SIMULATION COMPARISON OF EVAPORATIVE PADS AND FOGGING ON AIR TEMPERATURES INSIDE A GROWING SWINE BUILDING

P. Panagakis, P. Axaopoulos

ABSTRACT. Evaporative pads and fogging were compared with regards to resulting air temperatures inside a growing swine building and reduction of apparent growing swine heat stress. Four strategies were studied via simulation, namely: strategy 'a' = no cooling, strategy 'b' = use of evaporative pads, strategy 'c' = use of fogging with the same amount of water evaporating as within the evaporative pads, and strategy 'd' = use of fogging with the necessary water evaporating so as to result in the same intensity of heat stress as strategy 'b'. Indices such as the THI, the number of hours that the THI was above 85, and the duration and intensity of heat stress were used. Among all, strategy 'b' was considered the most effective, because it resulted in smaller daily inside dry–bulb temperature variation, maximum reduction of apparent heat stress intensity, and lower total consumption of water.

Keywords. Evaporative pads, Fogging, Heat stress, Simulation, Swine housing.

It is well documented (Curtis, 1985) that pigs are relative sensitive to high environmental temperature when compared to other species of farm animals. The major reason for their limited capacity to cope with high environmental temperatures is their inability to sweat (Mount, 1979). Several studies (Bond et al., 1959; Roller and Goldman, 1969; Nichols et al., 1982; Nienaber et al., 1987; Lopez et al., 1991; Huynh et al., 2005) have shown that elevated environmental temperatures are among the most important parameters, others being the extent of skin wetness, stocking density, air speed at pig level, etc., that cause minor or severe heat stress problems to swine and consequently hinder their growth performance and impede their welfare.

Evaporative cooling of ventilating air has long been recommended (MWPS–34, 1990) as an effective means to increase growing swine comfort under hot weather conditions. Two popular methods of evaporative cooling are evaporative pads and fogging (i.e., use of fine mist to cool the inside air temperature). Bridges et al. (1992) used a fogging strategy that was initiated above 25°C and determined the temperature inside a growing–finishing unit as being equal to the outside wet bulb temperature + 2°C. In an earlier study, Gates et al. (1991) compared evaporative pad cooling with fogging for growing–finishing swine and concluded that both systems compare favorably with regards to minimizing the inside temperature humidity index (THI). However, the authors arrived at this conclusion under the assumptions of negligible conduction heat losses, no animal heat production, and negligible solar heat gain. To our belief, these three assumptions, along with the conclusion reached by Axaopoulos et al. (1992) that the THI does not appear to be the most appropriate index for describing swine heat stress under Greek summer (May to September) conditions, considerably mask the conclusion reached. In addition, both studies comparing the evaporative pads system with fogging provided no information with regards to the water evaporating per animal. Unfortunately, no literature exists on this issue. Therefore, the objective of this study was to compare, via simulation, the effects of evaporative pads and fogging on air temperatures inside a growing swine building and on the apparent heat stress reduction of growing swine, taking into account:

- All the energy inputs associated with the heat and moisture balance of a growing swine building.
- Not only the THI, but also indices such as the number of hours that the THI exceeds 85 and the duration and the intensity of heat stress.

EVAPORATIVE PADS VS. FOGGING

Both the evaporative pads and the fogging operated (fig. 1) when the inside dry–bulb temperature exceeded the upper critical temperature (UCT), which was calculated to be 26.1°C (Bruce, 1981), and the inside relative humidity was not above 80% (Bridges et al., 1992). The following time–dependent equations were used to calculate the dry–bulb temperature and relative humidity inside the swine building:

\[ \sum (M_a C_a) \frac{dT_i}{dt} = Q_a + Q_b + Q_f + Q_s - \gamma \cdot Q_m \]  \hspace{1cm} (1)

\[ \rho_i V_i \frac{dW_i}{dt} = m_a \cdot (W_o - W_i) + W_f + \delta \cdot W_m \]  \hspace{1cm} (2)
ENERGY INPUTS

**Pig Sensible and Latent Heat Production**

Pigs are homoeothermic and strive to keep their body temperature at 39°C through the control of total heat dissipation exchange with their environment (Mount, 1968). Total heat dissipation is the sum of sensible and latent heat production. Values of both sensible and latent heat production are calculated using individual animal heat production (measured experimentally in environmental chambers at 20°C) and the influence of various housing factors such as relative humidity, flooring system, stocking density, feeding and watering systems, etc. (Sällvik and Pedersen, 1999).

