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SHORT COMMUNICATION

A correct enthalpy relationship as thermal comfort index for livestock

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Abstract Researchers working with thermal comfort have been using enthalpy to measure thermal energy inside rural facilities, establishing indicator values for many situations of thermal comfort and heat stress. This variable turned out to be helpful in analyzing thermal exchange in livestock systems. The animals are exposed to an environment which is decisive for the thermoregulatory process, and, consequently, the reactions reflect states of thermal comfort or heat stress, the last being responsable for problems of sanity, behavior and productivity. There are researchers using enthalpy as a qualitative indicator of thermal environment of livestock such as poultry, cattle and hogs in tropical regions. This preliminary work intends to check different enthalpy equations using information from classical thermodynamics, and proposes a direct equation as thermal comfort index for livestock systems.

Keywords Heat exchange · Rural installations · Animal production

Introduction

Background

The characteristics of internal environment installations for livestock are analyzed by thermodynamics, acoustics, and light and air quality parameters. The external environment is the main factor for heat exchange between internal and external environments. Microclimate conditions in installations are used as decisive elements to trigger acclimatization systems. Some authors consider that a joint analysis of dry bulb temperature and relative air humidity is important to check zootechnical environments explained by certain situations of thermal comfort and heat stress to which animals are subjected (Teeter 1990; Esmay 1982). These variables are responsible for quantifying the capacity of thermal energy in the environment, summarized by this physical variable which represents psychrometric characteristics of humid air responsible for thermal exchange processes (Çengel and Boles 2001). As a comfort index, enthalpy indicates environmental conditions related to heat stress suffered by animals (Moura et al. 1997; Nääs et al. 1995; Silva et al. 2003, 2006). This variable is often used as a comfort indicator for installations and cooling systems, indicating a quantity of thermal energy to be removed from the environment to enable thermal conditions of survival inside an installation. Enthalpy has been primarily used for humans (Chu and Jong 2008). Strategies based on physical measures of the environment help to decide on necessary steps to mantain animals in sufficient comfort in terms of thermal exchange (Chu et al. 2005). Since the beginning of the twentieth century, a lot of environmental variables, such as dry bulb temperature, black globe temperature, relative humidity, air velocity, radiation, and others, have helped to calculate comfort indexes (Medeiros et al. 2005). However, each one of the variables can present a dominant effect in determined situations, not necessarily additive or linear (Fairey 1994).

Problem statement

There are many differences among limits chosen by authors as comfort conditions in livestock systems and the indexes

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used, such as effective temperature, ETI (Houghton and Yaglou 1923), temperature and humidity, THI (Thom 1959), and black globe temperature and air humidity (Buffington 1977). The difficulty of transferring the understanding of variables involved in these indexes to productive fields produces an opportunity to formulate new indexes which consider environmental properties, and for which users only need to know three variables: dry bulb temperature, relative humidity, and local barometric pressure. For laying hens, enthalpy tables have been formulated to attend to this need in the different stages of development and to present bands referring to situations of thermal comfort, critical heat stress and alertness. Values adopted on enthalpy tables are calculated from air temperature and relative humidity (Barbosa Filho et al. 2006, 2007), without considering other meteorological variables inherent to different regions, such as, for example, atmospheric pressure. However, for consistent use of physical quantity as a thermal comfort index, more rigorous criteria are necessary. Considering the analyzed topics, each region in the country must have an adequate value of enthalpy that represents a specific situation in terms of barometric pressure, local temperature, and relative air humidity. The aim of this work was to check different enthalpy equations used by researchers, through classical thermodynamics, seeking a basis to use this variable as a thermal comfort index for animal production, considering the specific characteristics of determined regions.

Materials and methods

According to Macari and Furlan (2001), the thermal comfort band for laying hens in the sixth week is between 21 and 24°C of dry bulb temperature, with ideal relative humidity around 60%. We compared three enthalpy equations , as described below: Furlan (2001), Barbosa Filho et al. (2007), and an Albright model extension (1990), formulated in this work. The local barometric pressure of 714 mmHg of Piracicaba, São Paulo State, Brazil, was used to calculate the new equation.

