Evaporation of water to the ambient air is generally a cost-effective solution to alleviate heat stress but is also critically discussed due to its increasing effect on indoor humidity. The objectives of this study were to investigate the impact on indoor temperature, humidity, and ventilation rate using a high-pressure fogging system inside a mechanically ventilated research facility for fattening pigs. Data were sampled quasi-continuously throughout four fattening periods, including information on water consumption and energy use of the fogging system, as well as on the average weight gain of the pigs. It was found that for the pigs, the positive effects of the reduction of sensible indoor temperature exceeded the negative effects of the increased humidity, mirrored e.g. by a reduction of the temperature–humidity index (alert situations were reduced from 15.5% to 0.8%) and an increased weight gain of the animals during hot summer conditions. Main effects were achieved on days with mean daily outside temperatures above 14°C, during which water consumption of the fogging system averaged 4.9 l/day per pig.

The evaporation characteristics, such as the evaporative fraction as well as the time constant for reaching a steady state, were evaluated using a transfer-function model. Both were influenced largely by temperature and saturation deficit. While the evaporative fraction was 100% during steady state and 63% of steady state was reached within 65 s during warm and dry ambient conditions (28°C; 53% relative humidity RH), the evaporative fraction dropped to 89% and 65% for moderate (21°C; 69% RH) and cold/humid (13°C; 83% RH) indoor conditions, respectively, and the time to reach steady state was nearly doubled for the latter. The information about such evaporation characteristics is crucial for an accurate control of fogging and ventilation.
poultry, feature only a limited possibility to dissipate latent heat by vapourisation of water from the respiration tract and wetted skin (Bruce, 1981; Clark & McArthur, 1994). Curtis (1983) suggested that for pigs kept at an ambient air temperature of 30 °C, an increase of 18% relative humidity is equivalent to an increase in air temperature of 1 °C. Similarly, Huynh (2005) found that the upper critical temperature limit for an increased respiration rate and rectal temperature dropped by about 2 °C when relative humidity was raised from 50% to 80%. Evaporative cooling systems are an economical and effective solution to alleviate heat stress provided that the reduction in indoor temperature compensates at least for the increase in indoor humidity. Additional cooling for the animals can be provided at selected pen areas, e.g. by floor cooling or showers. According to investigations of Shi et al. (2006), the number of pigs lying in the sleeping area was affected considerably by floor temperature. With regard to evaporative cooling systems, latent heat loss of the animals can be improved by providing showers at separate pen areas and subsequent evaporation of water from the wetted body surface. Likewise, evaporation of small water droplets (misting/fogging) is utilised for increasing the latent heat content of the ambient air in large or full pen areas, which consequently improves sensible heat loss of the animals due to a higher temperature gradient. Equally important, indoor air cooling can counteract an increase of atmospheric emissions, such as NH₃ or CH₄ during hot summer conditions (Haeussermann et al., 2005, 2006a).

The fraction of the supplied water amount that evaporates while being airborne increases with decreasing water droplet sizes, and thus differs for fogging (pressure >5 MPa) and misting systems (pressure ≤ 5 MPa). One advantage of fogging systems is that evaporation can still take place when the indoor air humidity is relatively high, while evaporation from the respiration tract or the wetted body surface drops considerably at a high ambient air humidity.

Basically, evaporation of water molecules from the liquid phase to vapour occurs until a dynamic equilibrium is reached. The air then becomes saturated. For the evaporation, thermal energy is transformed adiabatically, i.e. no heat is added from or lost to the ambient air (Haas, 2002). The amount of sensible heat removed from the air equals the amount of heat absorbed by the water as latent heat of vapourisation, 2.4 kJ g⁻¹ [H₂O] at 25 °C. In consequence, the measurable dry-bulb temperature of the air is lowered and the relative humidity as well as the water vapour content of the ambient air rise. Theoretically, the air can be cooled to the wet bulb temperature if the change takes place under ideal adiabatic conditions (ASHRAE, 2003). According to Abdel-Ghany and Kozai (2006), the cooling efficiency ηₑ of a fogging system can be estimated by the extent to which the cooled air approaches the theoretical thermodynamic wet-bulb
temperature of the un-cooled air, recalculated by energy and mass balance simulation during the operation of fogging.