Based on the analysis by Sällvik and Pedersen (1999), the ratio (r) between sensible heat (Φs, W) and total heat (Φtot, W) production at swine house level is initially calculated at each time step using the following equations (Pedersen, 2002):

\[
\Phi_{tot} = 1000 + 12 \cdot (20 - T_i) \quad (3)
\]

\[
\Phi_s = \Phi_{tot} \cdot 0.62 - 1.15 \cdot 10^{-7} \cdot T_i^6 \quad (4)
\]

Then, total heat production at individual animal level (\(Q_{tot-hp}, W\)) was calculated using the following equation (Pedersen, 2002):

\[
Q_{tot-hp} = 5.09m^{0.75} + [1 - (0.47 + 0.003m)] \cdot (v \cdot 5.09m^{0.75} - 5.09m^{0.75}) \quad (5)
\]

where \(m\) is the pig weight (kg), and \(v\) is the multiple of maintenance.

Next, a correction for the inside air temperature was applied by multiplying with the factor \(F_T = [1 + 4 \cdot 10^{-5}(20 - T_i)^3]\) (CIGR, 1984), while another multiplication with the number of housed animals gave an estimate of the total heat production in the swine building (\(Q_{tot-hp}, W\)). Finally, multiplying \(Q_{tot-hp}\) with \(r\) gave \(Q_s\) (i.e., the pigs’ sensible heat production used in eq. 1). The difference between \(Q_{tot-hp}\) and \(Q_s\) results in \(Q_l\), namely the pigs’ latent heat production (W). This value is converted to hourly moisture production (\(W_l\)) using the latent heat of water evaporation (\(h_f\), J/kg).

**Structural Heat Losses**

The heat flow through the building envelope (\(Q_b\)) is the sum of the heat fluxes entering or leaving each vertical wall, the roof, and the door. It can be expressed using the concept of sol–air temperature as follows:

![Conceptual representation of the psychrometric chart showing the operation of evaporative pads and fogging (system off in the shaded areas).](image-url)
\[ Q_b = \sum U_{bi} A_{bi} (T_i - T_{sa,i}) \]  

where \( U_{bi} \) is the overall heat transfer coefficient of each surface (W/m²·°C), \( A_{bi} \) is the surface area (m²) and \( T_{sa,i} \) is the sol–air temperature (°C).

The overall heat transfer coefficient \( (U_{bi}) \) can be calculated by applying the series thermal resistance theory, taking into account the composite layers that make up the envelope components. The sol–air temperature is calculated for each structural element using the following equation (ASHRAE, 1989):

\[ T_{sa,i} = T_o + \frac{\alpha I_{T,i}}{h_o} \]  

where \( T_o \) is the outside temperature, \( \alpha \) is the surface solar irradiation absorptance, \( I_{T,i} \) is the total solar irradiance on each envelope component surface (W/m²), and \( h_o \) is the external surface heat transfer coefficient (W/m²·°C). At any time step, the step calculates the total solar irradiance incident upon the surface of the four differently orientated walls (i.e., south, east, north, and west) and the roof. Its value depends on the orientation of each surface and the time of the year.

**Pen Floor Heat Losses**

The heat flow through the pen floor to the soil can be written in terms of the effective heat transfer coefficient \( (U_{ef}) \), which is defined by combining the heat transfer coefficients for pen floors \( (U_{fl}) \), pit walls \( (U_{pw}) \), and pit floor \( (U_{pf}) \) along the corresponding heat flow path to the ambient air. More specifically, the heat flow is computed from the following equation:

\[ \dot{Q}_f = U_{ef} A_{pf} (T_f - T_o) \]  

where \( U_{ef} \) is the effective pit heat transfer coefficient (W/m²·°C), and \( A_{pf} \) is the pen floor area (m²).

