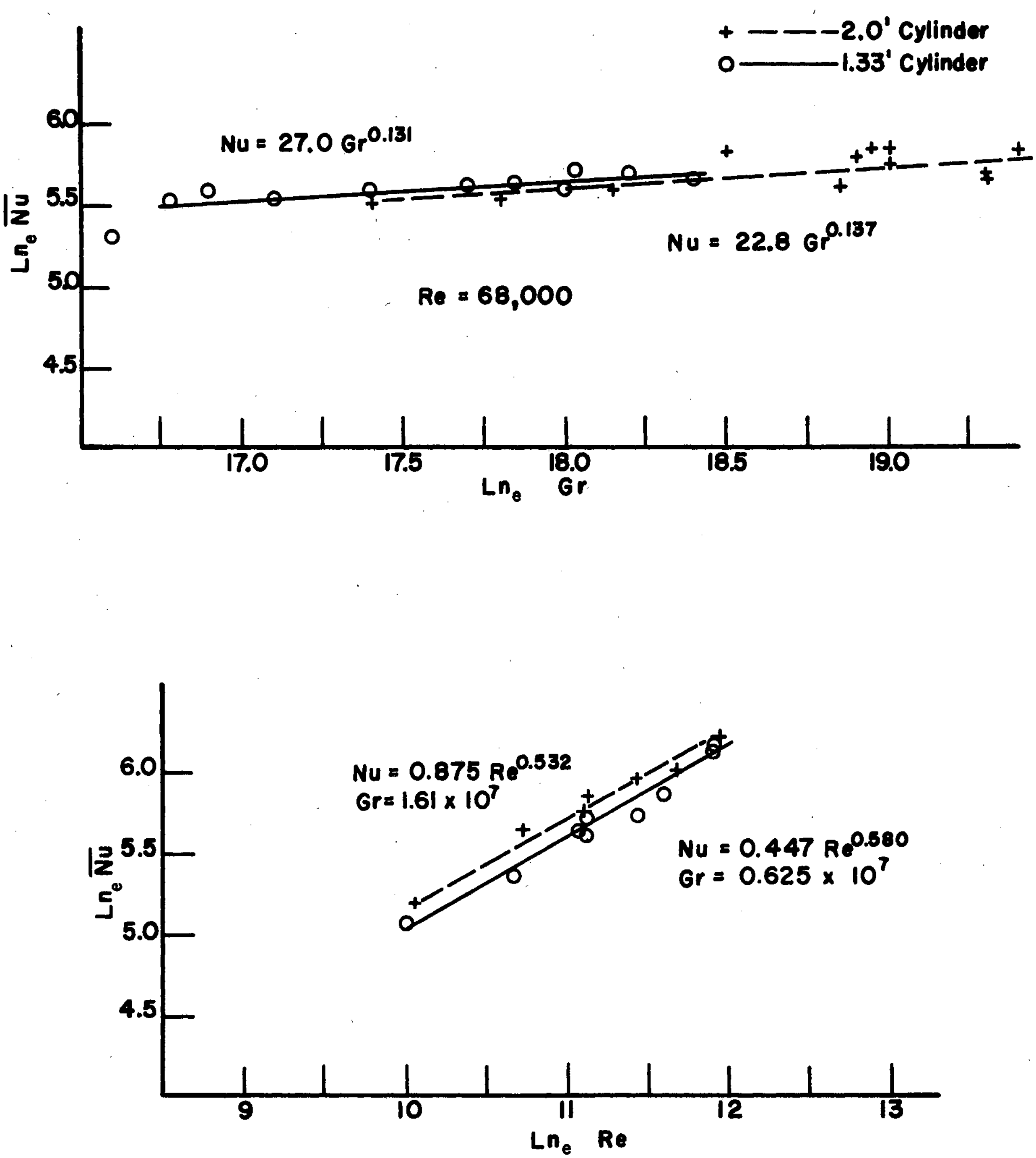


Figure 12. $\overline{\text{Nu}}$ Versus Gr and Re for 2.0' and 1.33' Cylinders Covered With Identical Cattle Hair Skins.

$$\overline{Nu} = 0.0426\ Re^{0.580}\ Gr^{0.131} \quad \text{for data from 1.33-foot cylinder}$$

Four sets of values of $\overline{Nu}$ were computed using both equations to compute values for data from each cylinder. Each set of values for $\overline{Nu}$ was then compared with the observed values. The four comparisons were plotted and analyzed using a covariance procedure for more than two sets of data. There were no significant differences among slopes or elevations.

Since the slopes in both pairs of the original component equations showed no significant difference between sets of data, as a final test the data for $\overline{Nu}$ versus Re for 1.33-foot cylinder were adjusted to correspond to a constant Gr of 1.61×10^{8} and the points for both cylinders were combined and fit with least squares regression to arrive at common slopes. The component equations are then

$$\left.\begin{aligned} \overline{Nu} &= 39.5\ Gr^{0.11} \\ \overline{Nu} &= 0.66\ Re^{0.557} \end{aligned}\right\} \quad \text{for data from 2-foot cylinder}$$

$$\left.\begin{aligned} \overline{Nu} &= 37\ Gr^{0.11} \\ \overline{Nu} &= 0.578\ Re^{0.557} \end{aligned}\right\} \quad \text{for data from 1.33-foot cylinder}$$

If the original hypothesis that scale of surface texture could be ignored is valid, each pair of component equations with common slopes should combine to form prediction equations which adequately describe heat transfer from the cylinders. Coefficients were determined from the component equations, which were combined by multiplication to form the prediction equations

$$\overline{Nu} = 0.0790\ Re^{0.557}\ Gr^{0.11} \text{ for data from 2-foot cylinder}$$

$$\overline{Nu} = 0.0778\ Re^{0.557}\ Gr^{0.11} \text{ for data from 1.33-foot cylinder}$$

Solution of these equations gives predicted values differing by only 1.5 percent.

The relative change in roughness of cattle hair surfaces, resulting from reducing the body geometry to two-thirds scale with no change in surface characteristics therefore has an insignificant influence on the heat transfer from the body. Construction of the model was completed using a length scale of 1.5 on the body and 1.0 on the surface The surface of the model was covered with Hereford hair skin using the method described for covering the cylinders.

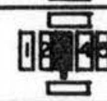
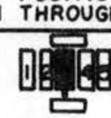
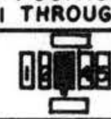
Schedule of Experiments

The procedure described by Murphy (56) was used to form component equations by observing the behaviour of the dependent parameter in response to variations in one of the independent parameters while all other parameters were held constant.

The schedule of experiments is outlined in Table V. Pi-2, Pr, is constant for air at temperature between 50 and 150 F and is omitted from the schedule. Holding Pr constant limits all prediction equations

TABLE V

SCHEDULE OF EXPERIMENTS

SERIES	NUSSELT	REYNOLDS	GRASHOF	SPACING/λ	ARRANGEMENT	FLOW
100	OBSERVE	VARY 8,000 – 150,000	0.5×10^8	∞		←
200	OBSERVE	50,000	VARY 0.2×10^8 – 1.0×10^8	∞		←
300	OBSERVE	50,000	0.5×10^8	0.125	VARY POSITION 1 THROUGH 5	←
400	OBSERVE	50,000	0.5×10^8	VARY 0 – 1.0		←
500	OBSERVE	0	VARY 0.2×10^8 – 1.0×10^8	0.125		
600	OBSERVE	0	0.5×10^8	VARY 0 – 1.0		
700	OBSERVE	VARY 8,000 – 150,000	0.5×10^8	∞		↓
800	OBSERVE	50,000	VARY 0.2×10^8 – 1.0×10^8	∞		↓
900	OBSERVE	50,000	0.5×10^8	0.125	VARY POSITION 1 THROUGH 5	↓
1000	OBSERVE	50,000	0.5×10^8	VARY 0 – 1.0		↓
1100	OBSERVE	VARY 8,000 – 150,000	0.5×10^8	∞		↑
1200	OBSERVE	50,000	VARY 0.2×10^8 – 1.0×10^8	∞		↑
1300	OBSERVE	50,000	0.5×10^8	0.125	VARY POSITION 1 THROUGH 5	↑
1400	OBSERVE	50,000	0.5×10^8	VARY 0 – 1.0		↑
1500	OBSERVE	50,000	0.5×10^8	∞		VARY 0-360°

to air as the cooling fluid, but this is not a limitation of consequence. Re is varied from 8,000 to 150,000 and Gr is varied from 0.2×10^8 to 1.0×10^8.

The equation $\overline{Nu} = f(Pr, Re, Gr)$ assumes no variation in conditions surrounding the object and no influence from direction of air flow. Cooling by inducing air movement would seldom be done for individual cows. The presence of other animals would alter air flow patterns as would the arrangement and spacing of the cows. Two additional parameters were included to investigate the importance of group patterns; distance between cattle or spacing, which can be made dimensionless by relating it to λ, and arrangement which has no dimensions. Cattle spacing index, distance between animals divided by λ, was varied from 0 to 1.0. In the cattle arrangement series, the test model was located in all positions from windward to leeward edge of the group. In Experimental Series 1500, the direction of horizontal air flow was varied relative to direction the model was facing. In all other horizontal air flow tests, the model was facing broadside to the wind.

CHAPTER VIII

EXPERIMENTAL PROCEDURE

Apparatus and Equipment

The 11 by 24-foot tunnel shaped test area used for horizontal air flow was similar to that used for the preliminary cylinder studies. A variable speed fan positioned at one end with axis horizontal generated air flow (Figure 13). The distance between model centerline and fan was twenty feet to produce uniform velocity at front, center and rear of the model. Velocity differences were observed between elevations near the floor and those at body level, but the variations were small and the physical arrangement was typical of a cattle cooling system using horizontal air flow. Air velocities were measured with a vane anemometer and checked periodically with a hot wire anemometer at the point to be occupied by the center of the model trunk. After air flow was measured, the model was positioned for test. A strobe light provided monitoring of fan rpm.

The 1000-watt strip heater in the plenum chamber of the model was connected to a variable transformer for changing energy input. For very low input, the strip heater was turned off, and the recirculating blower motor was the only source of heat. The input energy to the

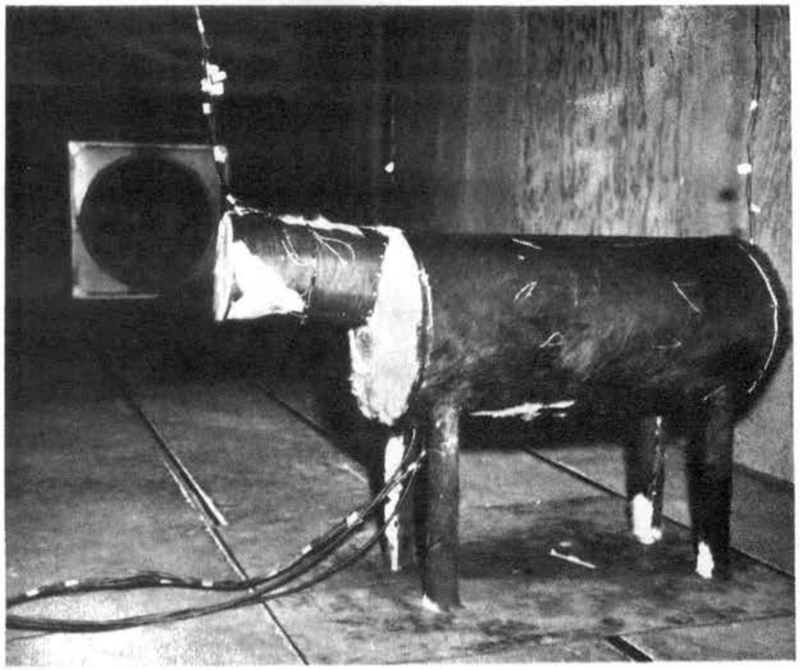


Figure 13. Test Arrangement for Horizontal Air Flow.

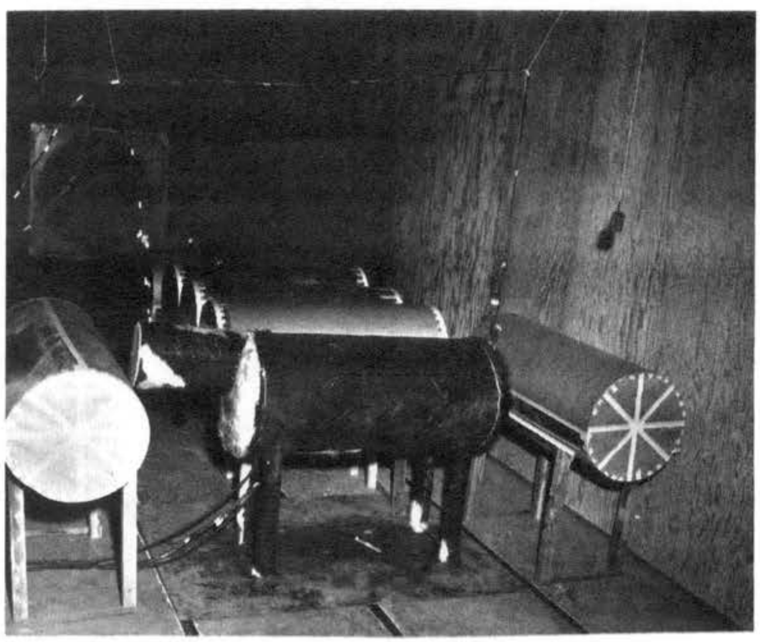


Figure 14. Test Arrangement for Group Interference Studies with Model on Leeward Side.

heater, blower and transformer was measured with a watt-hour meter. Energy loss in the transformer was subtracted from the total input energy.

For group studies, interference of adjacent animals was simulated by metal cylinders the same size and height as the model (Figure 14). Surfaces facing the model were covered with brown paper to reduce reflection.

For tests with air directed downward, the fan was mounted overhead with axis vertical 11 feet above the floor. The sidewalls of the enclosure were moved out to permit airflow in all horizontal directions. Air velocities, measured at the elevation of the centerline of the model trunk, were somewhat erratic because of interference of air bouncing from the floor, but again, are similar to those expected in typical cattle cooling systems with downward air flow.

For upward flow, the fan was set in the same overhead location but reversed to force air upward. An eight by eight-foot plywood lined enclosure, six feet high, with slotted floor was constructed (Figure 15). The enclosure was sealed with plywood between the top of the walls and the perimeter of the fan frame situated two feet higher. Part of one sidewall has hinged to provide access to the interior. With the door closed, all incoming air was drawn through the slotted floor. Figure 16 shows an animal group in position.

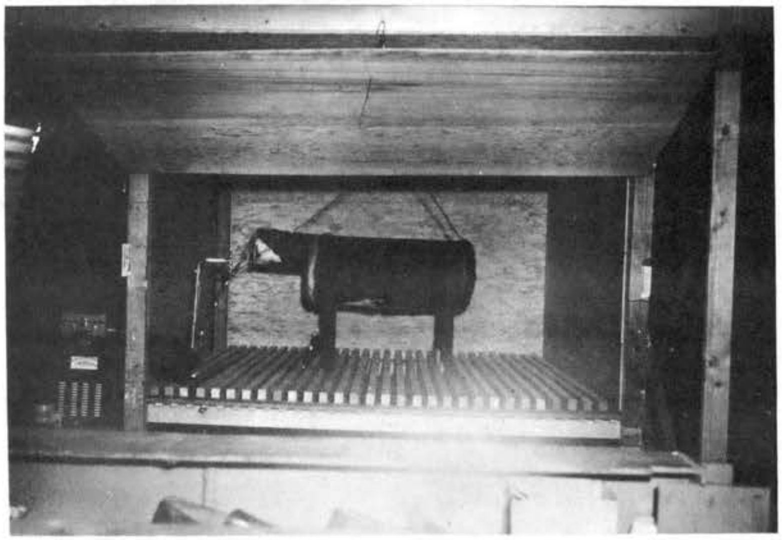


Figure 15. Slotted Floor Enclosure Used for Upward Air Flow Studies

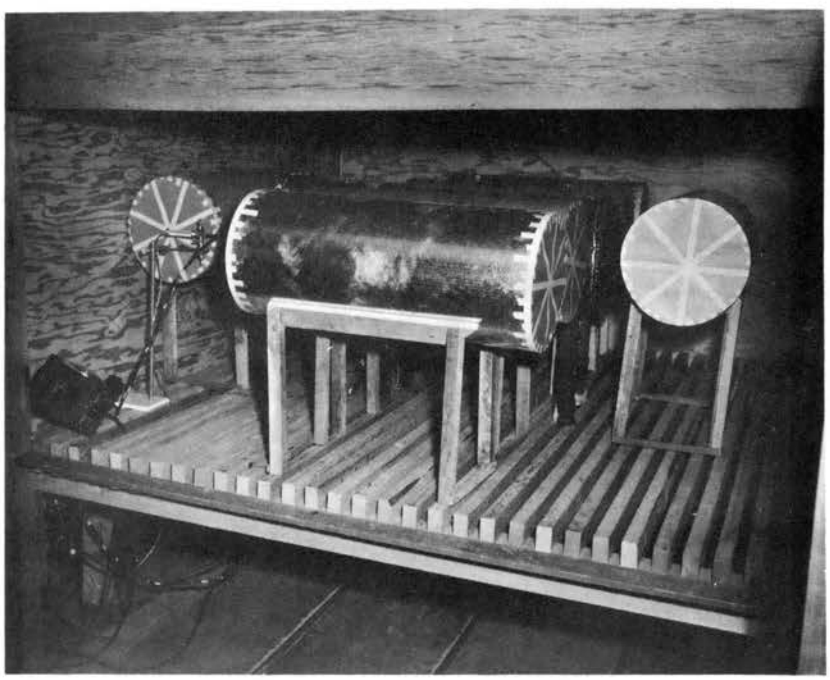


Figure 16. Animal Group in Upward Air Flow Enclosure.

