Mathematical Model for Thermal Environment and Broiler Chickens Performance Prediction in Acclimatized Housings

Valcimar F. Carvalho  
Federal University of Lavras

Tadayuki Yanagi Jr.  
Federal University of Lavras

Hongwei Xin  
Iowa State University, hxin@iastate.edu

Richard S. Gates  
University of Kentucky

F. Damasceno  
Federal University of Lavras

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Abstract
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Keywords
Cooling system, Wind tunnel ventilation, Thermal comfort

Disciplines
Bioresource and Agricultural Engineering

Comments

Authors
Valcimar F. Carvalho, Tadayuki Yanagi Jr., Hongwei Xin, Richard S. Gates, F. Damasceno, and S. R.P. Moraes

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Mathematical Model for Thermal Environment and Broiler Chickens Performance Prediction in Acclimatized Housings

V. F. Carvalho¹, T. Yanagi Jr.¹, H. Xin², R. S. Gates³, F. Damasceno¹, and S. R. P. Moraes⁴

¹ Engineering Dept., UFLA, MG, Brazil.  
² Agricultural & Biosystems Engineering Dept., Iowa State University, U.S.  
³ Biosystems & Agricultural Engineering Dept., University of Kentucky, U.S.  
⁴ Veterinary Dept., Goias State University, Brazil.

Abstract. This research aimed to develop and validate a mathematical model to predict thermal environment, main physiological responses, and productive performance of broiler chickens throughout acclimatized housing equipped with negative pressure tunnel ventilation systems associated with evaporative pad cooling, misting system, or both systems. Experimental results, used to validate the model showed that standard errors between simulated and measured values were smaller than or equal to 1.27, except for relative humidity, daily feed intake, and respiration rate, which were 5.43%, 1.77 g/day, and 2.15 breathes/min, respectively. It was concluded that the proposed mathematical model simulated satisfactorily all response variables modeled, allowing simulation of several production scenarios.

Keywords. Cooling system, Wind tunnel ventilation, Thermal comfort.

Introduction

A simulation of thermal environment inside of broiler houses and, main physiological and productive responses of broiler chickens throughout the houses may provide the designer and the animal producer important information helpful for decision-making.

Mathematical and computational model development and application allow reducing time and costs of design or redesign. In this context, several models have been proposed to solve field problems and to facilitate the understanding of various physical processes such as models to predict heat transfer between an animal and its surroundings (Mitchell, 1976; Mahoney & King, 1977; McArtur, 1991; Gebremedhin & Wu, 2000; Aerts et al., 2003); to predict heat or mass transfer in agricultural facilities (Medeiros, 1997), models to optimize some evaporative cooling systems (Gates et al., 1991; Singletary et al., 1996), and to depict physiological responses of poultry (Medeiros, 2001), among others.

For a detailed study about possible problems and solutions related to acclimatized houses, it is necessary to know the thermal environment profile along the house as well as its influence on broiler chickens physiological responses and productive performance. Therefore, it is necessary to have a mathematical model able to predict these variables. It provides conditions to make several simulations to evaluate different building materials, lodging density, ventilation rate and air infiltration rate through curtain, external environment thermal conditions, and evaporative cooling pads efficiency or misting effectiveness, among others.

Thus, the goal of the present work is to develop and validate a steady-state mathematical model to predict the thermal conditions and the main physiological responses and performance indexes of broiler chickens throughout acclimatized houses equipped with tunnel ventilation system, with negative pressure, and evaporative pad cooling and or misting systems, or both of them, working separately or simultaneously.

Development of Mathematical Model

Aiming for the prediction of the thermal environment, of main physiological responses and of the productive performance of broiler chickens throughout houses acclimatized with tunnel ventilation, with negative pressure and evaporative cooling pad and/or misting systems, a mathematical model was developed assuming the following hypothesis: (a) steady-state heat and mass transfer; (b) building material considered homogeneous, with constant properties; (c) heat generation inside the house by chickens of equal weight is constant; (d) humidity production by chickens of equal weight is constant; (e) uniform and one-dimensional heat flux; (f) exhausting system air flow is constant; (g) uniform convection efficiency on house external
and internal surfaces; (h) misted water flow in every control volume specified is homogeneous and constant, and; (i) thermal conditions within every control volume are constant.