The effective heat transfer coefficient is calculated as:

\[ U_{ef} = U_{fl} + \frac{A_{pw} U_{pw} + A_{pf} U_{pf}}{A_{fl}} \]  

where \( U_{fl} \) is the overall heat transfer coefficient of pen floor (W/m²·°C), \( A_{pw} \) is the pit walls area (m²), \( U_{pw} \) is the pit wall heat transfer coefficient (W/m²·°C), \( A_{pf} \) is the pit floor area (m²), and \( U_{pf} \) is the pit floor heat transfer coefficient (W/m²·°C).

The pit of the swine building was considered as a below–grade wall structure. The pit wall heat transfer coefficient is determined from equation 10 (CIRA, 1982), which is used for the estimation of below–grade wall heat losses. This equation is in adequate agreement with the results of detailed two–dimensional transient computer modeling (Shipp and Broderick, 1981):

\[ U_{pw} = \frac{2\lambda}{\pi H} \ln \left( 1 + \frac{\pi H}{2R} \right) \]  

where \( \lambda \) is the soil thermal conductivity (W/m·°C), \( H \) is the pit depth (m), and \( R \) is the pit wall thermal resistance m²·°C/W).

The pit floor heat transfer coefficient is calculated by applying the series thermal resistance theory for the pit floor, the manure, and the pit air. The pen floor heat transfer coefficient is calculated using the slab thermal resistance between the swine building air and the pit air.

**Evaporative Pad Cooling**

Air leaving the evaporative pads is cooled, and its dry–bulb temperature \( (T_{ei}) \) is calculated at each time step using the following equation:

\[ T_{ei} = T_o - (T_o - T_{wo}) n_{ef} \]  

where \( T_{wo} \) is the outside wet–bulb temperature (°C), and \( n_{ef} \) is the evaporative pad efficiency (taken as equal to 0.80 in our case).

Due to adiabatic process, the wet–bulb temperature \( (T_{wie}) \) of air leaving the evaporative pads and entering the swine building is equal to the outside wet–bulb temperature \( (T_{wo}) \). Therefore, the humidity ratio of air leaving the evaporative pads and entering the swine building is calculated from \( T_{ei} \) and \( T_{wie} \).

**Fogging Cooling**

The fogging cooling term is calculated using the following equation:

\[ \dot{Q}_m = \beta \cdot W_{mf} \cdot h_{fg} \]  

where \( \beta \) is the fraction of water evaporating in the room, and \( h_{fg} \) is the latent heat of vaporization of water (kJ/kg). In our analysis, \( \beta \) was considered equal to 1.0 and constant under the assumptions (Bottcher and Baughman, 1990) that the: (1) very fine fog evaporated completely, (2) interior psychrometric conditions did not vary greatly or approach saturation, and (3) interior air velocities and fogging pressure remained relatively constant. It should be noted that if \( \beta \) is less than 1.0 then the amount of water used would increase accordingly.

**Ventilation Heat Losses**

At each time step, the values of the ventilation rate are determined using one of the following equations for temperature and relative humidity, respectively. The higher value of the ventilation rate is selected (Albright, 1990) and the corresponding ventilation heat loss term \( (\dot{Q}_V) \) is substituted into equation 1:

\[ \dot{Q}_V(T_{j}) = \frac{v_i \cdot (Q_j - T_{j})}{1000 \cdot c_p \cdot (T_i - T_j)} \]  

where \( Q_j \) is the temperature control ventilation rate (m³/s), \( v_i \) is the specific volume of the inside air (m³/kg), \( c_p \) is the specific heat of air (kJ/kg·°C), and \( T_j \) is either \( T_{ei} \) if the evaporative pads are on or \( T_o \) if fogging is on.

\[ \dot{Q}_V(RH_{j}) = \frac{v_i \cdot (W_{j} + W_{f})}{3600 \cdot (W_{i} - W_{o})} \]  

where \( Q_j \) is the relative humidity control ventilation rate (m³/s), and \( W_{j} \) is either \( W_{me} \) if the evaporative pads are on or \( W_{mf} \) if fogging is on.
Table 2. Average monthly real outside and predicted inside dry-bulb temperatures and relative humidity values (values in parentheses are standard errors).[a]