Instrumentation for Temperature Measurement

The variables to be measured included temperature differential, i.e., the difference between temperature of hair surface and free air stream. In addition, mean temperature of surrounding surfaces was needed to determine heat exchange by radiation. Hair surfaces, air, and surrounding surface temperatures were the only temperature measurements used to develop the desired equations. However, measurements were also taken at 27 points on the internal metal surface and 49 points on the skin to monitor the distribution of heat and temperature.

All temperatures were sensed by 22-gage iron-constantan thermocouples connected to a 0-300 F indicating pyrometer except those of the hair and skin, where 36-gage thermocouples were used. Air temperature was measured by a radiation shielded thermocouple suspended in the free air stream on the windward side of the model. For measuring wall temperatures, thermocouples were imbedded in the surface at points surrounding the model. Internal metal surface temperatures were sensed by thermocouples soldered to the metal.

The procedure and equipment used for determining hair surface temperatures were different than used in the preliminary cylinder studies. Although satisfactory measurements could be taken with the probe described in Chapter V, there were two objections to its use. First, a measurement was subjective, i.e., dependent upon human judgment when touching the hair surface with the sensing wires. This was especially objectionable if measurements were taken by different individuals and when measuring at points physically difficult to reach. Second, the measurements required about 15 to 30 seconds at each of 98 locations.

To eliminate these objections, permanent thermocouples were installed on the hair and skin surface throughout the body surface. Hair surface thermocouples consisted of one-half inch of tightly twisted iron-constantan wire welded at the tip. The entire length rested against the outer hair surface (Figure 17). Adjacent to each hair surface thermocouple, a measurement was taken of skin temperature, sensed with a thermocouple made by converging the parallel iron-constantan wires at the tip and welding them to form a point sensor which was then inserted into the skin surface.

Temperature differential θ_w had to be characterized by a single number, requiring identification of the average hair surface temperature. This could have been accomplished by summing the temperature for the 49 points adjusted for the percentage of total area each represented. However, Bowen (13) describes a method whereby an average temperature can be obtained by connecting a group of thermocouples in a parallel circuit. When several thermocouples are connected in parallel, the electromotive force (emf) developed across their common connections represents the arithmetic mean of the emf's of the individual

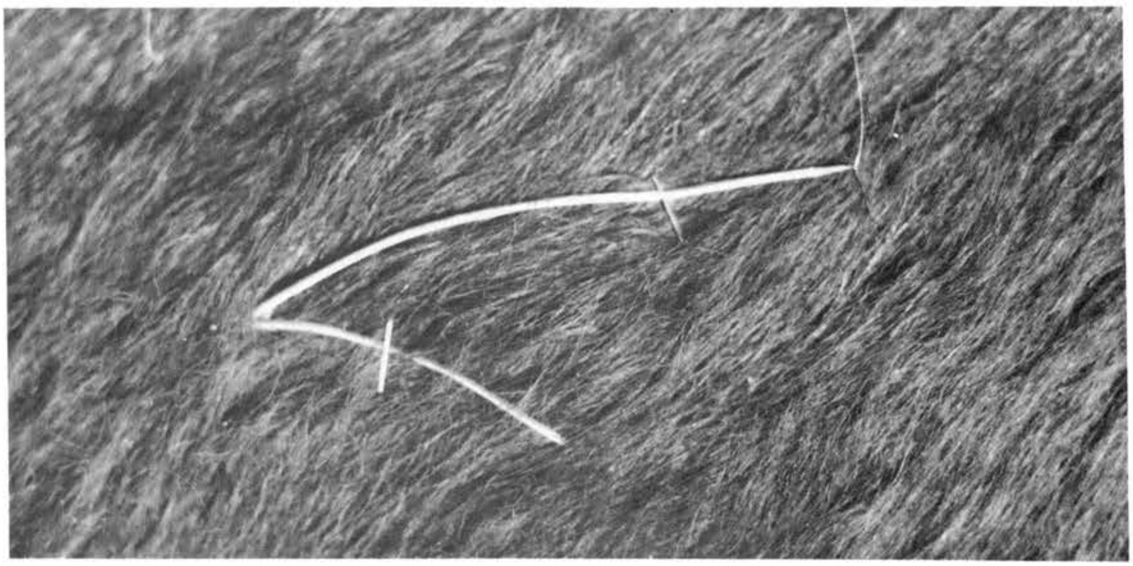


Figure 17. Thermocouples for Measuring Hair Surface Temperature (top) and Skin Temperature (bottom).

couples in the circuit. To utilize this principle for determining average temperature over a given area, each thermocouple in a circuit must represent area segments of equal size. Errors can also result from mismatch of branch resistances caused by differences in wire length. The equation for determining maximum possible error is given as

$$F = \frac{T_{max} \cdot R_{max}}{R_{min}}$$

where

F = maximum error, F

$\overline{R}$ = the arithmetic average of all branch resistances, ohms

R_{max} = the maximum difference in the resistance of any branch resistance and arithmetic average of all branch resistances, $\overline{R}$, ohms

T_{max} = the maximum difference in any individual couple temperature and the average temperature as indicated by the parallel set, F

R_{min} = the minimum resistance of any branch, ohms.

The distribution of thermocouple sites can be seen in Figure 13. Four circuits were used to determine hair surface temperatures with four corresponding circuits used for skin temperatures. The four circuits indicated average temperature for the legs, head and neck, forward trunk and rear trunk. Each circuit included nine thermocouples except the legs which had sixteen.

Since the overall surface temperature was relatively uniform, the first term in the numerator of the error equation was quite small.

However, the circuits were designed to further minimize error by proper selection of lead length. The main leads of each circuit were of 22-gage wire with resistance of 0.88 ohms per foot. The branch wires from the leads to the point of measurement were 36-gage with 14.1 ohms resistance per foot. To minimize resistance inequities within a circuit, the resistance created by additional lead length to a branch was compensated by a proportionate reduction in length of 36-gage wire within the branch. Since the ratio of resistance was 16 to 1, a one-inch reduction in 36-gage branch line nullified the resistance in 16 inches of lead wire. Thus, the second term in the numerator of the error equation was kept very small.

To further reduce maximum error, a 150-ohm resistor was installed in each circuit near the point of connection to the indicator. This increased total resistance of each branch about five fold, thus increasing the value in the denominator of the equation. The maximum possible error was verified as negligible by tests on each circuit prior to installation using points of known temperature.

The four circuits did not represent equal areas so average temperatures were determined by the equation

$$T_{ave} = \frac{T_1A_1 + T_2A_2 + T_3A_3 + T_4A_4}{A_{total}}$$

where

T = temperature $\qquad$ A = area

Subscripts indicate circuit number

With this instrumentation and thermocouple system, all required temperatures for an operating condition could be measured and recorded in about one minute, a time span which, for this study, could be considered instantaneous. The uniformity of temperature within the area measured by each averaging circuit was checked periodically with the probe.

A computer program was written to make the calculations for each test. Air properties such as conductivity, density, viscosity and coefficient of thermal expansion were evaluated for each test run by equations in the program relating a tabulated value to existing film temperature (arithmetic mean of surface and air temperature).

Correction for Radiant Heat Exchange

The energy input determined from the watt-hour meter less consumption by the transformer was heat dissipated by the model. Since losses by conduction were essentially zero, heat was dissipated by non-evaporative convection and by radiation. Quantitative separation was achieved by calculating the net radiant loss and subtracting this amount from the total. The equations used for calculation of radiant loss were

$$E_{b1} = 0.1713 \times 10^{-8} (T_1 + 460)^4$$

$$E_{b2} = 0.1713 \times 10^{-8} (T_2 + 460)^4$$

$$J_1 = \epsilon_1 E_{b1} + (1 - \epsilon_1) (J_2 F_{12} + J_1 F_{11}$$

$$J_2 = \epsilon_2 E_{b2} + (1 - \epsilon_2)\ (J_1 F_{21} + J_2 F_{22})$$

$$Q1 = [(J_1 - J_2) F_{12} + \epsilon_1 F_{13} (E_{b1} - E_{b2})] A_1$$

where

Subscript 1 refers to model

Subscript 2 refers to surrounding walls and floor

Subscript 3 refers to openings in wall area from which radiation to model was considered zero.

E_b = total emissive power of black body, BTU/hr-ft^2

T = temperature, F

J = total radiosity, BTU/hr-ft^2

ϵ = emissivity, dimensionless

F = shape factor (F_{12} indicates percentage of radiosity from body 1 which impinges on body 2), dimensionless

Q_1 = net radiant heat flow from model, BTU/hr

The solution of these equations required values for surface emissivity and shape factors in addition to data collected. An emissivity (1.0 - reflectance) was selected for an appropriate wavelength from curves in Figure 4 of Chapter IV, which give monochromatic values for reflectance of various hair surfaces. Wein's Displacement Law states that maximum radiation intensity for a given temperature occurs at the wave band where temperature times wavelength equals a constant, 5216.6 R-microns. Thus, for the range of hair surface temperatures observed during the tests, most of the radiation emitted from the hair surface was in the wavelength range from 850 to 900 millimicrons. Therefore, the value selected for emissivity of the Hereford hair surface was 0.72. The emissivity value used for new unpainted plywood was 0.9. (32).

Shape factor determinations were based on the method described by Perry and Speck (60), cited in Chapter IV. Using spheres to represent the animal and subdividing the surrounding walls into various rectangles, curves developed by Hamilton and Morgan (34) were used to determine the required shape factors for all wall arrangements used. Shape factors used are listed in Table VI.

TABLE VI

SHAPE FACTORS FOR CALCULATION OF RADIANT HEAT EXCHANGE FROM MODEL TO SURROUNDING SURFACES

Shape Factors	Horizontal Flow	Free Flow	Downward Flow	Upward Flow
F_{11}	0	0	0	0
F_{12}	0.923	0.870	0.879	1.000
F_{21}	0.026	0.050	0.050	0.110
F_{22}	0.790	0.770	0.710	0.890
F_{13}	0.077	0.130	0.121	0.000

CHAPTER VIII

DATA ANALYSIS

Formation of Prediction Equations

Component equations were developed with the data collected from Experimental Series 100, 200, 500, 700, 800, 1100, and 1200, relating air flow to energy transfer from a single animal. The observed average Nusselt number was plotted against two independent parameters, Re and Gr, on both logarithmic and arithmetic scales. Linear, quadratic and exponential curves representing the best fit were then determined using the method of least squares. Equations representing the curves and respective correlation indices (R^2) are listed in Table VII.

The selection of component equations to combine for prediction equations was based on comparison of correlation indices and on the behaviour of the equations, particularly as abscissa values approach zero and infinity. Simplicity of final equations was also considered.

The correlation indices of the curves relating Nu to Re are the highest for the quadratic equation and, except for Series 1100, are the lowest for equations representing a straight line in logarithmic space. The lower logarithmic correlation, based on deviations of the logarithmic values, reflects the slight increase in slope with increasing

TABLE VII

COMPONENT EQUATIONS AND CORRELATION INDICES FOR HORIZONTAL, UPWARD, DOWNWARD, AND FREE AIR FLOW

HORIZONTAL FLOW	Series 100 ($\overline{Nu}$ versus Re)	
Linear	$\overline{Nu} = 70 + 0.00239\ Re$	R^2=0.972
Quadratic	$\overline{Nu} = 59.5 + 2.92 \times 10^{-3}\ Re - 0.00425 \times 10^{-6}\ Re^2$	R^2=0.978*
Exponential	$\overline{Nu} = 0.59\ Re^{0.537}$	R^2=0.956
	Series 200 ($\overline{Nu}$ versus Gr)	
Linear	$\overline{Nu} = 140 + 3.08 \times 10^{-7}\ Gr$	R^2=0.285
Quadratic	$\overline{Nu} = 115 + 15.46 \times 10^{-7}\ Gr - 1.24 \times 10^{-14}\ Gr^2$	R^2=0.360
Exponential	$\overline{Nu} = 28.68\ Gr^{0.096}$	R^2=0.360*
DOWNWARD FLOW	Series 700 ($\overline{Nu}$ versus Re)	
Linear	$\overline{Nu} = 88 + 2.09 \times 10^{-3}\ Re$	R^2=0.964
Quadratic	$\overline{Nu} = 83 + 2.28 \times 10^{-3}\ Re + 0.00124 \times 10^{-6}\ Re^2$	R^2=0.964*
Exponential	$\overline{Nu} = 0.691\ Re^{0.524}$	R^2=0.914
	Series 800 ($\overline{Nu}$ versus Gr)	
Linear	$\overline{Nu} = 249 - 4.87 \times 10^{-7}\ Gr$	R^2=0.407
Quadratic	$\overline{Nu} = 253 - 7.30 \times 10^{-7}\ Gr + 0.260 \times 10^{-14}\ Gr^2$	R^2=0.418
Exponential	$\overline{Nu} = 710\ Gr^{-0.065}$	R^2=0.420*
UPWARD FLOW	Series 1100 ($\overline{Nu}$ versus Re)	
Linear	$\overline{Nu} = 72 + 2.38 \times 10^{-3}\ Re$	R^2=0.970
Quadratic	$\overline{Nu} = 56 + 3.65 \times 10^{-3}\ Re - 0.014 \times 10^{-6}\ Re^2$	R^2=0.990*
Exponential	$\overline{Nu} = 0.621\ Re^{0.535}$	R^2=0.990
	Series 1200 ($\overline{Nu}$ versus Gr)	
Linear	$\overline{Nu} = 178 + 6.69 \times 10^{-7}\ Gr$	R^2=0.664
Quadratic	$\overline{Nu} = 172 + 10.02 \times 10^{-7}\ Gr - 0.36 \times 10^{-14}\ Gr^2$	R^2=0.670*
Exponential	$\overline{Nu} = 36.0\ Gr^{0.1}$	R^2=0.588
FREE CONVECTION	Series 500 ($\overline{Nu}$ versus Gr)	
Linear	$\overline{Nu} = 67.4 + 1.78 \times 10^{-7}\ Gr$	R^2=0.480
Quadratic	$\overline{Nu} = 58.5 + 5.27 \times 10^{-7}\ Gr - 0.268 \times 10^{-14}\ Gr^2$	R^2=0.600
Exponential	$\overline{Nu} = 6.85\ Gr^{0.137}$	R^2=0.637*

*Highest correlation for the three curves.

Re in logarithmic space, a phenomenon characteristic of forced convection heat transfer from cylinders in cross-flow. However, for the relatively narrow range of Re used in the experiments, all curves tested for $\overline{Nu}$ versus Re have high correlation, so correlation differences for these curves were considered of secondary importance in the selection of the most representative form of component equation.

In $\overline{Nu}$ versus Gr relationships, the quadratic curve has the highest correlation in two cases and the logarithmic relationship has the highest in the other two. The exponential equation was selected as the best component relationship on the basis of the unrealistic behaviour of the quadratic curve at high values of Gr. Because the Gr^2 term has a negative coefficient, the quadratic curves in Series 100, 500 and 1200 would reach a maximum followed by a sharp drop to a negative value. Since $\overline{Nu}$ would never decrease with increasing Gr except under conditions of opposing flow, the behaviour is better described by the exponential equation.

A relatively simple prediction equation would result from combining pairs of exponential component equations by multiplication. The prediction equations would then be of the form

$$\overline{Nu} = C_1 \, Re^a \, Gr^b$$

However, this equation would misrepresent the behaviour of $\overline{Nu}$ as the value of Re approaches zero. For a zero value of Re, the equation predicts a zero value of $\overline{Nu}$. This is contrary to reality wherein $\overline{Nu}$, as

Re approaches zero, will approach a value representing free convection.