Enthalpy: theorization

Absolute enthalpy (*H*) is a thermodynamic property defined as the sum of internal energy of a system (*U*) and the multiplication of pressure (p) and volume (*V*) (ASHRAE 1993):

$$H = U + pV$$

Atmospheric air is a mixture of many gases like nitrogen (N_2), oxygen (O_2), hydrogen (H_2) and other elements in minor quantities, plus water vapor. Atmospheric air without water is dry air, used as a reference to calculate enthalpy due to its almost constant mass. Referring thermal comfort as used in air refrigeration engineering, the temperature varies from −10 to 50°C; in this band, dry air is treated as ideal gas. Its specific heat capacity (c_{pd}) is 1,006 kJ/kg°C, with an error smaller than 0.2%, at constant pressure. Water vapor can also be treated as ideal gas, at constant pressure, because the saturated water pressure (p_s) is 12.3 kPa at temperatures of 50°C. This is a low value compared to barometric pressure. For pressure lower than this value, error is also neglected and the specific heat of water vapor (c_{pv}) is 1,805 kJ/kg°C and presents a error smaller than 0.2% (Çengel and Boles 2001). Thus, humid air is a mixture of dry air added to water vapor from the environment (Esmay 1982). Considering that air components are treated as perfect gases when the temperature is on the band related above, any interaction between them is disregarded. This facilitates the analysis of each factor's contribution in absolute enthalpy or total enthalpy obtained by the sum of dry air enthalpy (H_d) and water vapor enthalpy (H_v):

$$H = H_d + H_v \tag{1}$$

Absolute enthalpy (*H*) is an extensive quantity illustrating a value of energy with reference to 0°C temperature and 0% of relative air humidity contained in a determined air mass, and its unit is kilojoules (*kJ*). On the other hand, specific enthalpy (*h*) is an intensive quantity, because it considers the amount of energy per mass and its unit is *kJ/kg* of dry air. Absolute enthalpy is the sum of dry air enthalpy (H_d) and water vapor enthalpy (H_v), so each one is the result of mass multiplied by the specific enthalpy value:

$$H = H_d + H_v = m_d.h_d + m_v.h_v \tag{2}$$

m_d is the dry air mass and m_v is the water vapor mass. The specific enthalpy, an intensive quantity, is energy measurement contained in dry air mass or in water vapor mass. The division of Eq. 2 by m_d factor results in:

$$\frac{H_d + H_v}{m_d} = h_d + \frac{m_v}{m_d} h_v = h_d + w.h_v \tag{3}$$

w is the mixture ratio or absolute humidity; i.e., the relationship between water vapor mass (m_v) and dry air mass (m_d). Partial enthalpies of dry air (h_d) and satured water vapor (h_v) inside the installations (ASHRAE 1993) are utilized according to Eq. 4:

$$h = h_d + w\,h_v \tag{4}$$

h_d is the sensible heat of dry air, expressed by Eq. 5:

$$h_d = c_{pd}.t \tag{5}$$

c_{pd} is the specific heat of dry air and t is temperature. The water vapor contained in the mixture of dry air contributes to total enthalpy, as well as sensible heat and the latent heat, shown in Eq. 6:

$$h_v = w.\left(L + c_{pv}.t\right) \tag{6}$$

c_{pv} is the specific heat of water vapor and L (2,501 kJ/kg) is the latent heat of vaporization. Substituting known values of constants, the expression shown in Eq. 7 is:

$$h = 1,006\,t + w.(2501 + 1,805\,t) \tag{7}$$

Albright (1990) indicates exactly this equation and shows its concept in a study in ASHRAE (1993). The mixture ratio (w) of water vapor mass and dry air mass is related to molecular weight function of water vapor and dry air and their corresponding pressures Eq. 8:

$$w = \frac{m_v}{m_d} = 0,622.\frac{p_v}{p_d} \tag{8}$$

The dry air pressure (p_d) is the difference between barometric (p_B) and vapor pressures ($p_d = p_B - p_v$). Eq. 9 shows an approximation, due to the barometric pressure, of around 760 mmHg, a significant value when compared to vapor pressure value in terms of magnitude:

$$w = 0,622.\frac{p_v}{p_B - p_v} \cong 0,622.\frac{p_v}{p_B} \tag{9}$$

To illustrate this approximation, the following values were used: t=30°C, 60% of relative humidity, and barometric pressure of 714 mmHg. The vapor pressure in these condictions is 25 mmHg Eqs. 10 and 11. The calculated value before the approach is 0.022 and the approximate value is 0.021, and thus the approach can be adopted without compromising the results due to the insignificant difference of mixture ratios. Vapor pressure, p_v, related to saturation pressure, p_s, is calculated by Tétens equation (Berry et al. 1945):

$$p_s = 4,58\,.\,10^{7,5t/237,3+t} \tag{10}$$

The unit is millimeters of mercury (mmHg). The relative humidity air (RH) relates to the pressures of vapor and saturation ($p_v/p_s = RH/100$), thus replacing Eq. 10 in Eq. 11:

$$p_v = \frac{RH}{100}.p_s = \frac{RH}{100}.4,58\,.10^{(7,5.t/237,3+t)} \tag{11}$$

Considering relative air humidity (RH) and dry bulb temperature (t), it is possible to get a simplified equation for the mixture ratio (w), in Eq. 12:

$$w = \frac{0,622}{p_B} \times p_v \tag{12}$$

Thus, replacing the relationship found in Eq. 11 related to vapor pressure (p_v) in Eq. 12:

$$\begin{aligned} w &= \frac{0,622}{p_B}\,.\,\frac{RH}{100}\,.\,4,58\,.\,10^{(7,5.t/237,3+t)} \\ &= \frac{2,85}{p_B}\,.\,\frac{RH}{100}\,.\,10^{(7,5.t/237,3+t)} \end{aligned} \tag{13}$$

Therefore, Eq. 7 is restructured, replacing in Eq. 13:

$$h = 1,006.t + 0,0285.\frac{RH}{p_B}.10^{(7,5.t/237,3+t)}.(2501 + 1,805.t) \tag{14}$$

The unit of Eq. 15 is kJ/kg of dry air, and pressure is mmHg:

$$h = 1,006.t + \frac{RH}{p_B}.10^{(7,5.t/237,3+t)}.(71,28 + 0,052.t) \tag{15}$$

This is enthalpy's general equation of humid air, not considering situations such as sprinkling, nebulization, or even snow, as occurs in countries of the northern hemisphere. It is an extension of the Albright formula (1990), much cited by researchers who use enthalpy as a thermal comfort index in livestock systems. The first part of Eq. 15 depends only on temperature related to sensible heat measure in the environment that is transmitted or absorbed through dry exchanges such as through conduction, convection or radiaton. The second term depends on temperature and relative humidity. It is the latent heat responsible for humid heat exchange, such as local water evaporation or birds panting.

Reference adjustment

Accordingly, the enthalpy formula used in the animal comfort study, not considering extreme situations such as snow, is given by Eq. 15 in kJ/kg of dry air. That is an extension of Albright's formula. This equation is compared to equations described below. Eq. 16 is a model used by Barbosa Filho et al. (2007) and Eq. 17 is the model used by Villa Nova, cited in Furlan's thesis (2001).

$$h = \left(6,7 + 0,243\,t + \left(\left(\frac{RH}{100}\right) \times 10^{7,5t/237,3+t}\right)\right) \times 4,18 \tag{16}$$

$$h = \left(6,7 + 0,243\,t + 2,216 \times \left(\left(\frac{RH}{100}\right) \times 10^{7,5t/237,3+t} - 1\right)\right) \times 4,18 \tag{17}$$

The multiplicative factor is the converter of kilocalories to Joules. Emphasizing reference conditions, the enthalpy calculated is based on situations of 0°C and 0% of relative humidity that demonstrate the hypothetical lack of water vapor in the air; thus the mixture ratio is zero and, consequently, vaporization enthalpy is zero, or in other words, latent heat is zero. Concerning the sensible heat, dry air enthalpy, in conditions of 0°C, will also be zero, and thus in these initial conditions (0°C and 0% of relative air humidity), the enthalpy is zero. Under initial conditions, Eqs. 16 and 17 result in constant values for enthalpies and show a misconception of thermodynamics, since these values should be null:

$$If \quad t = 0^{\circ}C \quad and \quad RH = 0 \quad so$$

$$\begin{cases} h = 28,0 \; kJ/kg & (\text{equation } 16) \\ h = 18,7 \; kJ/kg & (\text{equation } 17) \end{cases}$$

Results

Meteorological data, such as temperature and relative air humidity, were used to simulate, in a climate chamber, the characterization of a typical summer day in a tropical country during the daytime. The hours chosen for the simulation are related to the most critical periods in terms of heat stress for laying hens in the sixth week (Barbosa Filho et al. 2007). For the values of temperature and relative air humidity, enthalpies were calculated by using Eqs. 15, 16 and 17.