**Heat Transfer**

For air temperature prediction ($t_{db}$) along the house length, it has to be divided into $n$ control volumes (CVs). Therefore, the outside air goes in a house with tunnel ventilation, negative pressure, and evaporative cooling pad system, through one of its extremities, passing through the evaporative cooling pad and then, it crosses the various CVs until it goes out through the other extremity, where the exhausting system is located. As the outside air passes through the evaporative cooling pad it is cooled in an evaporative way until a determined temperature value ($t_{PAD}$). Subsequently, as the air crosses the various CVs, its $t_{db}$ changes again due to the sensible heat loss and gain, and conversion of latent to sensible heat. In commercial houses for broiler chickens breeding used in Brazil outside air infiltration occurs, mainly through side-curtains, because house are not air tight. This air infiltration has to be considered for estimating $t_{db}$ in every CV. If the house also has an installed misting system as supplementary way of environment cooling, one has to consider this systems effectiveness in cooling air along the house.

In steady-state conditions, the energy balance in an acclimatized house is characterized for not accumulating energy in the CV. The sensible heats that go in (+) and that go out (−) every CV (i) can be expressed by the Equation 1, adapted from Albright (1990).

$$q_{in,i} + q_{vo,i} + q_{vi,i} + q_{w,i} + q_{sh,i} - q_{so,i} - q_{ms,i} = 0$$ (1)

The sensible heat dissipated by the chickens in every CV ($q_{s,i}$) can be estimated by equation 2, the net sensible heat dissipated by every chicken ($q_{s,\text{bird}}$) can be estimated by Equation 3 (Xin et al., 2001), which is a combination of the terms $q_{e,i}$ and $q_{s,i}$. According to Xin et al. (2001), Equation 3 was fitted for broilers grown in beds under commercial conditions (20 broiler housings) for $t_{db}$ and RH ranging from 20°C to 32°C and from 30% to 80%, respectively.

$$q_{s,i} = q_{s,\text{bird}} \cdot \frac{LD \cdot A_{\text{floor}}}{i}$$ (2)

$$q_{s,\text{bird}} = 1006 \cdot \rho \cdot V_{\text{ref}} \cdot (t_{db,\text{i}} - t_{db,\text{o}}) + 1006 \cdot \rho \cdot V_{\text{ref}} \cdot (t_{db,\text{i}} - t_{db,\text{o}}) + 1006 \cdot \rho \cdot V_{\text{PAD}} \cdot (t_{db,\text{i}} - t_{db,\text{o}})$$ (3)

The sensible heat transferred through the facility’s structure in $i^{\text{th}}$ CV ($q_{w,i}$) and $j^{\text{th}}$ material, can be calculated by Equation 4 (Albright, 1990).

$$q_{w,i} = \sum_{j=1}^{m} U_{ij} \cdot A_{ij} \cdot (t_{db,\text{o}} - t_{db,\text{i}})$$ (4)

According to Albright (1990), the heat transferred through facility’s floor ($q_{f,i}$) occurs mainly on its perimeter’s region, being proportional to the facility’s perimeter and to the difference between the external ($t_{db,o}$) and internal environment ($t_{db,i}$) values (equation 5). In broiler houses where beds are used, $q_{f,i}$ can be considered negligible (Albright, 1990; Medeiros, 1997).

$$q_{f,i} = F_{i} \cdot P_{i} \cdot (t_{db,o} - t_{db,i})$$ (5)

The terms $q_{w,i}$ and $q_{f,i}$ present in Equation 1, can be considered jointly, as shown in Equation 6, adapted from Albright (1990).

$$q_{w,i} + q_{f,i} = 1006 \cdot \rho \cdot V_{\text{ref}} \cdot (t_{db,\text{i}} - t_{db,\text{o}}) + 1006 \cdot \rho \cdot V_{\text{ref}} \cdot (t_{db,\text{i}} - t_{db,\text{o}}) + 1006 \cdot \rho \cdot V_{\text{PAD}} \cdot (t_{db,\text{i}} - t_{db,\text{o}})$$ (6)