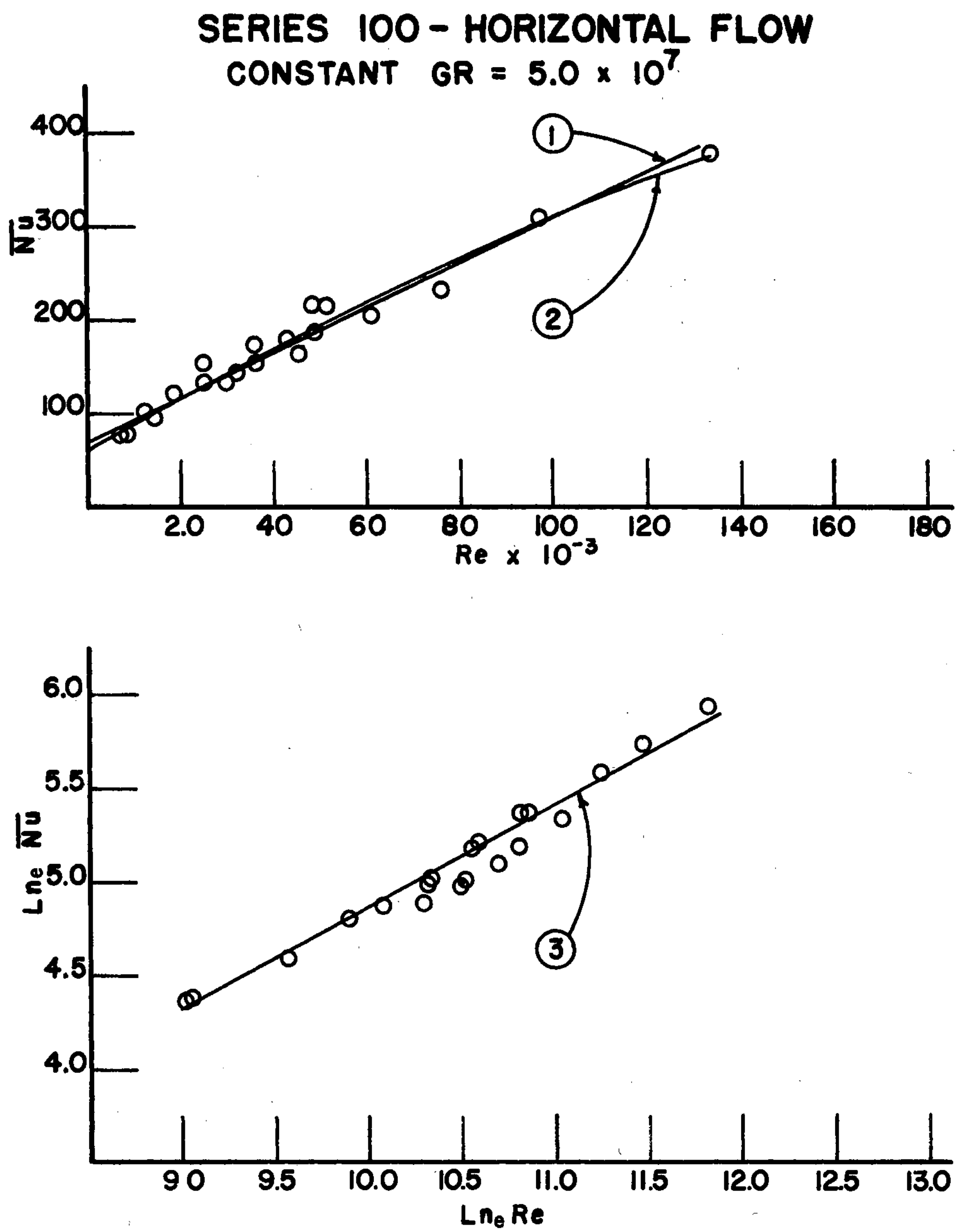
Since the behaviour at low Re cannot be predicted with an equation formed by multiplication of exponential component equations, the most simple equation can be formed by addition. Because both sets of data plot as straight lines with slopes other than zero on log paper, the component equations cannot be combined by addition (56). Therefore, rejecting the exponential $\overline{Nu}$ versus Re equation and considering the small difference in correlation indices between linear and quadratic curves, the simpler linear equation was selected as the most appropriate component equation. The equation now includes the conditions causing the slight curvature on log paper and can be combined by addition resulting in a prediction equation of the form

$$\overline{Nu} = C_1 + C_2 \, Re + C_3 \, Gr^a \qquad Pr = 0.72$$

This equation yields realistic predictions of $\overline{Nu}$ for all values of Re and Gr within and near the range of experimental data.

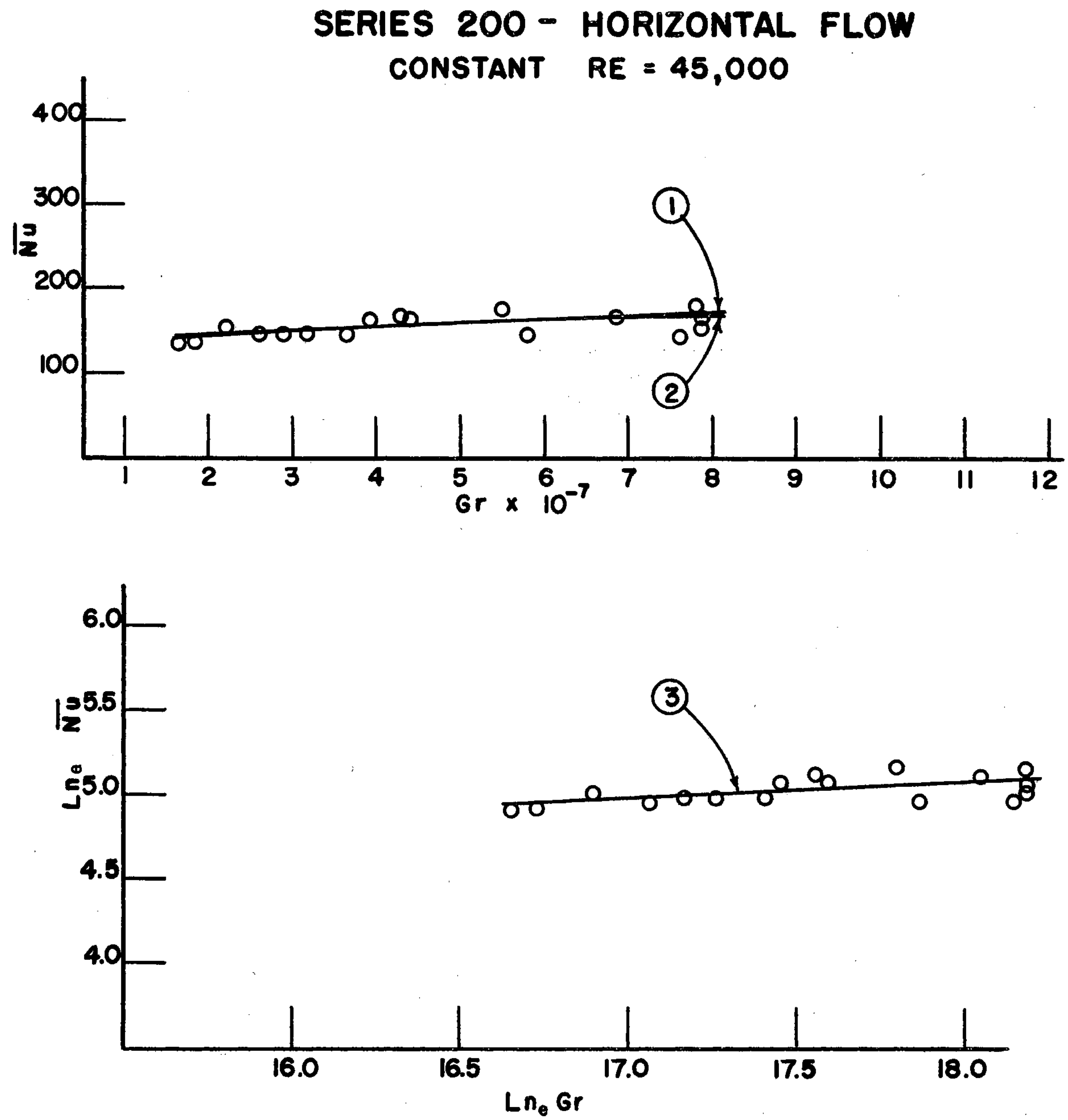
Single Animal in Horizontal Flow

The data for a single animal subjected to horizontal air flow were collected in Experimental Series 100 and 200 and are plotted in Figures 18 and 19. The linear, quadratic and exponential fitted curves are shown. As was expected, the influence of variation of Gr is slight but significantly different from zero as verified by statistical test



(1)	$\overline{Nu} = 70 + 0.00239\ Re$	$R^2 = 0.972$
(2)	$\overline{Nu} = 59 + 0.00292\ Re - 0.00425 \times 10^{-6}\ Re^2$	$R^2 = 0.978$
(3)	$\overline{Nu} = 0.59\ Re^{0.537}$	$R^2 = 0.956$

Figure 18. $\overline{Nu}$ Versus Re and Corresponding Component Equations for Experimental Series 100.



(1) $\overline{Nu} = 140 + 3.08 \times 10^{-7}\, Gr$ $\quad R^2 = 0.285$

(2) $\overline{Nu} = 115 + 15.46 \times 10^{-7}\, Gr - 1.24 \times 10^{-14}\, Gr$ $\quad R^2 = 0.408$

(3) $\overline{Nu} = 28.68\, Gr^{0.096}$ $\quad R^2 = 0.360$

Figure 19. $\overline{Nu}$ Versus Gr and Corresponding Component Equations for Experimental Series 200.

using cumulative T-distribution values at the 95 percent level. Combining the selected component equations (linear for $\overline{Nu}$ versus Re and exponential for Nu versus Gr as discussed above) for horizontal air flow forms the prediction equation

$$\overline{Nu} = 70 + 0.00239\ Re + 28.7\ Gr^{0.096} + C \qquad Pr = 0.72$$

The value for C was determined for each data point by substituting appropriate values in the equation

$$C = \overline{Nu} - 70 - 0.00239\ Re - 28.7\ Gr^{0.096}$$

where $\overline{Nu}$ is the value observed during both test series for recorded values of Re and Gr. The numerical mean value for all tests in Series 100 and the numerical mean value for all tests in Series 200 were then averaged to determine the value of C. Combining the value of C with the existing constant 70, resulted in the horizontal flow equation

$$\overline{Nu} = 0.00239\ Re + 28.7\ Gr^{0.096} - 90 \qquad Pr = 0.72$$

Using the prediction equation, a value of $\overline{Nu}$ was calculated for each set of conditions in the test series. This value was then plotted against the observed value for the corresponding conditions to determine if adjustments in the equation would improve its representation of test results. The comparison is plotted in Figure 20 with a straight line representing best least squares fit of data points and a line with 45-degree slope representing $\overline{Nu}$-observed equals $\overline{Nu}$-predicted.

Since the two lines are nearly identical in slope and elevation, no adjustments were made in the equation.

The data in Series 100 and 200 resulting from the study of horizontal flow effects were obtained with the model facing broadside to the air stream direction. Series 1500 was conducted to determine any heat transfer differences that might occur with change in animal orientation relative to direction of air movement. The results are plotted in Figure 21 which shows insignificant differences in convective cooling with change in animal orientation.

Single Animal in Downward Flow

Figures 22 and 23 show the heat transfer relationships for a single animal with downward air flow. The negative slope of the curves relating $\overline{Nu}$ to Gr indicate a slight suppression of heat transfer caused by bouyant forces when in direct opposition to forced fluid flow. The difference in slope from zero was barely significant at the 95 percent level. Combining the component equations by addition and solving for the constant by relating to observed values of $\overline{Nu}$ yielded the equation

$$\overline{Nu} = 0.00209\ Re + 710\ Gr^{-0.065} - 139 \qquad Pr = 0.72$$

Comparison of the line representing the least squares fit of $\overline{Nu}$-observed versus $\overline{Nu}$-predicted with the line representing $\overline{Nu}$-observed equals $\overline{Nu}$-predicted again revealed little possibility for improvement by adjustment of any factors in the equation.

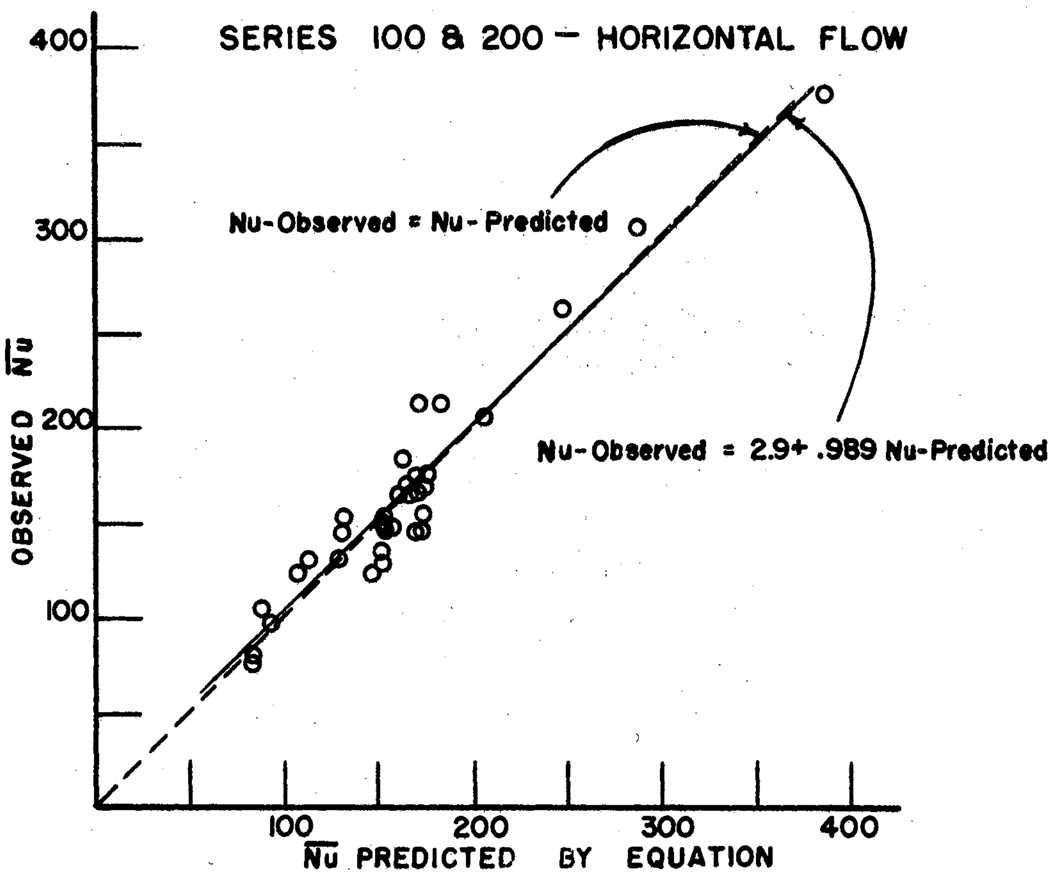


Figure 20. Comparison of Computed Versus Observed Values of Nu in Horizontal Flow.