<table>
<thead>
<tr>
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<th>Outside</th>
<th>Strategy ‘a’</th>
<th>Strategy ‘b’</th>
<th>Strategy ‘c’</th>
</tr>
</thead>
<tbody>
<tr>
<td>Month</td>
<td>$T_{oi}$ ($°C$)</td>
<td>$R_{H_i}$ (%)</td>
<td>$T_{ii}$ ($°C$)</td>
<td>$R_{Hi}$ (%)</td>
</tr>
<tr>
<td>May</td>
<td>20.1 (0.17)</td>
<td>59.0 (0.49)</td>
<td>23.5 (0.15)</td>
<td>54.0 (0.40)</td>
</tr>
<tr>
<td>June</td>
<td>24.6 (0.17)</td>
<td>59.0 (0.60)</td>
<td>27.8 (0.16)</td>
<td>54.0 (0.47)</td>
</tr>
<tr>
<td>July</td>
<td>27.1 (0.13)</td>
<td>47.0 (0.41)</td>
<td>30.6 (0.09)</td>
<td>45.0 (0.34)</td>
</tr>
<tr>
<td>August</td>
<td>27.0 (0.14)</td>
<td>48.0 (0.47)</td>
<td>29.8 (0.11)</td>
<td>45.0 (0.37)</td>
</tr>
<tr>
<td>September</td>
<td>23.2 (0.15)</td>
<td>56.0 (0.51)</td>
<td>26.3 (0.14)</td>
<td>52.0 (0.41)</td>
</tr>
</tbody>
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[a] Strategy ‘a’ = no cooling, strategy ‘b’ = evaporative pads, and strategy ‘c’ = fogging with the same amount of water evaporating as within the evaporative pads.

**HEAT STRESS INDICES**

Four heat stress indices were used in the analysis, namely the THI (Roller and Goldman, 1969), the hours that the THI exceeded 85 (Fehr et al., 1983), and the duration and intensity of heat stress (Hahn et al., 1987).

The THI was calculated based on the definition given by Roller and Goldman (1969):

$$\text{THI} = 0.45 \cdot T_{wib} + 1.35 \cdot T_i + 32 \quad (15)$$

where $T_{wib}$ is the inside wet-bulb temperature ($°C$).

Panagakis et al. (1991) defined the duration of heat stress as the number of hours that the inside dry-bulb temperature exceeds the UCT, whereas the heat stress intensity was defined using the following equation:

$$I = \int \int_{T_i} \Delta T \cdot \Delta t \quad (16)$$

where $I$ is the heat stress intensity ($°Ch$), $\Delta T$ is the difference between the predicted inside dry-bulb temperature and the UCT ($°C$), and $\Delta t$ is the time during which animals are housed under temperatures higher than the UCT (h).

**RESULTS AND DISCUSSION**

Initially, simulation tests were run for the following strategies: ‘a’ = no cooling, ‘b’ = use of evaporative pads, and ‘c’ = use of fogging with the same amount of water evaporating as within the evaporative pads.

Average monthly real outside and predicted inside dry-bulb temperatures and relative humidity values are tabulated in table 2. When strategy ‘a’ or strategy ‘c’ is used, average monthly inside dry-bulb temperature exceeds the UCT during all months except May, whereas when strategy ‘b’ is used, this happens only during July and August. This finding alone is not very informative with regards to the heat stress likelihood of the animals. As Xin and DeShazer (1989) pointed out, a diurnally fluctuating temperature is equivalent to a steady temperature only if the fluctuating temperature is within the thermoneutral zone, which is bounded (Bruce, 1981) by a lower critical temperature and an upper critical temperature. Figure 2 clearly shows that this is not the case in our analysis, as for both strategy ‘a’ and strategy ‘b’ the diurnally fluctuating temperature often exceeds the upper critical temperature of the pigs, and therefore is not within the thermoneutral zone. Consequently, the four heat stress indices mentioned above and the water evaporating were
that when strategy 'a' is used, all heat stress indices are worse in tables 3 and 4, respectively. It becomes clear from table 3 estimated for each of the above three strategies and are shown in tables 3 and 4, respectively. It becomes clear from table 3 that when strategy 'a' is used, all heat stress indices are worse than with strategy 'b' or strategy 'c'.