After having passed through the evaporative cooling pad, $t_{db,o}$ becomes $t_{PAD}$ and can be estimated by the cooling efficiency Equation 7 (ASHRAE, 1992).

$$t_{PAD} = t_{db,o} - \beta \cdot (t_{db,o} - t_{water})$$ (7)

Sensible heat used for evaporating the misted water ($q_{water}$) is obtained by equation 8, where $\beta_i$ is the misted water fraction and $h_{fg,i}$ is the latent heat of water vaporization in the $i^{\text{th}}$ control volume, in J kg$^{-1}$, calculated respectively by Equations 9 (Singletary et al., 1996) and 10 (Albright, 1990).

$$q_{water} = \beta \cdot V_{w} \cdot h_{fg,i}$$ (8)

$$h_{fg,i} = 2.914 \times (10^{0.0106 \cdot t_{db,i}}) + 0.094 (10^{0.0363} - 1) \cdot t_{db,i} + 0.0106 (10^{0.0077} - 1) \cdot \text{RH}_i$$ (9)

Substituting equations 2 to 6 and 8 in equation 1, it is possible to calculate $t_{db,i}$ in the CV (i), as shown in equation 11.

$$t_{db,i} = \frac{\sum U_{ij} \cdot A_{ij} \cdot F_{i} \cdot P_{i} \cdot \rho \cdot V_{\text{ref}} \cdot (t_{db,\text{o}} + 1006 \cdot \rho \cdot V_{\text{ref}} \cdot (t_{db,\text{i}} - t_{db,\text{o}}) + 1006 \cdot \rho \cdot V_{\text{ref}} \cdot (t_{db,\text{i}} - t_{db,\text{o}}) + 1006 \cdot \rho \cdot V_{\text{PAD}} \cdot (t_{db,\text{i}} - t_{db,\text{o}}) - \beta \cdot V_{w} \cdot h_{fg,i}}{\sum U_{ij} \cdot A_{ij} \cdot F_{i} \cdot P_{i} \cdot \rho \cdot V_{\text{ref}} \cdot (t_{db,\text{i}} - t_{db,\text{o}}) + 1006 \cdot \rho \cdot V_{\text{ref}} \cdot (t_{db,\text{i}} - t_{db,\text{o}}) + 1006 \cdot \rho \cdot V_{\text{PAD}} \cdot (t_{db,\text{i}} - t_{db,\text{o}}) - \beta \cdot V_{w} \cdot h_{fg,i}}$$ (11)

**Mass Transfer**

Mass balance (air + water) in the broiler house can be determined analogously to the heat balance, dividing the mass balance into CVs. First the outside air is cooled as it passes through the evaporative
cooling pad and it goes in the house with certain initial humidity. From this instant, the air changes due to the humidity produced by the animals, by the air infiltration that occurs in every CV, and in the case of existing, the water evaporation by the misting system, being express by continuity Equation 12, adapted from Albright (1990).

\[ \dot{m}_{w, j} = \dot{m}_{ref, j} + \dot{m}_{PAD, j} \quad (12) \]

The mass flow of the air that passes through the evaporative cooling pad (\(\dot{m}_{PAD, j}\)), that infiltrates (\(\dot{m}_{inf, j}\)), that comes from the previous CV (\(\dot{m}_{i-j}\)) and that goes out the actual CV (\(\dot{m}_{j}\)) can be calculated by Equations 13, 14, 15 and 16, respectively.

\[ \dot{m}_{PAD, j} = \rho_{PAD} \dot{V}_{PAD} W_{PAD} \quad (13) \]

\[ \dot{m}_{inf, j} = \rho_{inf} \dot{V}_{inf, j} W_{inf, j} \quad (14) \]

\[ \dot{m}_{i-j} = \rho_{i-j} \dot{V}_{i-j} W_{i-j} \quad (15) \]

\[ \dot{m}_{j} = \rho_{j} \dot{V}_{j} W_{j} \quad (16) \]

The water mass applied by the misting system (\(\dot{m}_{w, mist, j}\)) in every CV can be estimated by Equation 17 and the mass produced by the chickens (\(\dot{m}_{p, j}\)) in every CV, by Equation 18. The mass production by every chicken \(\dot{m}_{p, bird, j}\) is estimated by Equation 19 (Xin et al., 2001), where LHP is the percentage of sensible heat dissipated per chicken as latent heat, Equation 20.