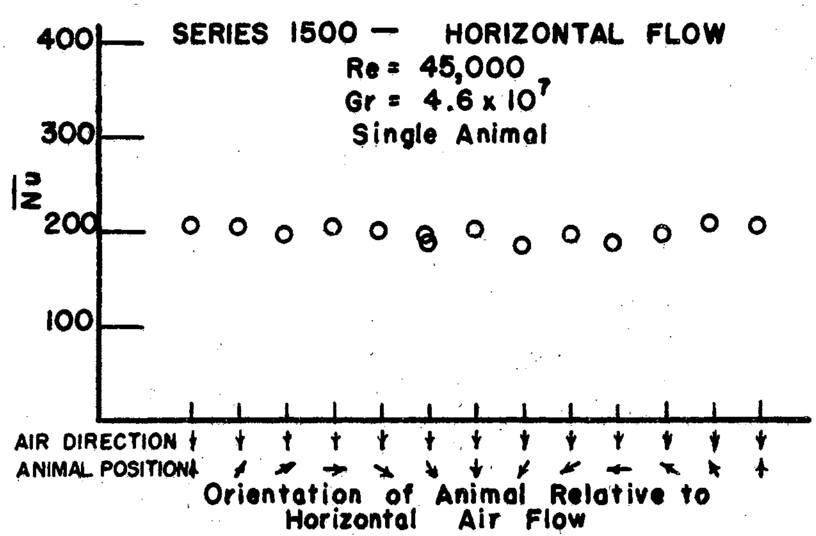
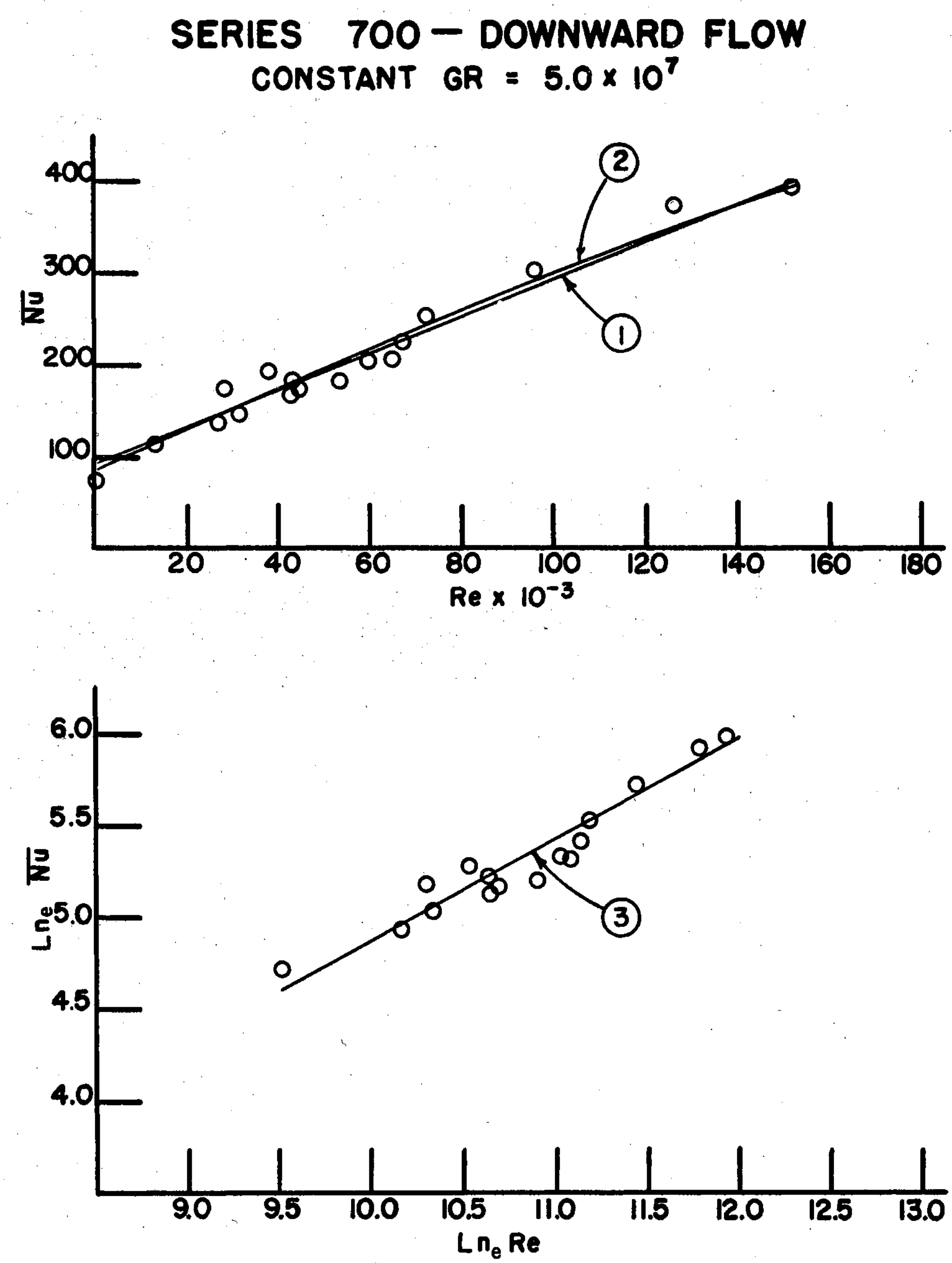


Figure 21. Heat Transfer as Affected by Orientation of Animal Relative to Direction of Horizontal Flow.

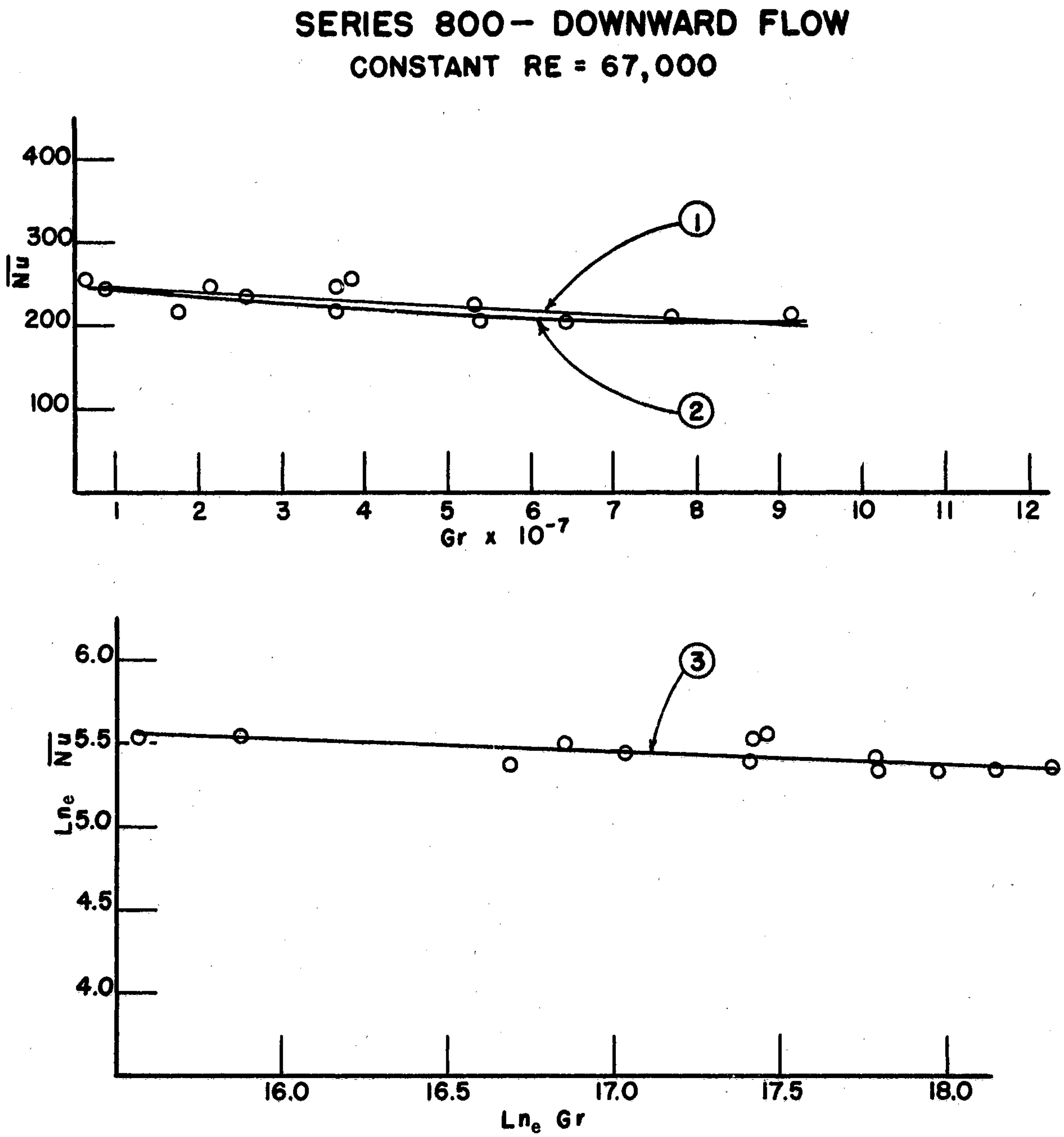


(1) $Nu = 88 + 0.00209\ Re$ $\qquad R^2 = 0.964$

(2) $Nu = 83 + 0.0027\ Re - 0.00124 \times 10^{-6}\ Re^2$ $\qquad R^2 = 0.964$

(3) $Nu = 0.691\ Re^{0.524}$ $\qquad R^2 = 0.914$

Figure 22. $\overline{Nu}$ Versus Re and Corresponding Component Equations for Experimental Series 700.



(1) $\overline{Nu} = 249 - 4.87 \times 10^{-7}\ Gr$ $\qquad R^2 = 0.407$

(2) $\overline{Nu} = 253 - 7.3 \times 10^{-7}\ Gr + 0.26 \times 10^{-14}\ Gr^2$ $\qquad R^2 = 0.418$

(3) $\overline{Nu} = 710\ Gr^{-0.065}$ $\qquad R^2 = 0.420$

Figure 23. $\overline{Nu}$ Versus Gr and Corresponding Component Equations for Experimental Series 800.

Single Animal in Upward Flow

Convection heat transfer relationships from the model in air flowing vertically upward were investigated in Experimental Series 1100 and 1200 and results are shown in Figures 24 and 25. The relationships and component equations are similar to those for horizontal flow. The slope of the line on logarithmic coordinates for Series 1200 is significantly different from zero but not different from the corresponding slopes for horizontal flow. The prediction equation for $\overline{Nu}$ when air is flowing upward, determined by the same method as described for horizontal and downward flow, is

$$\overline{Nu} = 0.00238\ Re + 36\ Gr^{0.1} - 135 \qquad Pr = 0.72$$

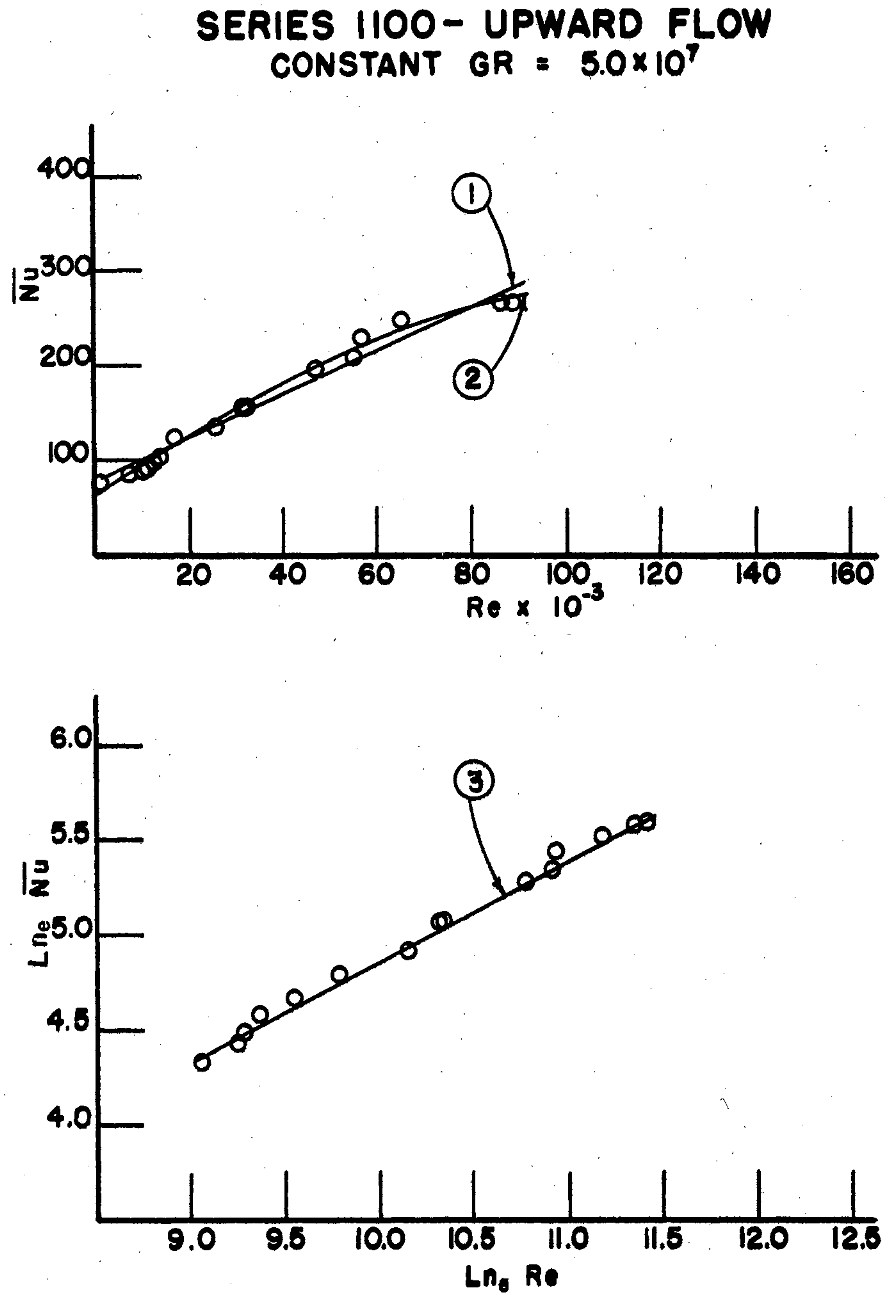
The observed values were again compared statistically to predicted values and the prediction equation needed no adjustment to accurately represent experimental data.

Free Convection

Free convection relationships were investigated in Experimental Series 500, the data for which are plotted in Figure 26. Since Re equals zero and is not involved in this series of experiments, the component equation is also the final prediction equation. The equation is

$$\overline{Nu} = 6.85\ Gr^{0.137} \qquad Pr = 0.72$$

Similar predictions are made by the equation for upward flow with Re equal to zero, resulting in the equation

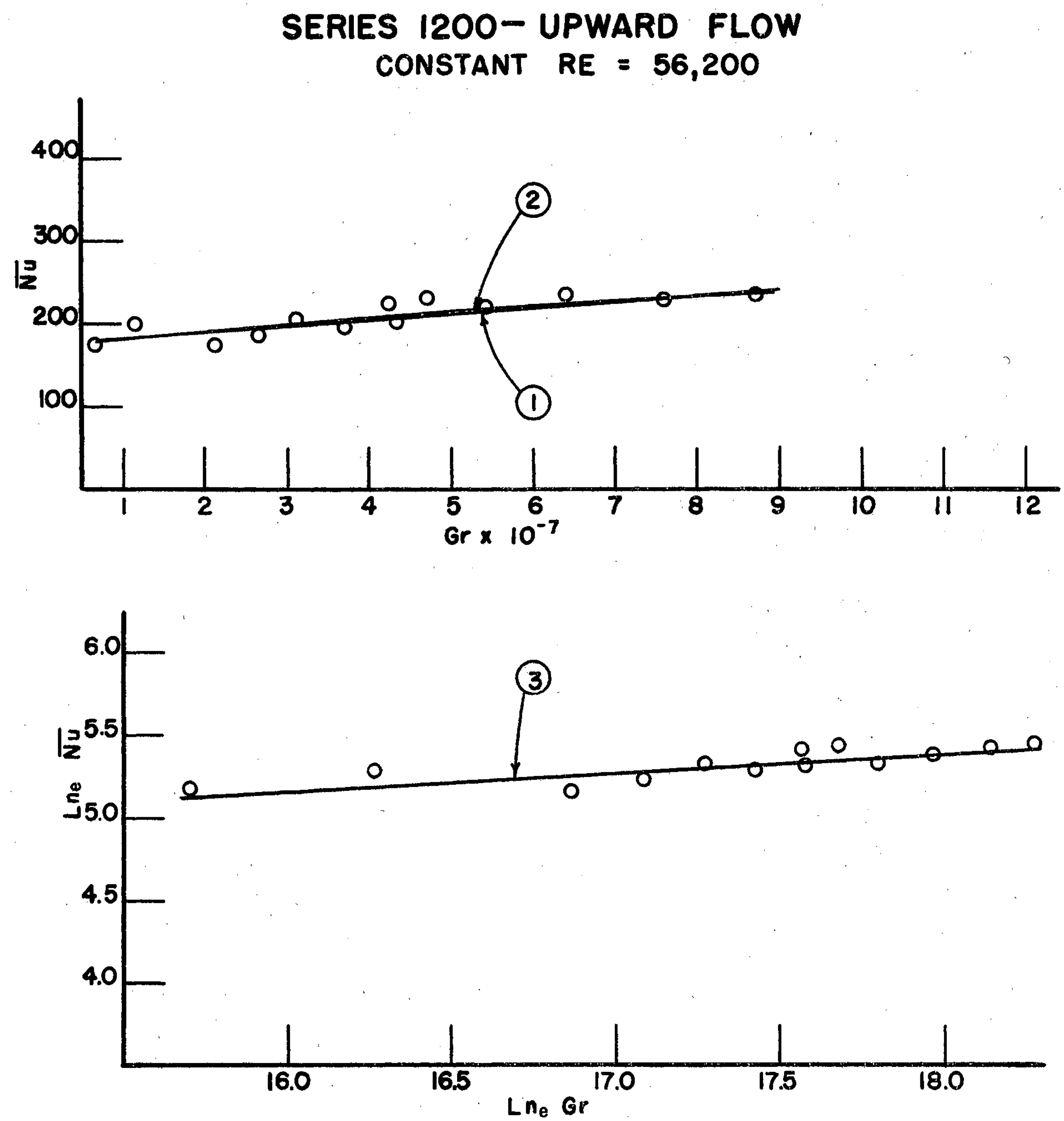


(1) $\overline{Nu} = 72 + 0.00238\ Re$ $\qquad R^2 = 0.970$

(2) $\overline{Nu} = 56 + 0.00365\ Re - 0.014 \times 10^{-6} Re^2$ $\qquad R^2 = 0.990$

(3) $\overline{Nu} = 0.621\ Re^{0.536}$ $\qquad R^2 = 0.990$

Figure 24. $\overline{Nu}$ Versus Re and Corresponding Component Equations for Experimental Series 1100.