A 3×5 ANOVA analysis (StatSoft, 2001) with strategy ('a' to 'c') and month (May to September) as the independent variables and each of the four heat stress indices as the dependent variable showed that strategy and month had a highly significant effect (P < 0.01). Use of post-hoc comparisons (Steel and Torrie, 1980) revealed that during each five-month period, strategy 'b' was the most effective (P < 0.05) compared to strategy 'a' and strategy 'c' (4 h vs. 76 h, respectively). However, strategy 'b' did not differ from strategy 'a' and strategy 'c' with regards to heat stress duration. A possible explanation is that use of strategy 'b' results in smaller inside evaporative pads, and To = ambient outside dry-bulb temperature).

Figure 4. Intensity of heat stress according to strategy used (strategy 'a' = no cooling, strategy 'b' = evaporative pads, and strategy 'c' = fogging with the same amount of water evaporating as within the evaporative pads).

Figure 3. Effect of strategy on inside dry–bulb temperature during Julian day 176 (strategy 'a' = no cooling, strategy 'b' = evaporative pads, strategy 'c' = fogging with the same amount of water evaporating as within the evaporative pads, and To = ambient outside dry–bulb temperature).

Table 3. Heat stress indices.[a]

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<td>70.6</td>
<td>12</td>
<td>168</td>
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<td>78.7</td>
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</table>

[a] Strategy 'a' = no cooling, strategy 'b' = evaporative pads, and strategy 'c' = fogging with the same amount of water evaporating as within the evaporative pads.
A $2 \times 5$ ANOVA analysis (StatSoft, 2001) similar to the above compared strategy 'b' and strategy 'd' from May to September. It was shown again that strategy and month were highly significant ($P < 0.01$). Post-hoc comparisons (Steel and Torrie, 1980) showed that during all five-month periods, strategy ‘d’ was more effective ($P < 0.01$) than strategy ‘b’ with regards to heat stress duration reduction (261 h vs. 483 h; 46% reduction). Figure 6 refers to the hottest day and shows the inside dry-bulb temperatures when strategy ‘b’ and strategy ‘d’ are used. It is interesting to note that when the ambient outside dry-bulb temperature peaks (14:00 h; 36.8 °C) strategy ‘b’ keeps the inside dry-bulb temperature at 29.5 °C lower. On the contrary, strategy ‘d’ can maintain the inside dry-bulb temperature only 3.1 °C lower (i.e., 33.7 °C). Nevertheless, figure 6, like figure 3, also shows that strategy ‘b’ results in smaller daily variation in inside dry-bulb temperature in comparison to strategy ‘d,’ thus, as explained above, preventing animals from “cooling down” during nighttime and apparently resulting in higher heat stress duration. Strategy ‘d’ was better ($P < 0.01$) with regards to the THI value (74.5 vs. 76.1) and similar ($P > 0.05$) with regards to the hours that the THI exceeded 85 (4 h vs. 7 h).

Finally, figure 7 shows that the total amount of water evaporating when strategy ‘b’ is used is 19.5 times lower compared to strategy ‘d’. This difference is obviously in favor of strategy ‘b’ and should not be overlooked, especially in areas with scarce water resources.

**CONCLUSIONS**

Simulation comparison of evaporative pads (strategy ‘b’) and fogging (strategy ‘c’ or ‘d’) on air temperatures inside a growing swine building, and reduction of growing swine apparent heat stress, proved that both cooling methods are significantly better compared to no cooling (strategy ‘a’). Among all, strategy ‘b’ was the most effective because it resulted in smaller daily inside dry-bulb temperature variation, maximum reduction of apparent heat stress intensity, and lower total consumption of water. Follow-up experimental studies are required to confirm these conclusions using experimental data from various types of swine buildings.

**REFERENCES**