\[ \dot{m}_{w, mist, j} = \beta_{w, mist} \cdot \dot{m}_{w, bird, j} \quad (17) \]

\[ \dot{m}_{p, j} = \dot{m}_{p, bird, j} \cdot A_{LD, j} \quad (18) \]

\[ \dot{m}_{p, bird, j} = LHP \cdot \frac{q_{LHP, m, bird, j}}{1000.2450} \quad (19) \]

Substituting Equations 13 to 17 in Equation 12 and assuming that the densities of the air that goes in and out the CV are constant and calculated from \(t_{db, j-1}\), one can obtain the humidity ratio (\(W_{j}\)) for the air that leaves the CV by Equation 21.

\[ W_{j} = \frac{\dot{m}_{PAD, j} \dot{V}_{PAD} W_{PAD} + \dot{m}_{inf, j} \dot{V}_{inf, j} W_{inf, j} + \dot{m}_{i-j} \dot{V}_{i-j} W_{i-j} + \dot{m}_{j} \dot{V}_{j} W_{j}}{\rho_{j} \dot{V}_{j} W_{j}} \quad (21) \]

The methodologies presented by Albright (1990) and Wilhelm (1976) were used for the calculation of the remaining psychrometric variables. Bisection method (Chandra & Singh, 1995) was used to determine the wet-bulb air temperature \(t_{wb, j}\).

**Indexes of the Thermal Environment and Productivity of the broilers**

The thermal environment is evaluated in every CV through temperature-humidity indexes (THI), developed by Thom (1959) or DeShazer & Beck (1988), Equations 22 and 23, respectively.

\[ THI_{Thom, j} = t_{db, j} + 0.36 \cdot (t_{wb, j} + 41.5) \quad (22) \]

\[ THI_{DB, j} = 0.6 \cdot t_{db, j} + 0.4 \cdot t_{wb, j} \quad (23) \]

For the evaluation of the productivity, it was used the environmental thermal index for productivity of broiler chickens (TEIbc), Equation 24, proposed by Medeiros et al. (2005), in which values between 21 and 24 (comfortable) are associated to the maximum productivity; between 25 and 27 (moderately comfortable), bird weight loss around 1 to 5%; between 28 and 30 (discomfort), loss of 5.1 to 15%; between 31 and 34 (extremely discomfort), loss of 15.1 to 30%; and for values above of 35 (dangerous), loss of 30.1 to 87%.

\[ TEI_{bc, j} = 45.6026 - 2.3107 \cdot t_{db, j} - 0.3683 \cdot RH_{j} + 9.7092 \cdot V_{j} + 0.05492 \cdot t_{wb, j}^{2} + 0.00121 \cdot RH_{j}^{2} + 0.65329 \cdot t_{db, j}^{2} + 0.012968 \cdot t_{wb, j} - 0.300928 \cdot t_{wb, j} - 0.05952 \cdot RH_{j} - V_{j} \quad (24) \]

**Physiological Responses**

Prediction of the main broiler’s physiological responses (surface temperature, \(t_{si}\); rectal temperature, \(t_{ri}\) and respiration rate, RR) in every CV can be estimated by the Equations 25, 26 and 27, respectively, proposed by Medeiros (2001). These equations were developed for Avian Farm broiler chickens, with age of 22 to 42 days old for \(t_{wb, j}\) ranging from 16°C to 36°C, RH of 20% to 90% and mean air velocity of 0.0 to 3.0 m s\(^{-1}\). Determination coefficients of .87, 0.96 and 0.92 were found for \(t_{wb, j}\), \(t_{db, j}\) and RR, respectively.

\[ t_{si, j} = 37.202431 - 0.021789 \cdot t_{wb, j} - 0.020151 \cdot RH_{j} - 0.240519 \cdot V_{j} + 0.0033517 \cdot t_{wb, j}^{2} - 0.00016 \cdot RH_{j}^{2} - 0.021473 \cdot V_{j}^{2} + 0.011687 \cdot t_{wb, j} \cdot RH_{j} - 0.011574 \cdot t_{wb, j} \cdot V_{j} - 0.00066 \cdot RH_{j} \cdot V_{j} \quad (25) \]
Productive Indexes