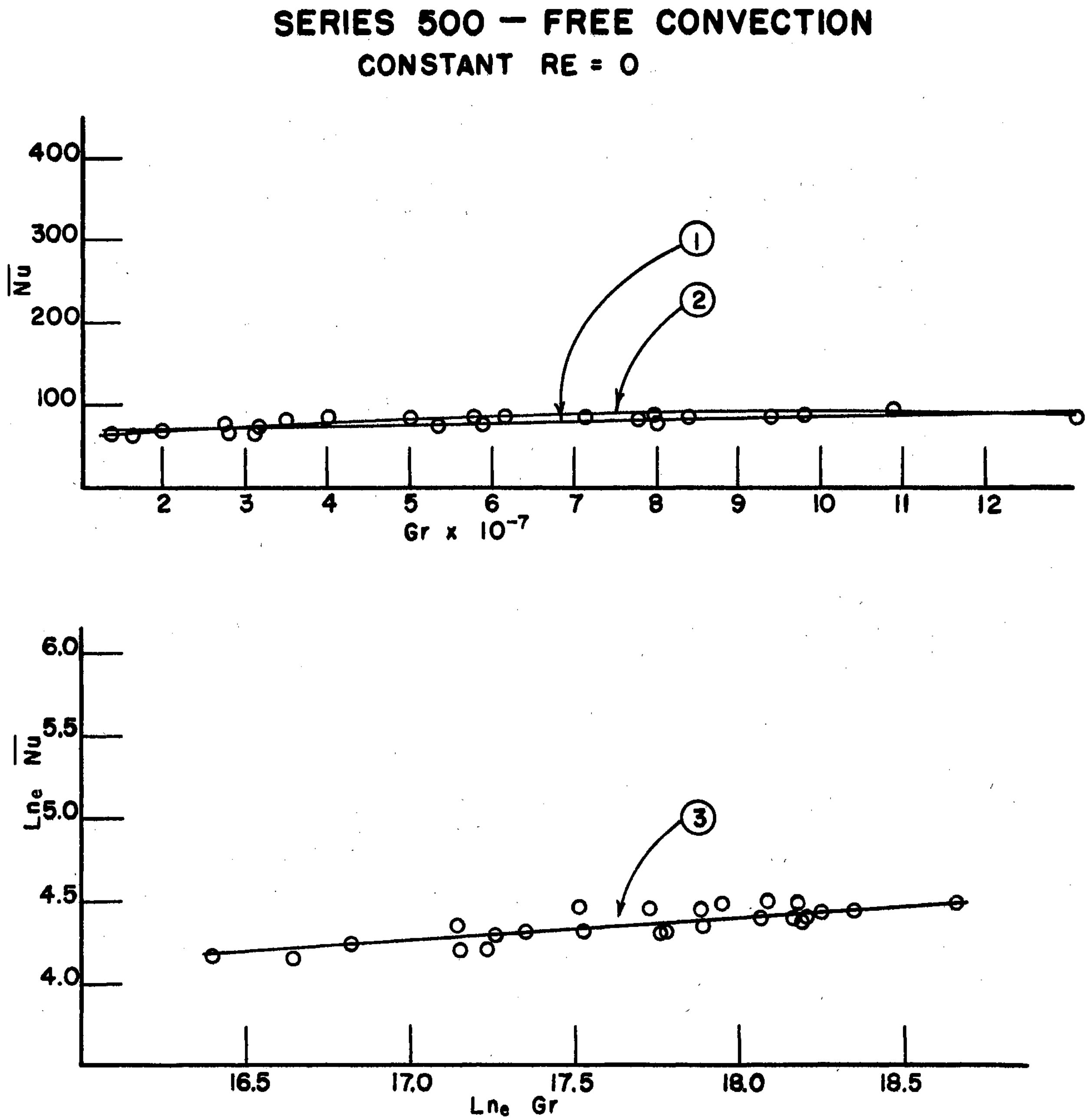


(1) $\overline{Nu} = 178 + 6.69 \times 10^{-7}$ Gr $R^2 = 0.664$

(2) $\overline{Nu} = 172 + 10.02 \times 10^{-7}$ Gr $- 0.36 \times 10^{-14}$ Gr $R^2 = 0.670$

(3) $\overline{Nu} = 36.0\ Gr^{0.10}$ $R^2 = 0.588$

Figure 25. $\overline{Nu}$ Versus Gr and Corresponding Component Equations for Experimental Series 1200.



① $\overline{Nu} = 67 + 1.78 \times 10^{-7}\ Gr$ $\qquad R^2 = 0.480$

② $\overline{Nu} = 59 + 5.3 \times 10^{-7}\ Gr - 0.268 \times 10^{-14}\ Gr$ $\qquad R^2 = 0.600$

③ $\overline{Nu} = 6.85\ Gr^{0.137}$ $\qquad R^2 = 0.637$

Figure 26. $\overline{Nu}$ Versus Gr and Corresponding Component Equations for Experimental Series 500.

$$\overline{Nu} = 36\ Gr^{0.1} - 135 \qquad Pr = 0.72$$

Effect of Presence of Other Animals

The influence on convective heat transfer of the presence of other animals and their interference with air currents was investigated for the three directions of free air stream. The effect of group density, or amount of space between animals, was studied as well as the relation between cooling effect and location within the group. The influence of density was studied for free convection also.

Spacing Between Animals

The relationship between $\overline{Nu}$ and group density was investigated in Experimental Series 400, 600, 1000 and 1400. The plotted data for mixed and free convection are shown in Figures 27 through 30. The general pattern of the relationship is similar for all directions of air flow. Cooling is suppressed by about 20% when the animals are in contact with one another and by lesser amounts when free space between animals is very small. For distances equivalent to about one or two inches between mature cattle, the convective cooling is as effective as for all lesser densities. No equations or curves were developed for the relationships.

The reduced cooling under extreme crowding is not present in the data representing free convection (Figure 30). A possible explanation for this lack of cooling suppression is that sufficient passage may

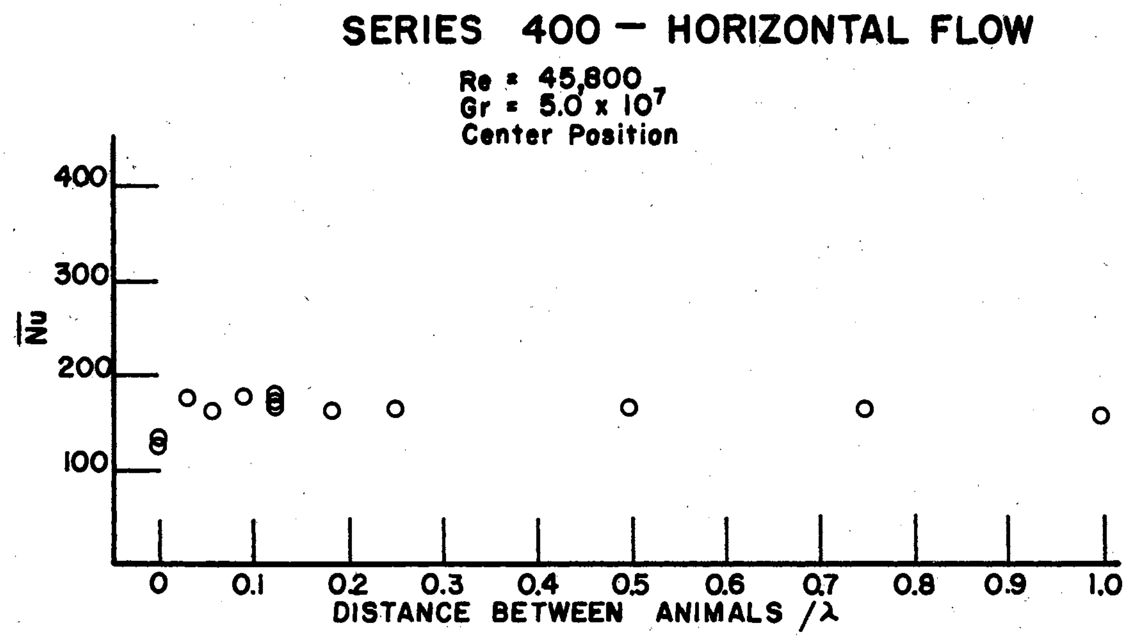


Figure 27. Effect of Group Density on Convection Heat Transfer in Horizontal Flow.

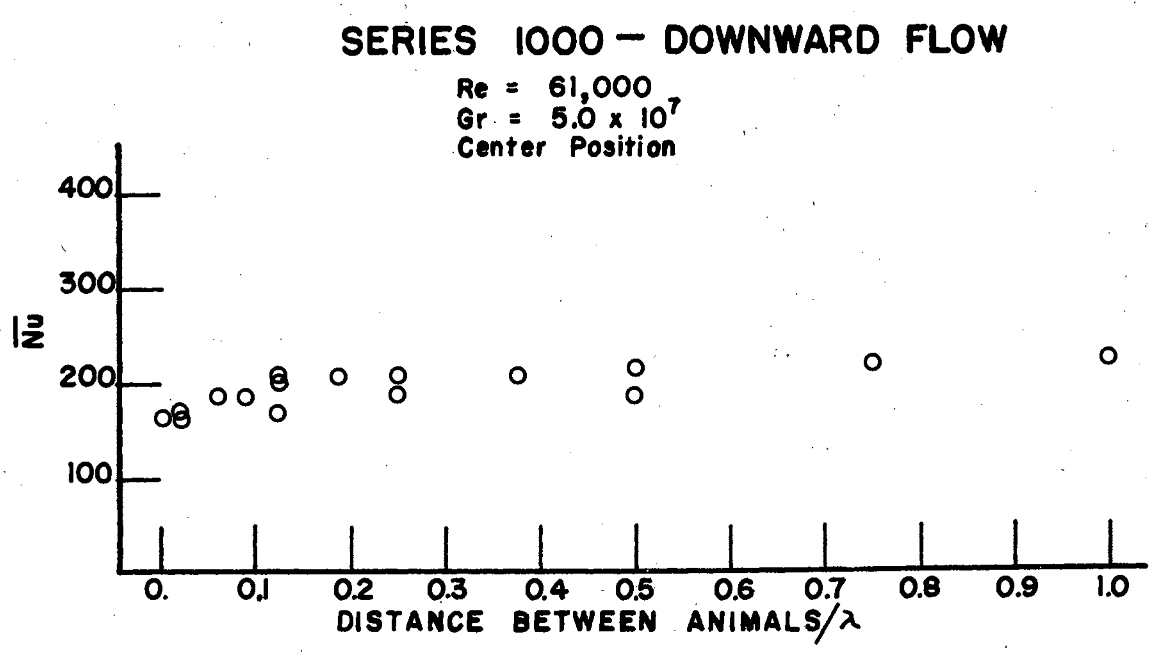


Figure 28. Effect of Group Density on Convection Heat Transfer in Downward Flow.

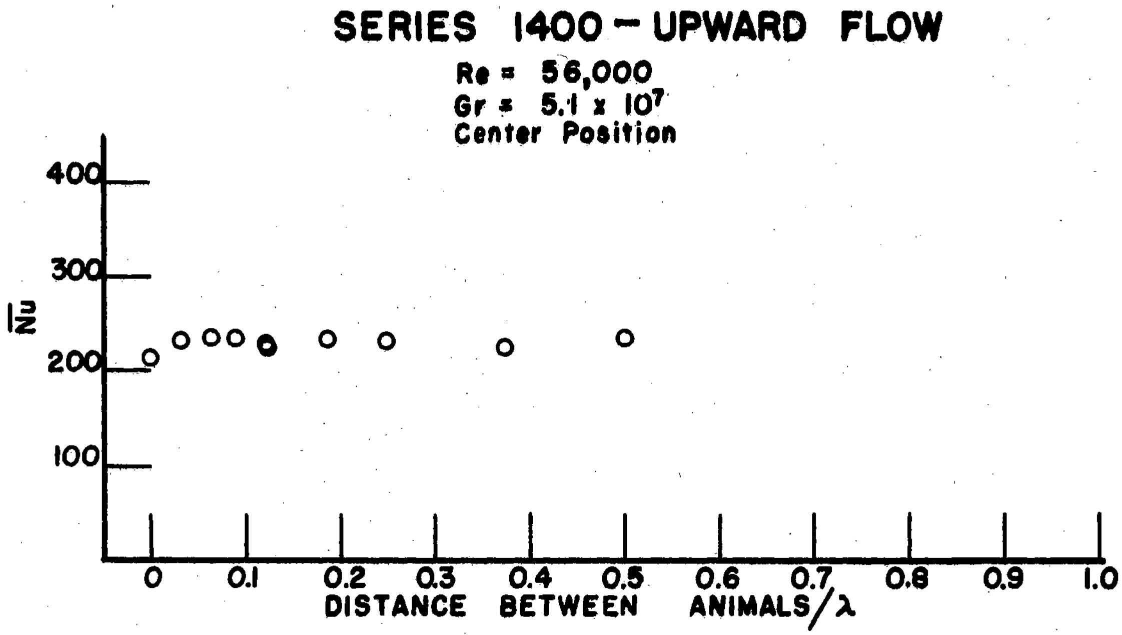


Figure 29. Effect of Group Density on Convection Heat Transfer in Upward Flow.

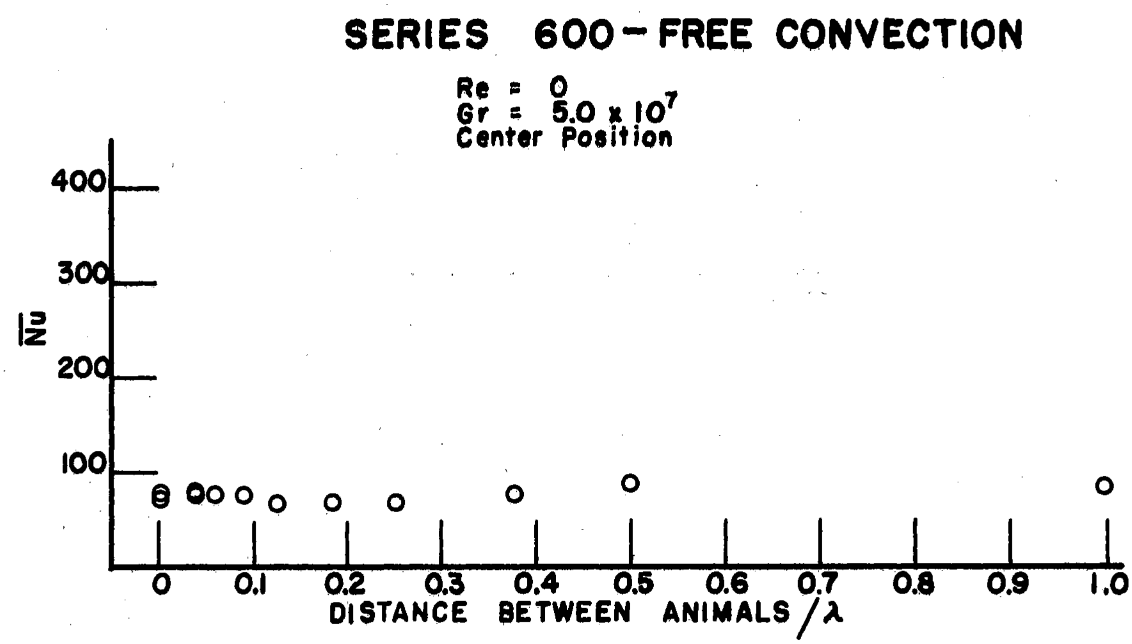


Figure 30. Effect of Group Density on Free Convection Heat Transfer.

be available around the group and through the outer portion of the hair coat for the low air velocities associated with free convection to maintain uninhibited cooling.

Animal Location in the Group

The influence on convection heat transfer of location with respect to other animals present was investigated in Experimental Series 300, 900 and 1300 with six additional models placed in various arrangements. The data for these series are plotted in Figures 31 through 33. Although the location of the animal relative to others in the group was expected to be of little influence with air flow from above or below, the presence of other animals was thought to be an important consideration in horizontal flow. However, heat transfer is as unaffected by location in horizontal flow as in vertical flow. With all other conditions equal and with some open space between the animals, the location within the group is of minor importance in convection cooling.

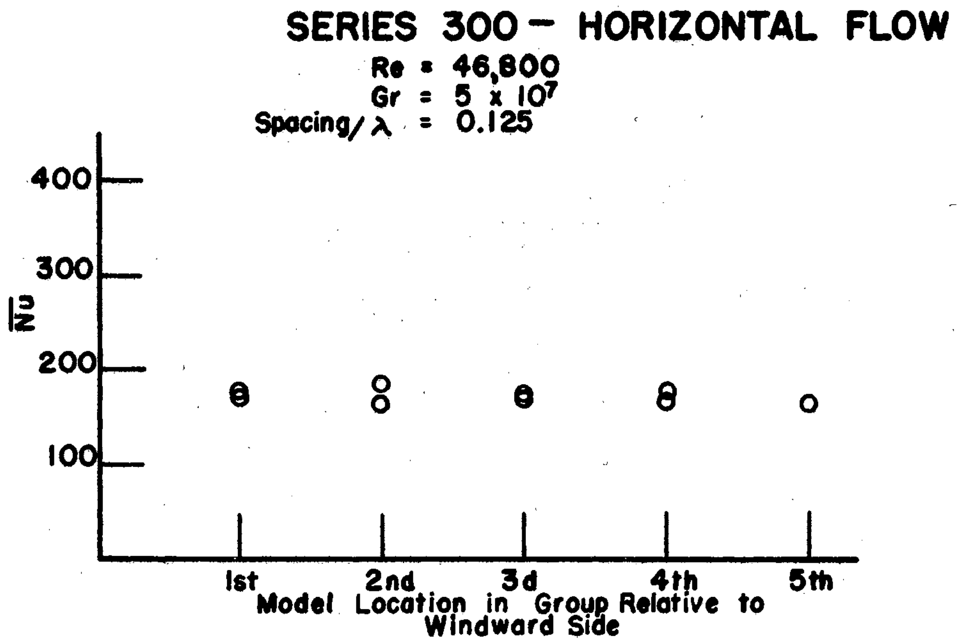


Figure 31. Effect of Location Within a Group on Convection Heat Transfer in Horizontal Flow.