The evaporative fraction and time constant for evaporation of a fluid mainly depend on saturation deficit, water vapour partial pressure, and fluid surface in contact with the ambient air. These factors, however, are also influenced by the following variables (Lilequist & Cehak, 1984; Luo et al., 1994; Haas, 2002): ambient temperature, relative ambient humidity, quality and temperature of the liquid, droplet radius, and air movement.

As observed from the equation of Clausius-Clapeyron, the saturation deficit, i.e. the difference between the water vapour partial pressure and the saturation pressure \( p_{sat} \) of the ambient air, increases in an exponential relation with the ambient air temperature \( T \), with \( p_{sat} \sim e^{-1/T} \) (Haas, 2002).

Within a curved surface, the bonding forces rise, the larger the droplet radius becomes (Lilequist & Cehak, 1984). Small droplets in fog range (droplets less than 40 \( \mu \)m in diameter, ASHRAE 2003) evaporate faster and still at a higher saturation pressure than large droplets or water that features a plane surface. Moreover, the time in which a water droplet will remain airborne increases, due to frictional forces, with decreasing droplet radius, and allows complete evaporation of small droplets prior to being transported to the outside or nearby surfaces.

In general the droplet concentration is highest close to the centrelines of the nozzles. Upright positioned nozzles and an artificial upward air stream by using additional fans optimised the drop distance of the water droplets and increased the evaporative fraction in investigations of Toida et al. (2006). According to investigations of Sidahmed et al. (2005), the transport of droplets with a diameter below 60 \( \mu \)m was affected considerably by friction and buoyancy. In contrast, Wrachien and Lorenzini (2006) found that frictional forces increased the evaporative fraction especially at large droplets with diameters above 1 mm.

The droplet radius is highly influenced by the system pressure. Using low-pressure (misting) systems, bigger droplet radii are reached, and benefits are obtained primarily by wetting the skin of the animals and subsequent vapourisation of water from their body surface. According to investigations of Timmons and Baughman (1983), as well as Wilson et al. (1983), the evaporative fraction \( \mu \) varied between 23% and 57% when the system pressure ranged from 0.28 to 3.4 MPa (Bottcher et al., 1991). Temperature reduction was up to 4.4 °C if nozzles were placed indoors as well as in the incoming air (Wilson et al., 1983).

High-pressure (fogging) systems realise in general a higher evaporative fraction, whereby primarily the latent heat content of the ambient air, and hence sensible heat loss of the animals rises. Wetting the litter and floor surfaces can be avoided if the system is dimensioned and controlled properly, which avoids increasing \( \text{NH}_3 \) emissions during moderate temperature conditions (Haeussermann et al., 2004). An accurate control of high-pressure fogging systems can either be reached by varying the supplied water amount, hence evaporation rate, or by varying the ventilation rate (Timmons & Baughman, 1983; Gates et al., 1991a, 1991b, Arbel et al., 1999, 2003). Notwithstanding, the effective evaporative fraction and evaporation rate depend on several factors and can vary considerably among different housing systems or outside conditions. In order to improve the control of a fogging system, information on evaporation characteristics of the specific system is needed. Such knowledge can be used either for an accurate on-line control or for simulating optimised control settings.

There were two objectives for this study: (1) to investigate effects of a high-pressure fogging system during year-round measurements in a research pig facility on: (i) inside temperature, humidity and ventilation rate; (ii) water consumption and energy use; and (iii) average daily weight gain of the pigs; and (2) to model the evaporation process and estimate main influencing variables on evaporation characteristics, such as evaporative fraction and time constant for reaching steady state.

2. Materials and methods

2.1. Research facility and ventilation

Investigations were carried out in a research facility for fattening pigs in the 'Talgut Unterer Lindenhof', University of Hohenheim (Hartung, 2001). Two equally designed and comparable compartments (length of 7.80 m, width of 7.40 m, height of 3 m; Fig. 1) allowed parallel investigations during a total of four fattening periods (FP 1–4; February 2003 until July 2004). During each fattening period, the measurements started and ended at an average animal body weight of approximately 30 kg (week 2) and 105 kg per pig (week 14), respectively. All pigs were moved in and out at the same time. Each fattening period was subdivided into four measuring sections of 3 week durations (MS 1–4), whereby the ventilation strategies were changed after each measuring section.