Productive indexes of broilers in every CV also can be estimated using the equations proposed by Medeiros (2001), being the Equation 28 attributed for mean daily feed intake (DFI, g/day), Equation 29 for mean daily weight gain (DWG, g/day), and Equation 30 for men feed conversion (FC, g/g), with coefficients of determination of 0.91, 0.89 and 0.72, respectively.

\[
\text{DFI}_i = 46.102818 - 0.425395 t_{dbi} - 0.031012 \cdot RH_{i} + 0.118907 V_i + 0.009092 t_{dbi}^2 + 0.00013 RH_{i}^2 + 0.0263 V_i^2 + 0.000893 t_{dbi} \cdot RH_{i} - 0.0006944 t_{dbi} \cdot V_i - 0.0000661 RH_{i} \cdot V_i
\]  
(26)

\[
\text{DWG}_i = 28.963697 + 11.306258 t_{dbi} + 0.030955 RH_{i} - 6.80328 V_i + 0.25476 t_{dbi}^2 + 0.002513 RH_{i}^2 + 1.3084 V_i^2 - 0.01389 t_{dbi} \cdot RH_{i} + 0.24676 t_{dbi} \cdot V_i - 0.02579 RH_{i} \cdot V_i
\]  
(27)

\[
\text{FC}_i = -0.000265 + 0.003704 t_{dbi} - 0.000258 RH_{i} - 0.049766 V_i - 0.00004651 RH_{i} - 0.000092 t_{dbi} \cdot RH_{i} - 0.0007484 \cdot V_i^2 + 0.00004651 RH_{i} - 0.00007064 t_{dbi} \cdot RH_{i} - 0.00002655 V_i - 0.0000661 RH_{i} \cdot V_i
\]  
(28)

Data Collection for Model Validation

For the model validation, data were collected in a commercial broiler house, located in middle of Goiás State in the central area of Brazil (16º 01' S of latitude, 49º 48' W of longitude, and 661 m of altitude), between May 28th, 2005 and May 30th, 2005, from 10:00 AM to 06:00 PM.

Scheme of the broiler house equipped with tunnel ventilation system (negative pressure) and evaporative cooling pad and misting systems are illustrated in Figure 1. Ten exhaust fans (64,55 m³/h) located in one of the extremities (Figure 1), suctioning the outside air through a inlet opening, four windows and evaporative cooling pad, with volumetric air flows equal to 13.43 m³/s, 17.64 m³/s and 11.97 m³/s, respectively, and, air infiltration through the curtains (15.76 m³/s) and through a lateral door (4.26 m³/s), located at 66 m away the same extremity.

Two evaporative cooling systems used to act concomitantly: a) evaporative cooling pad, with two 10.70 x 1.80 m panels, and, b) misting system, comprised of seven transversal lines located at the distances of 15, 25, 35, 45, 60, 80 and 110 m from the extremity opposite to the exhausters, being each line a set of ten water emitters with volumetric water flow of 6 L/h per emitter. Broiler chickens of Cobb breed at 37±1 d of age and 2.40±0.13 kg of body mass grown with a lodging density (LD) of 14 birds/m², for a total of 21,000 birds used in this study.

![Figure 1. Schematic top view of the broiler house showing equipment distribution and points of data measurement. Unit: m.](image-url)

For data collection, the house was divided in four sections (located at 30, 60, 90 and 120 m from opposite of exhausts fans), where the measurements were taken (Figure 1). The measurements of the air velocity inside the house, at the heights of 0.30 m, 1.00 m and 1.70 m from the floor, at the center and at the extremities (0.30 m apart from curtains) of each section were obtained by using a digital vane anemometer (±3%), with five repetitions. The same anemometer was used to measure the local atmospheric pressure (±0.3 kPa). Five portable sensors/recorders (± 3%), preprogrammed to collect data in intervals of one minute, were used in order to obtain t_{db} and relative humidity (RH) from the external environment and from...
the center of each section, at the height of 0.30 m from the floor. To validate the model only the data measured after at least 5 min of non interrupted functioning of evaporative cooling system were used.