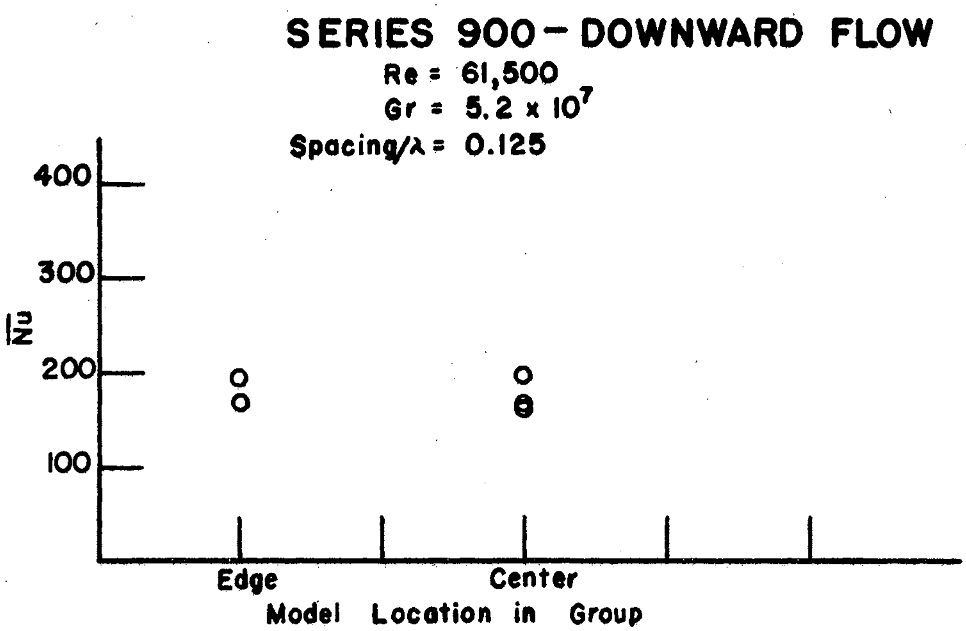


Figure 32. Effect of Location Within a Group on Convection Heat Transfer in Downward Flow.

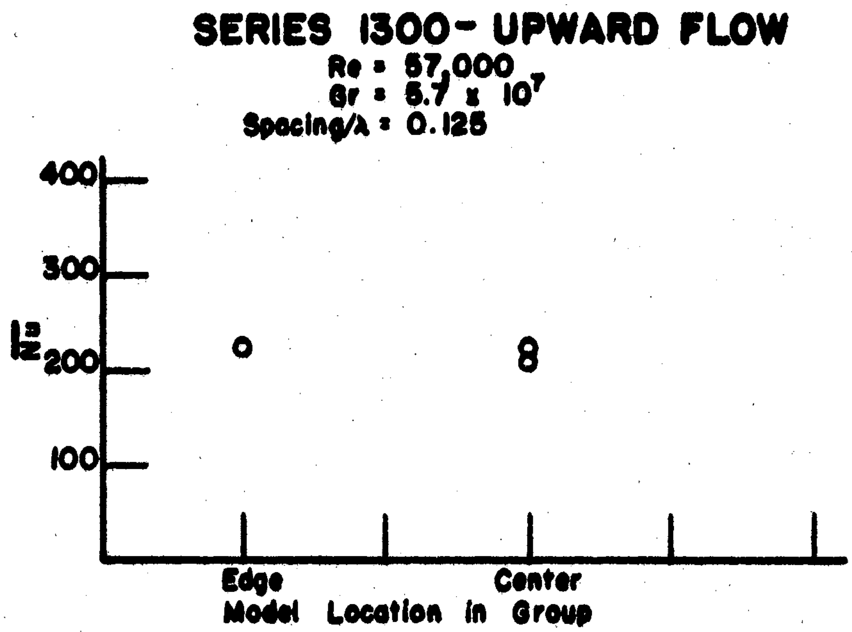


Figure 33. Effect of Location Within a Group on Convection Heat Transfer in Upward Flow.

CHAPTER IX

EVALUATION OF PREDICTION EQUATIONS

The prediction equations developed from the experimental data adequately represent convection heat transfer from the model. However, to verify that the information gained from the model is representative of convection from live animals, the predicted heat transfer coefficients were compared with coefficients computed from published data of controlled experiments on cattle.

Sufficient information was included in two independent series of experiments (9, 48, 77) to calculate heat transfer coefficients from Newton's Law of Cooling using observed values of energy transfer and temperature gradients. The coefficients thus calculated were compared to the value of the coefficient computed by substituting measured values of given conditions in the prediction equation developed from the model study.

Experiments at the Missouri Psychroenergetic Laboratory had as objectives the determination of the effect of wind on heat exchange, body temperature regulation, and surface temperatures of dairy cattle. Mature Jersey, Holstein, Brown Swiss and Brahman cattle were used Data are presented on the effects of low, medium and high air velocities

(0 to 0.6, 6 to 8, and 12 to 13 feet per second) at 18, 50, 65, 80, and 95 F on respiration rate, pulse rate, pulmonary ventilation rate, rectal temperature, evaporative cooling from the respiratory tract, heat production, total evaporative cooling, non-evaporative cooling and corresponding temperatures for hair surface, skin surface, wall and air. A summary of reported data used for model evaluation is given in Tables VIII through XI with values for $\overline{Nu}$ and heat transfer coefficient computed from Newton's Law of Cooling and from the prediction equations. For air temperatures below 67 F, the conditions were too far removed from the range used in the model studies to be valid for comparison.

Heat dissipation was reported as surface evaporative, respiratory evaporative, and non-evaporative cooling. The non-evaporative portion included conducted, radiated and convected heat transfer, but the conducted portion was considered negligible by the authors. Heat dissipated by radiation was calculated with the same procedure and equations as were used for determining radiation from the model. The experimental surroundings were geometrically similar to those of the model for upward air flow, so corresponding shape factors listed in Table VI of Chapter VI were used for calculation of net radiant energy exchange. Values for emissivity were obtained from the curves in Figure 4 of Chapter IV, assuming the reflectance characteristic for Brown Swiss and Brahman would be similar to those of Jersey, and the white and black portions of Holstein surfaces were such that

TABLE VIII

COMPARISON OF PREDICTED HEAT TRANSFER COEFFICIENTS WITH VALUES DETERMINED FROM OBSERVATIONS AT THE MISSOURI PSYCHROENERGETIC LABORATORY (JERSEY CATTLE $\lambda = 1.82'$)

		67 F Air			80 F Air			95 F Air	
	*Air Velocity - Ft/Sec	Free	9.1	12.5	Free	6.9	11.3	Free	13.1
	Dissipated Heat - BTU/hr-ft^2								
	*Non-Evaporative	23.6	34.3	32.8	24.4	22.1	24.0	10.7	6.2
	Radiant	11.8	7.2	6.8	7.0	5.2	4.1	3.0	1.5
	Convective	11.8	27.1	26.0	17.4	16.9	19.9	7.7	4.7
	Temperature Differential - Radiant, F	23.2	13.8	12.8	13.0	9.9	7.8	5.6	2.6
	*Wall Temperature, F	65.9	65.2	65.6	80.4	79.5	80.3	93.1	94.7
	*Hair Surface Temperature, F	89.1	79.0	78.4	93.4	89.4	88.1	98.7	97.3
	*Air Temperature, F	64.6	63.7	65.4	79.9	79.5	80.3	94.0	94.5
	Temperature Differential - Convective, F	24.5	15.3	13.0	13.5	9.9	7.8	4.7	2.8
OBSERVED	Heat Transfer Coef., $h = \frac{\text{Non-Evap Heat}}{\Delta T \text{ Convective}}$	0.48	1.77	2.00	1.29	1.71	2.55	1.64	1.68
OBSERVED	$\overline{Nu} = \frac{h\lambda}{k}$	59	218	245	155	206	308	195	230
	Re x 10^{-3}	0	100.5	137.0	0	73.0	120.0	0	134.0
	Gr x 10^{-8}	3.2	2.0	1.74	1.63	1.02	0.81	0.52	0.31
PREDICTED	$\overline{Nu} = 6.85\ Gr^{0.137}$	106			95			79	
PREDICTED	$\overline{Nu} = 0.00209\ Re + 710\ Gr^{-0.065} - 139$		276	355		226	329		371
PREDICTED	Heat Transfer Coef., $h = \frac{\overline{Nu}\ k}{\lambda}$	0.87	2.24	2.90	0.78	1.87	2.72	0.66	3.12

*Values taken directly from publications (48, 77)

TABLE IX

COMPARISON OF PREDICTED HEAT TRANSFER COEFFICIENTS WITH VALUES DETERMINED FROM OBSERVATIONS AT THE MISSOURI PSYCHROENERGETIC LABORATORY

(HOLSTEIN CATTLE $\lambda = 2.06'$)

		67 F Air			80 F Air			95 F Air	
	*Air Velocity - Ft/Sec	Free	7.05	12.9	Free	6.6	12.8	Free	12.9
	Dissipated Heat - BTU/hr-ft^2								
	*Non-Evaporative	35.8	42.0	43.2	22.1	34.6	30.3	13.3	19.6
	Radiant	16.4	12.6	9.2	11.7	6.5	5.5	5.2	2.6
	Convective	19.4	29.4	34.0	10.4	28.1	23.3	8.1	17.0
	Temperature Differential - Radiant, F	21.9	15.9	11.2	14.7	8.1	6.8	6.2	3.1
	*Wall Temperature, F	65.5	65.2	64.1	80.3	80.0	80.7	94.0	94.6
	*Hair Surface Temperature, F	87.4	81.1	75.3	95.0	88.1	87.5	100.2	97.7
	*Air Temperature, F	64.2	64.9	63.6	80.2	80.0	80.7	95.0	94.2
	Temperature Differential - Convective, F	23.2	16.2	11.7	14.8	8.1	6.8	5.2	3.5
OBSERVED	Heat Transfer Coef., $h = \frac{\text{Non-Evap Heat}}{\Delta T \text{ Convective}}$	0.84	1.82	2.90	0.70	3.47	3.42	1.58	4.85
OBSERVED	$\overline{Nu} = \frac{h \lambda}{k}$	115	251	403	95	475	468	212	653
	Re x 10^{-3}	0	87.2	161.0	0	79.0	153.0	0	150.0
	Gr x 10^{-8}	4.4	3.14	2.32	2.59	1.45	1.21	0.83	0.57
PREDICTED	$\overline{Nu} = 6.85\ Gr^{0.137}$	111			104			85	
PREDICTED	$\overline{Nu} = 0.00209\ Re + 710\ Gr^{-0.065} - 139$		241	399		234	392		398
PREDICTED	Heat Transfer Coef., $h = \frac{\overline{Nu}\, k}{\lambda}$	0.80	1.74	2.86	0.75	1.71	2.87	0.63	2.96

*Values taken directly from publications (48, 77)

TABLE X

COMPARISON OF PREDICTED HEAT TRANSFER COEFFICIENTS WITH VALUES DETERMINED FROM OBSERVATIONS AT THE MISSOURI PSYCHROENERGETIC LABORATORY
(BROWN SWISS CATTLE $\lambda = 2.0'$)

		67 F Air			80 F Air			95 F Air	
	*Air Velocity - Ft/Sec	Free	7.05	12.9	Free	6.6	12.8	Free	12.9
	Dissipated Heat - BTU/hr-ft^2								
	*Non-Evaporative	25.8	33.2	38.3	14.7	31.4	30.6	8.1	8.1
	Radiant	11.0	8.4	5.5	6.6	5.2	3.5	3.1	1.2
	Convective	14.8	24.8	32.8	8.1	26.2	27.1	5.0	6.9
	Temperature Differential - Radiant, F	21.4	16.0	10.3	12.4	9.9	6.3	5.2	2.3
	*Wall Temperature, F	64.9	65.1	64.1	79.9	80.0	80.7	94.0	94.6
	*Hair Surface Temperature, F	86.3	81.8	74.4	92.3	89.9	87.0	99.2	96.9
	*Air Temperature, F	64.2	64.9	63.6	80.2	80.0	80.7	95.0	94.2
	Temperature Differential - Convective, F	22.1	16.2	10.8	12.1	9.9	6.3	4.2	2.7
OBSERVED	Heat Transfer Coef., $h = \frac{\text{Non-Evap Heat}}{\Delta T \text{ Convective}}$	0.67	1.53	3.04	0.67	2.65	4.30	1.19	2.56
OBSERVED	$\overline{Nu} = \frac{h\lambda}{k}$	90	206	410	89	350	570	155	334
	Re x 10^{-3}	0	84.5	157.0	0	78.0	148.0	0	145.0
	Gr x 10^{-8}	3.85	2.86	1.96	1.94	0.98	1.03	0.62	0.40
PREDICTED	$\overline{Nu} = 6.85\ Gr^{0.137}$	108			96			81	
PREDICTED	$\overline{Nu} = 0.00209\ Re + 710\ Gr^{-0.065} - 139$		238	395		239	385		391
PREDICTED	Heat Transfer Coef., $h = \frac{\overline{Nu}\ k}{\lambda}$	0.80	1.77	2.92	0.73	1.80	2.90	0.62	2.99

*Values taken directly from publications (48,77)

TABLE XI
COMPARISON OF PREDICTED HEAT TRANSFER COEFFICIENTS WITH VALUES DETERMINED FROM OBSERVATIONS AT THE MISSOURI PSYCHROENERGETIC LABORATORY
(BRAHMAN CATTLE $\lambda = 2.0'$)

		67 F Air			80 F Air			95 F Air	
	*Air Velocity - Ft/Sec	Free	9.1	12.5	Free	6.9	11.3	Free	13.1
	Dissipated Heat - BTU/hr-ft^2								
	*Non-Evaporative	24.3	23.6	25.8	15.9	15.5	17.7	5.9	10.3
	Radiant	11.5	6.8	5.9	7.8	3.7	2.6	3.0	1.5
	Convective	12.8	16.8	19.8	8.1	11.8	15.1	2.9	7.8
	Temperature Differential - Radiant, F	22.6	12.9	11.1	14.8	7.1	5.0	5.7	3.1
	*Wall Temperature, F	65.2	64.7	65.4	80.2	79.5	80.3	92.5	94.4
	*Hair Surface Temperature, F	87.8	77.6	76.5	95.0	86.6	85.3	98.2	97.5
	*Air Temperature, F	64.6	63.7	65.4	79.9	79.5	80.3	94.0	94.5
	Temperature Differential - Convective, F	23.2	13.9	11.1	15.1	7.1	5.0	4.2	3.0
OBSERVED	Heat Transfer Coef., $h = \frac{\text{Non-Evap Heat}}{\Delta T \text{ Convective}}$	0.55	1.20	1.78	0.54	1.66	3.02	0.69	2.60
OBSERVED	$\overline{Nu} = \frac{h\lambda}{k}$	74	162	240	71	220	400	90	339
	Re x 10^{-3}	0	110.0	151.0	0	80.5	132.0	0	147.0
	Gr x 10^{-8}	4.03	2.52	2.00	2.42	1.17	0.83	0.62	0.44
PREDICTED	$\overline{Nu} = 6.85\ Gr^{0.137}$	109			104			82	
PREDICTED	$\overline{Nu} = 0.00209\ Re + 710\ Gr^{-0/065} - 139$		293	383		241	354		393
PREDICTED	Heat Transfer Coef., $h = \frac{\overline{Nu}\,k}{\lambda}$	0.81	2.17	2.84	0.79	1.81	2.66	0.63	3.01

*Values taken directly from publications (48, 77)

emissivity was the same as for Hereford. For the hair surface temperatures reported, a value for emissivity of 0.4 was used for Jersey, Brown Swiss and Brahman, 0.72 for Holstein, and 0.9 for the surrounding walls. The calculated net radiant heat exchange was then subtracted from non-evaporative heat loss to determine convected heat. Values of characteristic length used in the computations were determined from the recorded animal weights and the Brody dimensional relationships.

Forced air flow was generated by overhead fans blowing downward on the cattle. Cooling effects were therefore compared with those predicted by the equation for downward air flow. Although low air velocity was designated as ranging from 0 to 0.6 feet per second, the report described it as the normal rate of air movement without fans. Cooling under low velocity conditions was therefore compared to values computed from the prediction equation for free convection.