The two compartments were equipped alternatively either with a sensor liquid-feeding system or a mash-feeding system (Haeussermann et al., 2004). Both compartments comprised two pens with 27 pigs per pen (54 pigs per compartment; 0.9 m² per pig), a slotted concrete floor with a slurry pit underneath each pen, a central control corridor between the pens, and a mechanical ventilation system (negative pressure, under-floor extraction) with one separately controllable ventilation fan (Fig. 1) (Haeussermann et al., 2004). Fresh air was supplied via two porous air inlet ducts per compartment, each arranged centrally above the animal area. The arrangement of the air inlet ducts and exhaust shaft, in combination with the low-impulse velocity of the entering air through porous material, allows in general the assumption of nearly homogenous airflow patterns in main parts of the compartments. Contrary to this arrangement, the airflow pattern and drop distance of the entering air would be affected considerably by the position of the air inlet valve if the air entered through wall inlets (Eren Özcan et al., 2005).

2.1.1. Fogging system

Two separately controllable fogging lines per compartment were utilised to cool down and to humidify the air (Fig. 1). These were placed: one inside the compartments, above the central control corridor at a height of 2.5 m (3 nozzles per pen), and a second one in front of the air inlets (2 nozzles per...
inlet). The inside fogging lines were operated at all ventilation rates, while nozzles at the air inlets were activated only for ventilation rates higher than 20% of the ventilation capacity. Water supply $F_W$ was 88.5 ml min$^{-1}$ nozzle$^{-1}$ (885 ml min$^{-1}$ per compartment when both fogging lines were operated). The pressure of the pump was fixed at 7 MPa. The fine nozzle orifices produced a cone stream of water droplets in fog range (~1 μm), directed into the incoming air stream either in front or below the pore channels (Fig. 1). In order to avoid clogging, three filters with pore sizes of 1 μm, 5 μm and 10 μm were placed upstream.

2.1.2. Ventilation and fogging control

Altogether, four ventilation strategies were tested randomly. These included a conventional temperature (and humidity) controlled strategy without and with fogging (Reference and Fogging 2, respectively). Two control strategies were further adjusted by additional control inputs (animal activity; CO2 indoor concentration, Haeussermann et al., 2004). They featured fogging and either an increased or a decreased ventilation rate (Fogging 1 and 3, respectively).

At all ventilation strategies, the ventilation rate $I$ was controlled by the feedback of the indoor temperature, based on a setpoint temperature $T_{set}$ and a linear control range (Reference, Fogging 1 and 2: 3 °C; Fogging 3: 6 °C). $T_{set}$ decreased from 25 °C to 23 °C during the first 5 days, from 23 °C to 21 °C until fattening day 16, from 21 °C to 20 °C until day 32, from 20 °C to 18 °C until day 56, and from 18 °C to 16 °C until fattening day 70. In accordance with the growth stage of the animals, the maximum ventilation rate $I_{max}$ was adjusted from approx. 5500 m$^3$h$^{-1}$ at the beginning to approx. 7500 m$^3$h$^{-1}$ at the end of each fattening period (32 and 43 air volume changes an hour, respectively). The two digital ventilation controllers were connected with independent temperature and humidity sensors for indoor air control.

Fogging started either when the maximum temperature $T_{max}$ was exceeded, i.e. when the indoor temperature $T_i$ rose to more than 1.5 °C above $T_{set}$, or when the indoor relative humidity $H_{ri}$ dropped below 50%. The system was operated in an interval with a fogging duty cycle of 4 min fogging-on and 3 min fogging-off. Maximum indoor humidity for onset of fogging $H_{rmax}$ was 80%, and fogging cycles were interrupted when $H_{ri}$ exceeded $H_{rmax}$. In order to avoid the system being permanently turned on and off, fogging started again with a hysteresis $\Delta H_{rmax}$ of ~10% after exceeding $H_{rmax}$. Minimum indoor temperature equalled $T_{set}$, i.e. the fogging system was not operated below this temperature. Fogging events were logged by the ventilation controllers with a time frequency of 1 min.