Results and Discussion

The performed simulations for model validation showed that the average standard errors between the simulated and measured values for \( t_{db} \) and RH were 0.73°C and 5.43%, respectively. For the remaining variables, the values calculated by equations 18 to 26 using \( t_{db} \) and RH data simulated and measured by the \( t_{db} \)/RH sensors, were compared. Thus, the average standard errors found for \( THI_{thom} \), \( THI_{D&B} \), TEIbc, DFI, DWG, FC, \( t_s \), and RR were respectively equal to 0.65, 0.28, 0.57, 1.77 g/day, 1.27 g/day, 0.03 g/g, 0.04°C, 0.17°C and 2.15 breaths per min. Figure 2 shows the average standard errors for the studied variables in each section of data collect.

![Figure 2. Mean standard errors in four measurement sections inside broiler house (30, 60, 90 and 120 m) for studied variables.](image)

Simulated variables throughout an broiler house considering a specific thermal condition as an example (\( t_{db,a} = 24.4°C \) e \( RH_o = 74% \)), which was compared to field data measured at four sections along to the broiler house. Overall, a good fit of model was verified, with \( r^2 \) of 0.998, 0.890, 0.996, 0.978, 0.885, 0.998, 0.982, 0.972 and 0.684 for \( t_{db} \), RH, \( THI_{thom} \), \( THI_{D&B} \), \( t_s \), \( t_r \), DFI, DWG e FC, respectively. \( THI_{thom} \) assumed values lower than 74 until 43 m apart from the house entrance (close to the pad system), characterizing thermal comfort conditions to the birds. For the remaining house portion, \( THI_{thom} \) stayed between 74 and 79, indicating alert situations for broiler producers and danger for the birds, according to NWSCR (1976).

Thermal environment variations throughout the broiler house were observed. Results showed that the first half of broiler house is the best region for the broiler chicken to grow as compared to the other half. Differences between the mean values of \( t_{db} \), RH and \( THI_{thom} \) for the initial 25 m (1/5 of area) and for the final 25 m were 2.7°C, 7.0% and 3.0, respectively. For the physiological responses, the variation between the regions of the housing is unexpressive, once an increase of RR of until 6 breaths/min is acceptable when compared to the values observed when the chicken is subjected to acute heat stress, such as 250 breath/min (Ferreira, 2005). The same was observed for \( t_r \) which a negligible variation of 0.2°C was verified. Mount (1979) affirm that rectal temperatures ranging from 41.2°C to 42.2°C are considered as thermal comfort. For productive performance, the highest differences for FI, DWG and FC were 4.8 g/d, 3.2 g/d and 0.05 g/g, respectively.

According to the results, the 2.4 kg broiler chickens lodged at 14 chickens per m² in the first 1/5 of broiler house (close the pads) with 125 m of length presented a DFI of 20 kg smaller than the chickens at the end of house (close to the exhaust fans). Similarly, the DWG in the first 1/5 of broiler housing was 13.4 kg/day greater than at the end.

Conclusions

The mathematical model developed, considering heat and mass exchanges in steady-state, showed adequate to simulate the thermal environment and the main physiological and productive performance responses of broiler chickens grown in acclimatized house equipped with positive tunnel ventilation system.

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References


**Nomenclature**

\[ A_{i,j} \] Associate area at the control volume (CV) and structural component of house, m²;

\[ A_{i,\text{floor}} \] Floor area in the ith CV, m²;

\[ \beta_{i} \] Fraction of misting rate that evaporates in the ith CV;

\[ BM \] Body mass of birds, kg;

\[ DF_{i} \] Daily feed intake to each bird in the ith CV, g dia⁻¹;

\[ DW_{i} \] Daily weight gain to each bird in the ith CV, g dia⁻¹;

\[ f \] Cooling efficiency of the system, dimensionless;

\[ F_{i} \] Fator de ganho de calor no ith CV (obtido experimentalmente), W m⁻¹ K⁻¹;

\[ FC_{i} \] Feed conversion to each bird in the ith CV, g g⁻¹;

\[ h_{fg} \] Latent heat of vaporization of water at indoor wet bulb temp. in the ith CV, J kg⁻¹;

\[ LD \] Lodging densities, birds m⁻²;

\[ LHP \] Percentage of sensible heat productive from bird in latent form, %;
LT Light code, 0 for 100%-time light, 1 for 100%-time darkness, and between 0 and 1 for partial light/darkness during the hour, determined by: \( LT = \frac{LOT}{100} \times \min(1, \max(0.02, 6.02 - Ir)) \), where LOT is the light off time for the hour, %, and Ir is the intensity of outside solar radiation, W m\(^{-2}\).