In Fig. 1, the equations show similar profiles in growth and in decrease of the curves, but with different absolute values. Eq. 17, obtained by Furlan (2001), shows the enthalpy value (h) overestimated compared to Eq. 15, which is a result(s) of approximations and substitutions of physical constants in the equation cited by Albright (1990) and used in engineering calculations (ASHRAE 1993). Eq. 16 used by Barbosa Filho et al. (2007) shows three different situations. The first situation refers to values overestimated compared to Eq. 15. The second situation is illustrated in the interval indicated in Fig. 1. Values from Eqs. 15 and 16 are very close, so Eq. 16 is valid only for a restricted interval of values, in this case between 65 and 72% of relative air humidity and between 30 and 31.5°C of temperature. The third situation shows that, for higher values of temperature and humidity in Eq. 16, the enthalpy values were underestimated. That way, there are no conditions that can confirm the use of Eqs. 16 and 17 due to the inappropriate formula. Based on values of temperature and relative air humidity in conditions of thermal comfort (Macari and Furlan 2001), the band of enthalpy

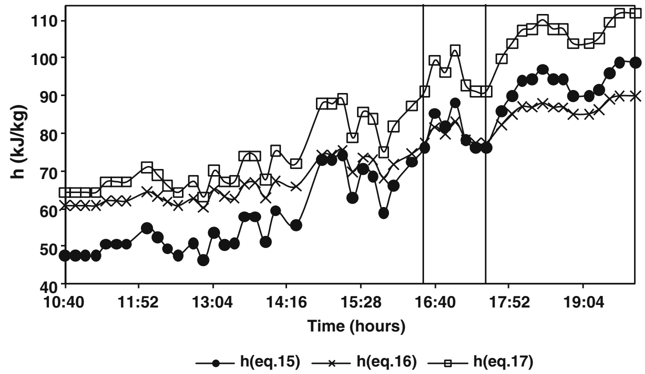

Fig. 1 Enthalpy equations (kJ/kg)

comfort, calculated by Eq. 15, is between 46 and 54 kJ/kg of dry air. This same equation delineates the real conditions of thermal exchange, because latent heat is a crucial factor to find heat exchange possibilities between the environment and the animals. This measure of heat is evidence of existing water vapor mass volume in the air, which is a limiting factor for animals to lose heat by panting. Eqs. 16 and 17 are not in accordance with values of 0°C of temperature and zero relative air humidity; the values resulting from these formulas are not zero. Enthalpy of humid air consists of two distinct parts, partial enthalpies of dry air and saturated air (Chu and Jong 2008) which represent sensible heat and latent heat, respectively, illustrated in the previous graph and annulled under the reference conditions. The work showed the approximation problems of the two enthalpy equations (Barbosa Filho et al. 2007; Furlan 2001) used in many scientific studies, but shows values as overestimated. The formula presented in this work Eq. 15 is a condition considering temperature, humidity and local barometric pressure. These properties are fundamental for a correct calculation of thermal comfort index and for knowing the thermoregulatory conditions of animals, and they are a direct variable for designing thermal conditioning systems (Chu et al. 2005, 2008).

Conclusions

This work demonstrates the reformulation of enthalpy cited by Albright (1990) to calculate thermal energy in the internal environment of installations. The theory consists in adjusting the formula so that it depends directly on temperature, relative air humidity and local barometric pressure, to ensure results of absolute energy values inside livestock systems. The equation is correct for calculating physical quantity and is appropriate as information about studies of cooling equipment designs. The analyzed equations (Barbosa Filho et al. 2007; Furlan 2001) result in wrong values compared to the Albright formula extension. Comparisons among the presented equations are limited to values of variables simulated in a climate chamber. To delineate in more detail the differences among equations, with limits of values of temperature, relative humidity and barometric pressure, analyses will be conducted and compared with experiments with animals to find the limits of thermal comfort and heat stress in relation to this new formula.

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