2.2. Measurements and calculations

A PC-controlled measuring and data-acquisition system (LabTech Notebookpro, Version 9.02) was utilised for continuous data-logging, obtaining one averaged value per minute. An overview of measured variables, frequencies, and their accuracies is provided in Table 1. Temperature and relative humidity were measured continuously with a combined sensor at one measuring point in the incoming air as well as inside the compartment at a height of 1 m (Fig. 1). Based on the measured indoor temperature $T_i$ in °C and the indoor relative humidity $H_{ri}$ in %, the temperature–humidity index $THI$ was calculated according to NWSCR (1976)

$$THI = (1.8 T_i + 32) - [0.55(H_{ri}/100) - ((1.8 T_i + 32) - 58)]$$

(1)

The indoor or outside humidity $X_{i/o}$ in g kg$^{-1}$ [dry air] was calculated from indoor and outside relative humidity $H_{ri/o}$ in % and saturation pressure $p_{sat}$ in Pa with

$$X_{i/o} = 0.622 \frac{H_{ri/o}}{100} \left(\frac{98000 - H_{ri/o}}{100 p_{sat}}\right)$$

(2)

where $p_{sat}$ was defined as

$$p_{sat} = 288.68 \left(1.098 + \frac{T_{i/o}}{100}\right)^{8.02}$$

(3)

The ventilation rate was measured continuously in the exhaust shaft using a calibrated measuring impeller (Table 1; Gallmann, 2003). Water consumption and energy use for humidifying were registered daily using a water meter and an...
energy meter, respectively; energy use for ventilating was registered weekly. The weight of the pigs $W_d$ and $W_e$ in kg was registered with a weighing scale before and at the end of each measuring section, respectively. The average daily weight gain $\Delta W$ in g d$^{-1}$ pig$^{-1}$ was calculated by

$$\Delta W = \frac{\sum_{i=1}^{n_p} W_{ei} - \sum_{i=1}^{n_p} W_{di}}{n_p \cdot t_{mes}} \times 1000,$$

where $n_p$ was the number of pigs per pen and $t_{mes}$ was the duration of the respective measuring section in days.

The fogging rate $X_f$ in kg kg$^{-1}$ [dry air] min$^{-1}$ was the product of the average water supply per nozzle $F_{w}$ in ml min$^{-1}$, the number of activated nozzles $N_f$ and the duration of fogging $t_f$ in s, divided by the volume of the compartment $V$ in m$^3$, the density of the air $\rho_f$ in kg m$^{-3}$, and the considered time step $t_f$ in s

$$X_f = \frac{F_{w}N_f t_f}{\rho_f V}.$$  \hspace{1cm} (5)

The calculation of the fogging rate required accurate recordings of the cycling intervals. These were mainly available during three separate data sets (Table 5), whereof data sets 1 and 2 were recorded during fattening period 2 (August 2003; $\Theta$; 28 °C) and fattening period 4 (June/July 2004; $\Theta$; 21 °C), respectively. The third dataset was recorded subsequent to the investigation period (November 2004; $\Theta$; 13 °C). Those measurements were performed in the empty building using a high measuring frequency of 1 s.

The evaporation rate $X_e$ in kg kg$^{-1}$ [dry air] min$^{-1}$ was calculated for each of the three data sets by the difference in indoor humidity $\Delta X_i$ in kg kg$^{-1}$ [dry air] during time step $t_f$ in s and the humidity balance in the compartment due to ventilation and moisture production, with

$$X_e = \left[ \frac{\Delta X_i}{M} - (X_o - X_i) \frac{\gamma_o - \phi_i}{\gamma_i \gamma_o - \phi_i} \frac{1}{\gamma_i V} \right] \times 60,$$  \hspace{1cm} (6)

where $X_i$, $X_o$ and $\gamma_i$, $\gamma_o$ were the humidity in kg kg$^{-1}$ [dry air] and the density in kg m$^{-3}$ of the inside and outside air, and $I$ was the ventilation rate in m$^3$/s$^{-1}$. The latent heat production of the animals $\phi_i$ in W pig$^{-1}$ for data set 1 and 2 was estimated as described in Berckmans et al. (1992), where $n_p$ was the number of pigs per compartment, and $\epsilon_i$ was the evaporation heat of water at inside temperature in J g$^{-1}$ [H$_2$O].