\( m_{\text{water}} \), \( m_{\text{inf}} \) Mass flow rate of water issuing from misting nozzles in the \( i \)th CV, kg s\(^{-1}\);

\( m_{\text{bird}} \) Mass flow rate of air produced by bird into the \( i \)th CV, respectively, kg s\(^{-1}\);

\( m_{\text{PAD}} \) Mass flow rate of air carried into the building through pad cool. ventilation, kg s\(^{-1}\);

\( m_{\text{mist}} \) Mass flow rate of air carried to \( i \)th CV due water application through misting system, kg s\(^{-1}\);

\( m_{\text{out}}, m_{\text{in}} \) Mass flow rate of air carried to into and to out of \( i \)th CV through ventilation, respectively, kg s\(^{-1}\);

\( P \) Housing perimeter in the \( i \)th CV, m;

\( \rho_i, \rho_{i-1} \) Air density in the actual CV (\( i \)) and previous CV (\( i-1 \)), kg m\(^{-3}\);

\( \rho_{\text{PAD}} \) Air density that pass through the pad cooling, kg m\(^{-3}\);

\( \rho_{\text{water}} \) Water density, kg m\(^{-3}\);

\( q_{e,i} \) The rate of conversion of sensible heat to latent within the \( i \)th CV, W;

\( q_{f,i} \) Sensible heat transfer to the floor of the building primarily at the perimeter in the \( i \)th CV, W;

\( q_{s,i} \) Sensible heat gain from “mechanical” sources such as motors and lights in the \( i \)th CV, W;

\( q_{\text{bird}}, q_{\text{water}} \) Sensible heat gain from bird within the \( i \)th CV, respectively, W kg\(^{-1}\) and W;

\( q_{\text{sun}} \) Sensible heat gain from the sun in the \( i \)th CV, W;

\( q_{\text{in}}, q_{\text{out}} \) The sensible heat contained in the ventilation air entering and leaving the \( i \)th CV, respectively, W;

\( q_{\text{water}} \) Sensible heat transfer through the house structure for the \( i \)th CV; i.e., wall, ceiling, doors, etc., W;

\( RH_i, RH_{i-1} \) Relative humidity of air in the actual CV (\( i \)) and previous CV (\( i-1 \)), %;

\( t_{db,i}, t_{db_{i-1}} \) Dry-bulb temperature in the actual CV (\( i \)) and previous CV (\( i-1 \)), °C;

\( t_{pad}, t_{pad,i} \) Pad cooling temperature, °C;

\( t_{dp} \) Dew point temperature in the \( i \)th CV, °C;

\( THI_{\text{Thom}}, THI_{\text{DeShazer & Beck}} \) Temperature-humidity index developed by Thom (1959) in the \( i \)th CV;

\( TVS \) Tunnel ventilation system;

\( V_{\text{water}}, V_{\text{PAD}} \) Volumetric flow rate of water from misting nozzles, m\(^3\) s\(^{-1}\);

\( V_{\text{in}}, V_{\text{out}}, V_{\text{inf}} \) Volumetric flow rate of air that infiltrate in the \( i \)th CV, m\(^3\) s\(^{-1}\);

\( V_{\text{air}}, V_{\text{air in}}, V_{\text{air out}} \) Volumetric flow rate of air in the actual CV (\( i \)) and previous CV (\( i-1 \)), m\(^3\) s\(^{-1}\);

\( W_{\text{water}}, W_{\text{air}} \) Humidity ratio of air in the actual CV (\( i \)) and previous CV (\( i-1 \)), kg kg\(^{-1}\);
$W_{air}$ Humidity ratio of air that infiltrate in the $i^{th}$ CV, kg kg$^{-1}$.

$W_{PAD}$ Humidity ratio of air through pad cooling, kg kg$^{-1}$. 