Heat transfer coefficients determined from the published data are plotted with values computed from the prediction equations in Figures 34 and 35. Each observed value is the result of measurements on only two to four cattle so is in a few cases somewhat erratic. For instance, the heat transfer coefficient observed for Jersey cattle in 95 F air is no better at high velocity than in still air. Similarly, Holstein in 80 F air are shown to be cooled as effectively at six feet per second as at twelve. With these exceptions the observed patterns are realistic and compare very favorably to the predicted values.

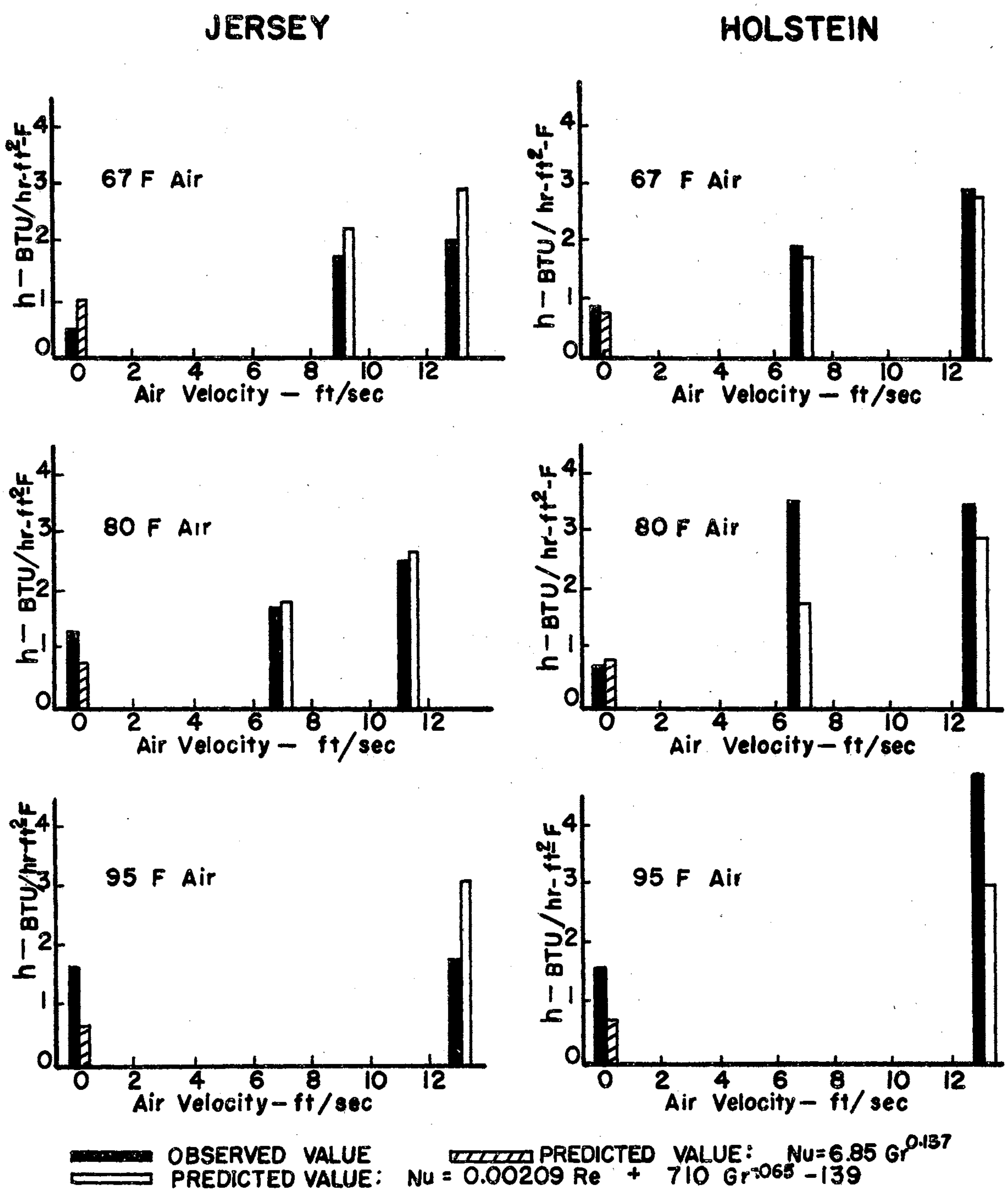


Figure 34. Comparison of Heat Transfer Coefficient Computed from Prediction Equations with Those Observed at Missouri Psychroenergetic Laboratory (Jersey and Holstein).

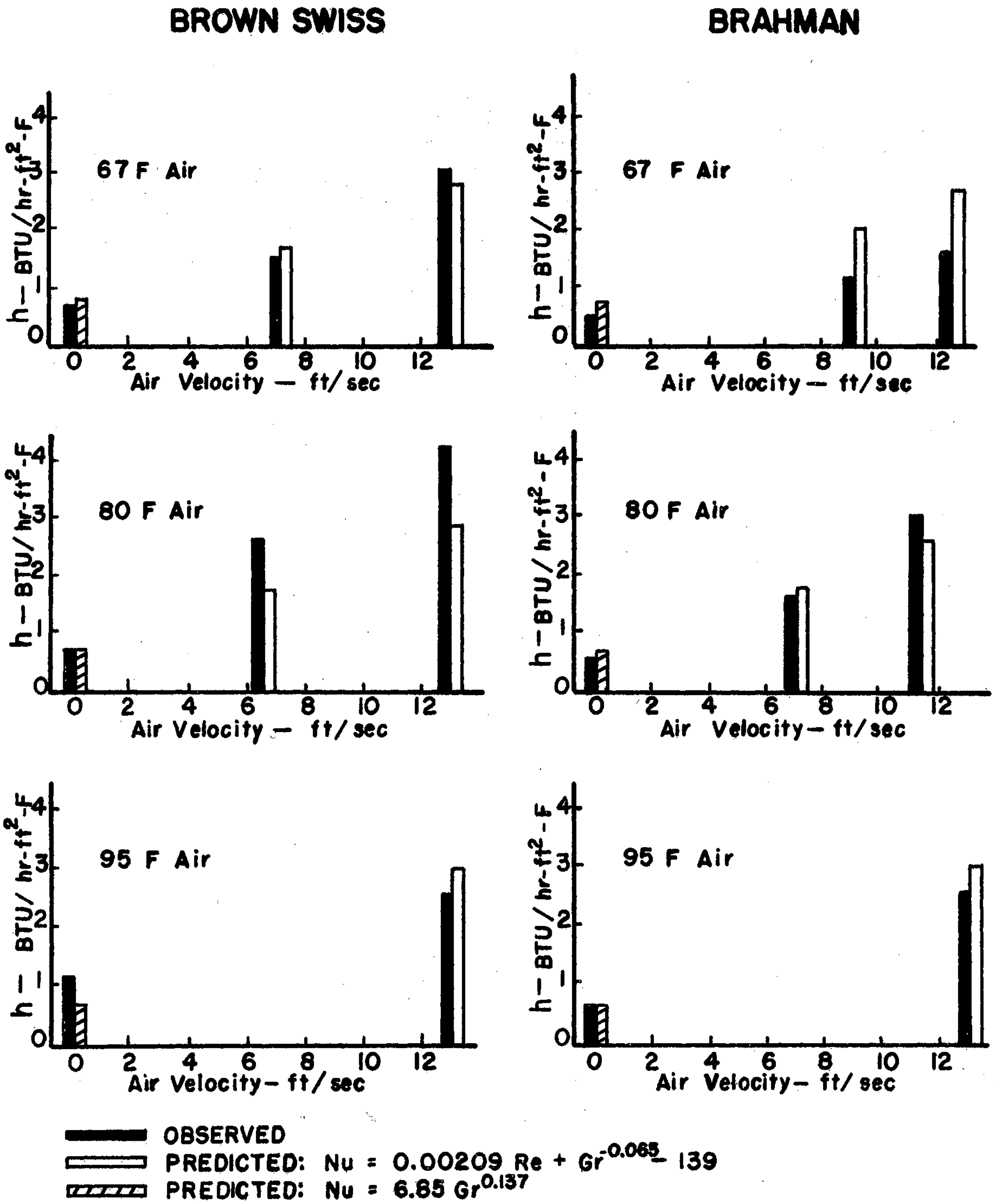


Figure 35. Comparison of Heat Transfer Coefficient Computed from Prediction Equations with Those Observed at Missouri Psychroenergetic Laboratory (Brown Swiss and Brahman).

The Hannah Dairy Research Institute reported studies limited to conditions of free convection (9). Research was conducted on two Aberdeen Angus steers and quantitatively defines energy metabolism and heat emission in air temperatures from 23 to 98 F. Values are reported for heat production, evaporative and non-evaporative heat dissipation, energy production, and corresponding temperatures for air, hair surface and surrounding walls. The non-evaporative heat dissipation again includes conduction, radiation and convection. In this report, the description of facilities is inadequate to permit calculation of radiation. However, the authors assume losses of heat from the animal by radiation and by convection are approximately equal in magnitude. This assumption was used in determining the coefficient of heat transfer from observed values for heat dissipated and temperature gradient. Results of computations from the data of four experiments and from the prediction equations are compared in Table XII. Comparisons include only those experiments in which air temperatures were 75 F or higher. Animal trunk diameter, based on Brody's relationships for an 1100 pound steer, was assumed to be 2.0 feet. The predicted values for heat transfer coefficient are slightly higher than those observed but are of the same order of magnitude.

TABLE XII

COMPARISON OF PREDICTED HEAT TRANSFER COEFFICIENTS WITH VALUES DETERMINED FROM OBSERVATIONS AT HANNAH DAIRY RESEARCH INSTITUTE

Experiment Number	2	5	8	11
Temperature, F				
Surface	93.9	100.0	91.9	99.4
Air	77.4	95.0	76.8	94.8
Differential	16.5	5.0	15.1	4.6
Heat Loss, $BTU/hr\text{-}ft^2$				
Non-evaporative	14.10	5.50	13.60	3.72
Radiant	7.05	2.75	6.80	1.86
	7.05	2.75	6.80	1.86
From Newton's Law of Cooling				
h = Non-evap/ΔT $BTU/hr\text{-}ft^2\text{-}F$	0.425	0.55	0.45	0.41
$\overline{Nu}$-observed = $h\lambda/k$	57	72	60	54
Grashof number x 10^{-8}	2.68	0.73	2.40	0.67
From prediction equation				
$\overline{Nu}$-predicted = $6.85\ Gr^{0.137}$	100	83	99	81
$h = \overline{Nu}\ k/\lambda$	0.75	0.63	0.75	0.62

A statistical analysis was made of all paired values of predicted and observed data for the experiments at Missouri Psychroenergetic Laboratory and at Hannah Dairy Research Institute. For each data point in the experiments, the observed value of $\overline{Nu}$ was plotted against the predicted value (Figure 36). Although there is appreciable scatter,

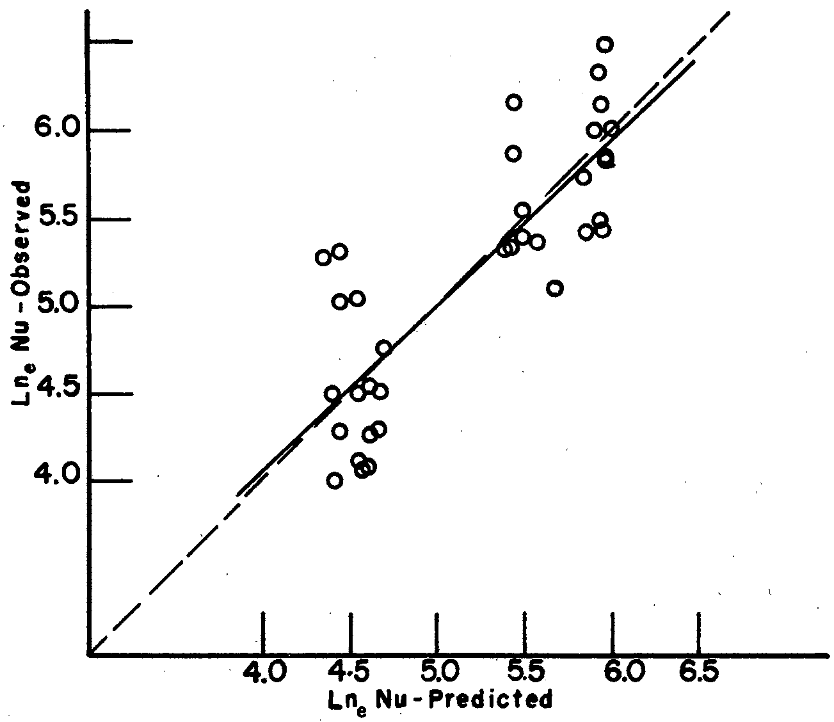


Figure 36. Statistical Comparison of Heat Transfer Coefficients Determined at the Psychro-energetic Laboratory and at Hannah Dairy Research Institute with Corresponding Predicted Coefficients.

particularly in the observed values, the regression line is very nearly identical to a line representing an ideal relationship. The $\overline{Nu}$-observed versus $\overline{Nu}$-predicted regression line is the same in elevation as the line representing $\overline{Nu}$-observed equals $\overline{Nu}$-predicted and has a slope of 0.944, which is not statistically different from 1.0. Thus, where comparisons are available, the heat transfer coefficients predicted by the equations developed from the model are the same as those computed from observations for the given conditions, and the thermal model predictions are representative of non-evaporative convection heat transfer characteristics of cattle.

CHAPTER X

DISCUSSION OF RESULTS

The equations formed from the model studies predict convective cooling coefficients from the model and from live cattle. Use is limited to conditions where the Grashof and Reynolds numbers lie in or near the range of values used in the experiments where cooling is beneficial. That is, where heat conservation may be important, the temperature differential and related Gr are too far removed from values considered in the study, and the equations might make erroneous predictions.

During preliminary investigations, calculations using characteristic ranges of air velocities, temperature differentials and cylinder dimensions indicated that the influence of free convection bouyancy forces were likely small, but could not arbitrarily be neglected. The component equations describing the behaviour of $\overline{Nu}$ with changing Gr verify that slight influence exists, but statistically is barely significant at the 95 percent level.

The magnitude of the effect of bouyant forces can be illustrated by comparing values of $\overline{Nu}$ computed from the three equations over a range of conditions. Predicted values are plotted in Figure 37 for

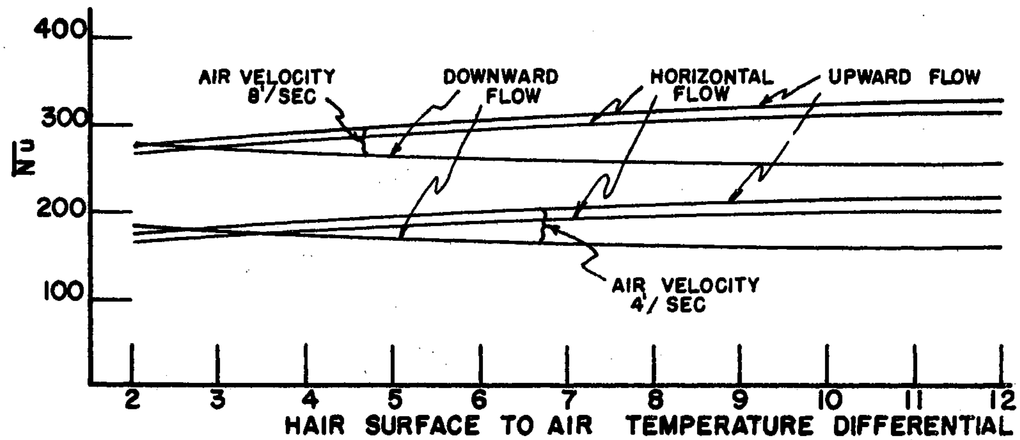


Figure 37. Comparison of Values Predicted by Equations for all Three Directions of Air Flow for a Range of Hypothetical Conditions.