Assumptions for the calculation of the evaporation rate were: (i) perfect mixing of the air in the room; (ii) transport of evaporated water according to the airflow; (iii) no contribution of water droplets to the process as long as they were not evaporated; and (iv) negligible evaporation from other sources, like floors or slurry pits.

### 2.3. Evaporation model

Evaporation characteristics, such as the time constant to reach 63% of the evaporation rate at steady state and the evaporative fraction in % of the water amount supplied by fogging, were calculated by considering the evaporation process as a dynamic system described by a single-input, single-output transfer function. The general structure of the model was described in Young (1984) and Aerts et al. (2003):

$$y(k) = \frac{B(z^{-1})}{A(z^{-1})} u(k) + \xi(k),$$  \hspace{1cm} (7)

where $y(k)$ was the model output (evaporation rate $X_e$ in kg kg$^{-1}$ [dry air] min$^{-1}$ [Eqn (6)]), $u(k)$ was the model input (water supply for fogging $X_f$ in kg kg$^{-1}$ [dry air] min$^{-1}$ [Eqn (5)]), both at time $k$. The two polynomials $A(z^{-1})$ and $B(z^{-1})$ are given by

$$A(z^{-1}) = 1 + a_1 z^{-1} + a_2 z^{-2} + \ldots + a_l z^{-l},$$  \hspace{1cm} (8)

$$B(z^{-1}) = b_0 + b_1 z^{-1} + b_2 z^{-2} + \ldots + b_n z^{-n},$$  \hspace{1cm} (9)

where $a_l$, $b_j$ are the model parameters to be estimated; $z^{-1}$ the backward shift operator with $z^{-1} y(k) = y(k-1)$; and $na$, $nb$ the orders of the respective polynomials (Aerts et al., 2003). The additive noise $\xi(k)$ was assumed to be a zero mean, serially uncorrelated sequence of random variables with variance $\sigma^2$ and accounted for measurement noise, modelling errors, and effects of unmeasured inputs to the process. This included uncertainties in the estimation of evaporation from water supply, animals, and wet surfaces during the measurements as well as noise on the input due to interrupted fogging cycles when maximum humidity was exceeded. The model parameters were estimated using the simplified recursive instrumental variable approach (Young, 1984). The resulting models were evaluated by means of the coefficient of determination $R_p^2$, the Young Identification Criterion YIC, and the standard error of the model parameters (Table 4). The agreement of the simulated and calculated evaporation rate

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Table 1 – Overview of measured variables, measuring principles, frequencies, and accuracies

<table>
<thead>
<tr>
<th>Variable</th>
<th>Measuring principle</th>
<th>Measuring range</th>
<th>accuracy</th>
<th>Measuring frequency</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air temperature (outdoors and indoors)</td>
<td>PT 100; resistance$^a$</td>
<td>−40 °C to 85 °C</td>
<td>±1 °C</td>
<td>1 min</td>
</tr>
<tr>
<td>Relative humidity (outdoors and indoors)</td>
<td>Capacitive$^a$</td>
<td>0–100%</td>
<td>±1%</td>
<td>1 min</td>
</tr>
<tr>
<td>Ventilation rate (exhaust shaft)</td>
<td>Measuring impeller$^b$</td>
<td>200–1000 m$^3$ h$^{-1}$</td>
<td>±20 m$^3$ h$^{-1}$</td>
<td>1 min</td>
</tr>
<tr>
<td>Animal weight</td>
<td>Weighing scale</td>
<td>0–300 kg</td>
<td>±0.5 kg</td>
<td>3 weeks</td>
</tr>
<tr>
<td>Water consumption (fogging system)</td>
<td>Water meter</td>
<td>unlimited</td>
<td>±0.11</td>
<td>1 day</td>
</tr>
<tr>
<td>Energy use (ventilation fan)</td>
<td>Energy meter</td>
<td>unlimited</td>
<td>±0.1 kW h</td>
<td>1 day</td>
</tr>
<tr>
<td>Energy use (high pressure pump)</td>
<td>Energy meter</td>
<td>unlimited</td>
<td>±0.1 kW h</td>
<td>1 week</td>
</tr>
</tbody>
</table>

---

$^a$ HygroClip, Rotronic Messgeräte GmbH, Ettlingen, Germany.

Table 2 – Indoor temperature, relative humidity, and ventilation rate without (Reference) and with evaporative indoor air cooling (Fogging 1–3) throughout four fattening periods

<table>
<thead>
<tr>
<th>Ventilation strategy (measuring days)</th>
<th>Reference (81 days)</th>
<th>Fogging 1\textsuperscript{a} (87 days)</th>
<th>Fogging 2\textsuperscript{b} (99 days)</th>
<th>Fogging 3\textsuperscript{c} (81 days)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Indoor temperature ($T_i$), °C</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mean</td>
<td>21.9</td>
<td>19.7</td>
<td>19.5</td>
<td>20.8</td>
</tr>
<tr>
<td>Inter-quartile range</td>
<td>19–24</td>
<td>18–21</td>
<td>18–21</td>
<td>19–22</td>
</tr>
<tr>
<td>Relative indoor humidity ($H_{ri}$), %</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mean</td>
<td>47</td>
<td>67</td>
<td>63</td>
<td>69</td>
</tr>
<tr>
<td>Inter-quartile range</td>
<td>42–53</td>
<td>55–77</td>
<td>53–73</td>
<td>57–71</td>
</tr>
<tr>
<td>Minimum–Maximum</td>
<td>22–73</td>
<td>33–100</td>
<td>28–100</td>
<td>35–99</td>
</tr>
<tr>
<td>Ventilation rate ($I_{pig}$), m\textsuperscript{3} h\textsuperscript{-1} pig\textsuperscript{-1}</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mean</td>
<td>79</td>
<td>72</td>
<td>67</td>
<td>56</td>
</tr>
<tr>
<td>Inter-quartile range</td>
<td>36–125</td>
<td>43–97</td>
<td>40–86</td>
<td>32–79</td>
</tr>
</tbody>
</table>

\textsuperscript{a} increased ventilation rate (temperature and activity controlled).
\textsuperscript{b} standard ventilation rate (temperature controlled).
\textsuperscript{c} reduced ventilation rate (doubled control range; CO\textsubscript{2} and temperature controlled).

The indoor temperature ($T_i$), measured during the investigation period, ranged from 15 to 36 °C without cooling (Reference), from 15 °C to a maximum of 30 °C using the evaporative cooling system with an increased or standard ventilation rate (Fogging 1 and 2), and from 16 to 31 °C applying evaporative cooling with a decreased ventilation rate (Fogging 3) (Table 2).

The main reduction of $T_i$ occurred at high temperatures, expressed by a reduction of the 75th percentile of $T_i$ by about 3 °C from 24 °C to 21 °C for Fogging 1 and 2 and by about 2 °C for Fogging 3. Temperature peaks as well as the variation of $T_i$ were clearly reduced by indoor air cooling. In consequence, the inter-quartile range of the indoor relative humidity $H_{ri}$ was shifted up from values between 42% and 53% towards values between 53% and 77% when using the cooling system, while the maximum value of $H_{ri}$ was shifted from 73% towards 100% (Table 2).

Main effects of evaporative indoor air cooling on $T_i$ were achieved during days with mean daily outside temperatures above 14 °C, and here mainly during daytime between 9:00 am and 9:00 pm (Fig. 2a). In this outside temperature range, the diurnal maximum value for $T_i$ was lowered by indoor air cooling on average by about 4 °C to 5 °C; maximum $T_i$ dropped by about 7 °C. $H_{ri}$ was in general rather low for ventilation without indoor air cooling during days with mean outside temperatures above 14 °C. Values below 40% $H_{ri}$ were measured especially during the afternoon (Fig. 2b). The reduction of diurnal peaks in $T_i$ for ventilation with evaporative indoor air cooling was mainly reached by counteracting these low values of $H_{ri}$ during daytime, hence evening diurnal fluctuations of $H_{ri}$ (Fig. 2). In consequence, $H_{ri}$ was kept closer to the optimum range of 50%–80% (CIGR, 1984).