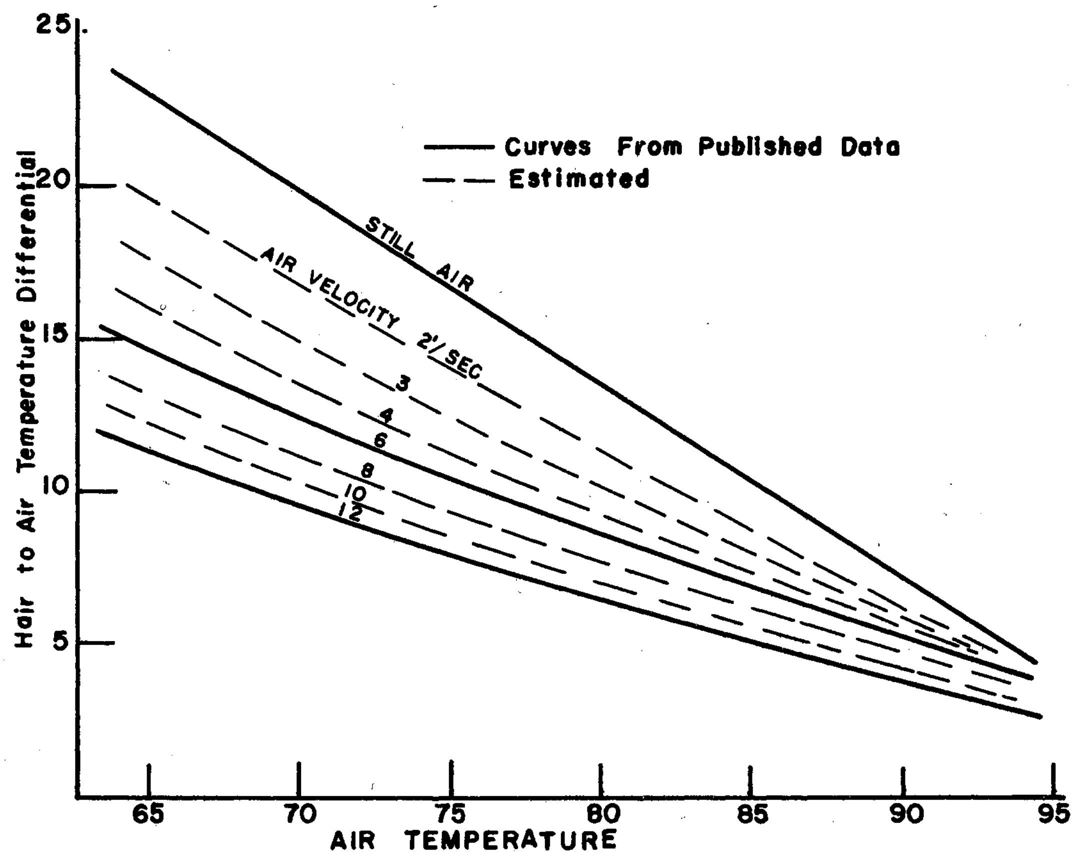


Figure 38. Hair to Air Temperature Differentials Based on Data from Missouri Psychroenergetic Laboratory.

two air velocities and a complete range of temperature differentials characteristic of situations requiring cooling. As the temperature differential decreases and influence of bouyancy forces diminishes, ideally the curves would converge to a value common to all equations. This tendency is exhibited by the curves although they actually cross at a positive value of Gr.

Cattle can withstand air temperatures up to 85 F without appreciable stress, so cooling would normally be of little benefit at temperatures below 85 F. If air at 85 F is moved at a rate of four feet per second, an examination of the data from Missouri Psychroenergetic Laboratory (Figure 38) indicates the hair surface to air temperature differential would be about 7 F. The predicted values for $\overline{Nu}$ for these conditions (Figure 37) would range from about 165 to 205, depending upon direction of air movement. Increasing air velocity to eight feet per second would lower the temperature differential to about 6 F and move the predicted $\overline{Nu}$ values to a range of 260 to 305. By interpolation, a six feet per second air velocity would result in a predicted $\overline{Nu}$ value range of 215 to 253. Therefore, based on the predicted values for $\overline{Nu}$, a one to two foot per second change in velocity of air at 85 F is as influential on heat transfer rate as is the direction the air is moving. As air temperatures increase above 85 F, the influence of direction of air flow decreases, but that of increased air velocity remains the same. Thus, although slightly better cooling is provided when air movement is upward, the improvement may be overshadowed by the advantage of locating the air source to produce a higher air velocity on the cattle.

Because of the relatively small differences in predicted heat transfer with different air flow directions, a single equation will give adequate results for all directions of air flow. Forced convection equations for heat transfer from cylinders in cross-flow are reported by Parker in the simple form $\overline{Nu} = C\ Re^{n}$ (59), which is the same form as component equations relating $\overline{Nu}$ to Re. The three exponential component equations are

$$\overline{Nu} = 0.590\ Re^{0.537} \quad \text{for horizontal air flow}$$

$$\overline{Nu} = 0.691\ Re^{0.524} \quad \text{for downward air flow}$$

$$\overline{Nu} = 0.620\ Re^{0.535} \quad \text{for upward air flow}$$

which can be represented by one simple equation

$$\overline{Nu} = 0.65\ Re^{0.53}$$

Computed values from the three component equations and the one approximating all three for a range of Re are compared in Table XIII.

TABLE XIII

PREDICTED VALUES OF $\overline{Nu}$ FOR THE THREE COMPONENT EQUATIONS AND AN APPROXIMATE EQUATION

Reynolds Number	10,000	40,000	80,000	120,000	160,000
Component Equations					
$\overline{Nu} = 0.590\ Re^{0.537}$	83	174	254	328	359
$\overline{Nu} = 0.691\ Re^{0.524}$	86	179	259	332	359
$\overline{Nu} = 0.620\ Re^{0.535}$	85	180	264	340	370
Approximate Equation					
$\overline{Nu} = 0.65\ Re^{0.53}$	86	178	259	337	362

This simple equation is compared with the three equations considered both Re and Gr in Figure 39, which is like Figure 37, but with the additional curve superimposed. The simple equation represents an approximate prediction for $\overline{Nu}$ for all directions of air flow. Its use must be limited to values of Re between 10,000 and 160,000, a range which encompasses air velocities of 0.9 to 14 feet per second on mature cattle.

Eckert and Drake (26) relate $\overline{Nu}$ and Re for cylinders in crossflow (Figure 40). They also include the relationship in equation form $\overline{Nu} = 0.43 + C\ Re^m$, but for all except very low values of Re, the constant, 0.43, is insignificant.

Parker includes values for the coefficient C, and exponent n, for various ranges of Re for use in the equation for cylinders in crossflow, $\overline{Nu} = C\ Re^n$. These values are listed in Table XIV.

TABLE XIV

COEFFICIENTS AND EXPONENTS FOR CALCULATION OF HEAT TRANSFER FROM A CIRCULAR CYLINDER TO AIR FLOW NORMAL TO ITS AXIS, BY EQUATION $\overline{Nu} = C\ Re^n$

Reynolds	C	n
1 - 4	0.891	0.330
4 - 40	0.821	0.385
40 - 4,000	0.615	0.466
4,000 - 40,000	0.174	0.618
40,000 - 400,000	0.0239	0.805

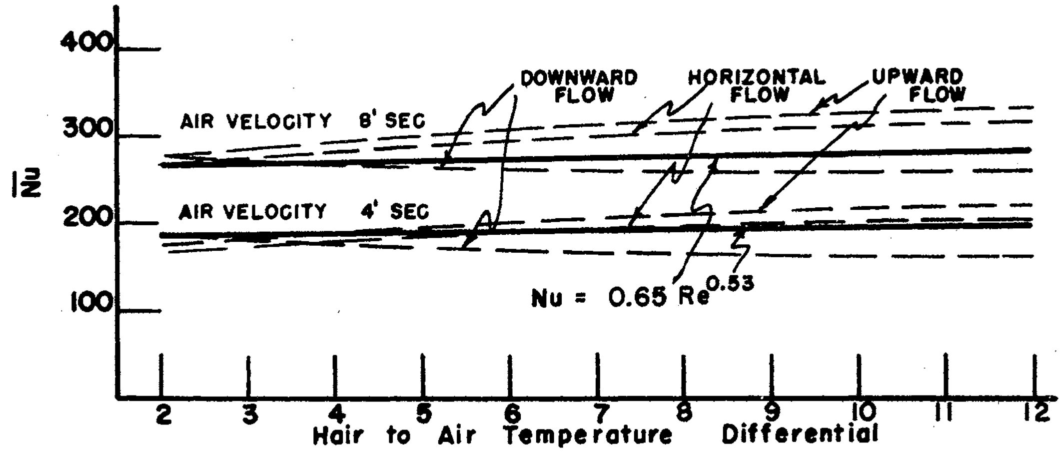


Figure 39. Comparison of Values of $\overline{Nu}$ Predicted by Composite Component Equation Disregarding Bouyant Forces With Values Predicted by the Equations for Three Directions of Air Flow.

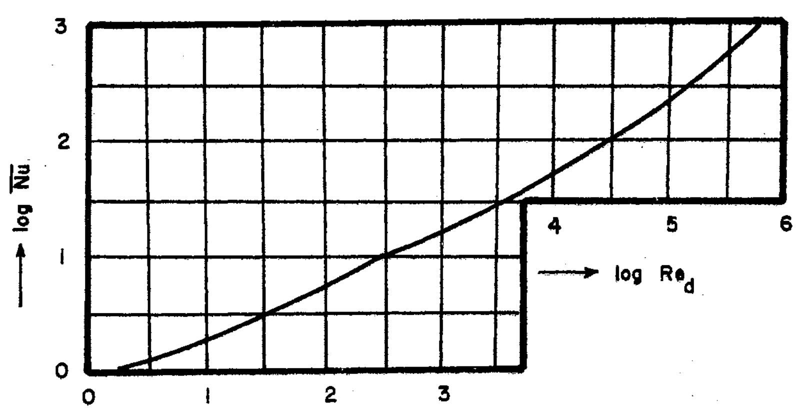


Figure 40. Average Film Heat Transfer Coefficient on a Cylinder in Flow of Air Normal to its Axis. Reproduced from Eckert and Drake (26).

Direct comparison of values from the model prediction equation $\overline{Nu} = 0.65\ Re^{0.53}$ cannot be made with values from Eckert and Drake's curve or the equation for cylinder in cross-flow because of differences in characteristic length, but examination shows the general relationship is the same. The equation from the model assumes trunk diameter, the diameter of the largest cylinder, as characteristic length in the Reynolds and Nusselt numbers. To relate the heat transfer predicted by the model to heat transfer from cylinders in cross-flow, a characteristic length more representative of some effective mean diameter of all cylinders used in the model must be employed. Considering the small diameter of the legs, a composite cylinder diameter would be considerably smaller than the diameter of the trunk.

Only rough approximations can be made but assume, for instance, an average h can be computed for the cylinders in the model by weighting the value of h for each cylinder according to the area it represents. An air velocity of 10 feet per second at mean film temperature of 90 F would be characterized by Reynolds numbers of 76, 200, 9, 500 and 34, 600 for model cylinders representing the trunk, legs, and head and neck, respectively. Re for the oblong head and neck section are based on the diameter of a circular tube with equal surface. For the same ambient condition, the Reynolds number for the model is 76, 200. Using values for the coefficient C and exponent n from Table XIV, the heat transfer coefficients for the cylinders are

$$\overline{Nu}_1 = 0.0239\,(76,200)^{0.805} = 203$$

$$h_1 = \overline{Nu}\,k/\lambda = 2.33 \text{ BTU/hr-ft}^2\text{-F}$$

trunk

$$\overline{Nu}_2 = 0.174\,(9,500)^{0.618} = 50$$

$$h_2 = \overline{Nu}\,k/\lambda = 4.58 \text{ BTU/ft}^2\text{-hr-F}$$

legs

$$\overline{Nu}_3 = 0.174\,(34,600)^{0.618} = 109$$

$$h_3 = \overline{Nu}\,k/\lambda = 2.76 \text{ BTU/hr-ft}^2\text{-F}$$

head and neck

Then weighting the values of h according to the surface area to get an average h,

$$h_{ave} = \frac{h_1A_1 + h_2A_2 + h_3A_3}{A_1 + A_2 + A_3}$$

$$= \frac{2.33\,(15.38) + 4.58\,(3.75) + 2.76\,(2.34)}{21.4}$$

$$= 2.77 \text{ BTU/hr-ft}^2\text{-F}$$

The heat transfer coefficient for the model is

$$\overline{Nu} = 0.65\,(76,200)^{0.53} = 252$$

$$h = \overline{Nu}\,k/\lambda = 2.88 \text{ BTU/hr-ft}^2\text{-F}$$

Comparison indicates the prediction equation from the model produces a heat transfer coefficient similar to that predicted by the equations reported for cylinders in cross-flow. The high correlation may include compensating differences in addition to the gross assumption regarding characteristic length. Turbulence levels, end effects, interference caused by cylinder assembly, and surface texture would all contribute to differences in the heat transfer.

A similar comparison can be made with the model prediction equation for free convection, $\overline{Nu} = 6.85\ Gr^{0.137}$. The equations for free convection on cylinders in the range $10^3 < GrPr < 10^9$ are

$$\overline{Nu} = 0.53\ (GrPr)^{0.25} \qquad \text{for horizontal axis}$$

$$\overline{Nu} = 0.59\ (GrPr)^{0.25} \qquad \text{for vertical axis}$$

Applying these equations to a typical condition used in the model studies wherein temperature differential is 10 F, mean film temperature is 90 F and weighting the values of h for each section according to surface area represented results in

$$h_{trunk} = 0.46\ \text{BTU/hr-ft}^2\text{-F}$$

$$h_{head\ and\ neck} = 0.556\ \text{BTU/hr-ft}^2\text{-F}$$

$$h_{legs} = 0.50\ \text{BTU/hr-ft}^2\text{-F}$$

$$h_{average} = 0.48\ \text{BTU/hr-ft}^2\text{-F}$$

Applying the equation from the model $\overline{Nu} = 6.85\ Gr^{0.137}$

$$h_{model} = 0.88\ \text{BTU/hr-ft}^2\text{-F}$$

Values from the model equation are roughly 80 percent higher than those from the horizontal cylinder. In addition to the possible causes of discrepancy suggested for forced convection comparisons, in free convection the model has cylinders in both axial and cross-flow.

Although the equations developed from the model cannot be compared directly to equations for other geometrical shapes, the qualitative relationships and predicted values of heat transfer coefficient for the model are similar to those predicted by equations for a cylinder in cross-flow.

CHAPTER XI

SUMMARY AND CONCLUSIONS

The objectives of this research were to determine the requirements of a thermal model for determining convective heat transfer from the surface of a bovine, to develop equations for predicting convective cooling, and to evaluate the equations by comparison with existing data. The study was limited to conditions where cooling would be beneficial.

The requirements of the thermal model were determined from the literature on the environmental physiology and the bioenergetics and growth of cattle. The internal heat distribution mechanism of cattle is complex, but effective, and when dissipating excess heat energy, capable of maintaining a uniform surface temperature. The bovine geometry was simulated by an assembly of objects with defined geometrical shapes with length, diameter and surface proportionately similar to corresponding parts of mature cattle. The model was scaled two-thirds the size of an 800-pound bovine. The surface was covered by cow hide with hair intact. Use of a surface with texture identical to full scale animals on the two-thirds scale model was tested for validity and found suitable. The model was equipped with an internal heat source and temperature sensing equipment for measurements to calculate heat transfer coefficients.

With the model and dimensional analysis procedures, equations were generated for mixed free and forced convection heat transfer for air movement in horizontal, upward, and downward directions. An equation was also developed for free convection heat transfer. The effects of animal arrangement and group density were investigated for all directions of air flow by placing the model in various positions and locations in a group of non-instrumented models.

The heat transfer coefficients predicted by the equations produced from the experimental data were compared with coefficients determined from observations at Missouri Psychroenergetic Laboratory and at Hannah Dairy Research Institute. The predicted values were not statistically different from those observed.

The effect of bouyant forces with different air flow directions was found to exist but was of proportionately less influence than changes in air velocity. Average Nusselt number varied with $Re^{0.53}$. This compares favorable with the empirical relationship for convective cooling from the human body where research physiologists first presumed and later verified that convective heat loss was proportional to the square root of the velocity of air movement.

The qualitative relationship also compares favorably with heat transfer from cylinders in cross-flow for the range of air velocities expected in climates modified to enhance cattle comfort.

The following conclusions can be drawn from the results of the investigation:

1. The relative change in roughness of cattle hair surfaces, resulting from reducing the body geometry to two-thirds full scale with no change in surface characteristics has an insignificant influence on heat transfer.

2. Convection heat transfer from the hair surface of a bovine can be predicted by the equations

$$\overline{Nu} = 0.00239\ Re + 28.7\ Gr^{0.096} - 90 \quad \text{for horizontal flow}$$

$$\overline{Nu} = 0.00209\ Re + 710\ Gr^{-0.065} - 139 \quad \text{for downward flow}$$

$$\overline{Nu} = 0.00238\ Re + 36\ Gr^{0.1} - 135 \quad \text{for upward flow}$$

$$\overline{Nu} = 6.85\ Gr^{0.137} \quad \text{for free convection}$$

3. The effect of bouyancy forces is slight, and reasonably accurate predictions of convection heat transfer can be made regardless of direction of air flow by the equation

$$\overline{Nu} = 0.65\ Re^{0.53} \qquad 10,000 < Re < 160,000$$

4. For a given air flow, about one or two inches of clearance between cattle is necessary to realize maximum benefit from convective cooling. At lesser spacings or where cattle are in physical contact with each other, cooling is suppressed. This conclusion, and conclusion five, considers only the interference to air flow and does not take into account the effects of heat generated by surrounding cattle.

5. For a given air flow, regardless of direction, the location of other cattle has no effect on the convection cooling coefficient provided sufficient space exists between animals.

6. Animal orientation relative to the direction of horizontal air flow does not affect convective cooling.

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