A reduction of the ventilation rate $I$ due to evaporative indoor air cooling was mainly shown during times with moderate values of $T_{in}$ while no reduction of the maximum value of $I$ occurred at times during which $T_{in}$ was extremely high. The 75th percentile of the ventilation rate per pig $I_{pig}$ was 125 m\textsuperscript{3} h\textsuperscript{-1} pig\textsuperscript{-1} (39 air volume changes per hour) for ventilation without evaporative indoor air cooling (Reference, Table 2). According to the feedback of the indoor temperature,
I_pig was reduced to 97 and 86 m³ h⁻¹ pig⁻¹ (30 and 27 air volume changes per hour) for Fogging 1 and 2, respectively. The mean value of I_pig dropped from 79 m³ h⁻¹ pig⁻¹ to 72 and 67 m³ h⁻¹ pig⁻¹, respectively, while the maximum ventilation rate, limited by the control settings, was not reduced at all (Table 2). In general, the percentage reduction of I_pig was directly related to the difference between T_i and T_t, in relation to the set control range. The less the fogging system was able to keep T_i within the control range between T_set and the control setting T_max, the less I_pig was reduced. A wider control range in combination with varying the control inputs between temperature and CO₂ control (Fogging 3) did furthermore reduce the 75th percentile of I_pig to 79 m³ h⁻¹ pig⁻¹ but featured also slightly higher values of T_i (Table 2).

During mean daily outside temperatures above 14 °C, the ventilation rate dropped on average by about 22% for Fogging 1 and 2 and by about 33% for Fogging 3.

### 3.2. Water consumption and energy use

The measured water consumption for fogging averaged 4.91 l d⁻¹ pig⁻¹ during mean daily outside temperatures above 14 °C. In comparison, Panagakis et al. (1996) estimated a water consumption of 0.81 l h⁻¹ pig⁻¹ (19 l d⁻¹ pig⁻¹) using a misting system under Greek summer conditions. For outside temperatures lower than 14 °C, the measured water consumption was clearly lower: around 1.31 l d⁻¹ pig⁻¹. Hereby, humidifying was mainly utilised to increase the indoor humidity, in order to avoid dry indoor conditions below 50%.

The measured energy consumption of the used energy-saving fans averaged 0.024 kWh d⁻¹ pig⁻¹ without fogging and was reduced by about 25% to 0.018 kWh d⁻¹ pig⁻¹ when operating the fogging system. The mean daily energy consumption of the high-pressure pump for creating the fine fog averaged 0.073 kWh d⁻¹ pig⁻¹, thus was approximately three to four times higher than the energy consumption of the ventilation fans used. Nonetheless, the relation between energy-savings due to a reduced ventilation rate and additional energy-cost due to fogging depends clearly on the characteristics and utilisation of the respective components, such as type of fans and number of pigs per high-pressure pump. This relation can vary considerably and requires general investigations on an optimised utilisation and the respective energy consumption of high-pressure pumps.

### 3.3. Net temperature reduction and average daily weight gain

The applied specific feeding level (i.e. the amount of feed energy intake in MJ) in relation to energy for maintenance in MJ) decreased from 4.4 to 2.8 during each fattening period. Hence, the upper critical temperature for dry indoor conditions was approximately 24 to 26 °C (Bruce, 1981). Without fogging, this temperature threshold was exceeded on several days during summer, indicating the temporary heat stress for the animals. Thereby the reduction of the diurnal maximum value of T_i exceeded the negative effects of the simultaneously increased values of H_s on the physiological responses according to Curtis (1983) and Huynh (2005) (i.e. drop of the upper critical temperature for increasing respiration rate and rectal temperature) by approximately 2 to 4 °C (mean/max net reduction of diurnal peaks in T_i).

The achieved cooling effect was also confirmed by the calculated temperature-humidity index T_DH without cooling, 15.5% of the values were at least in an alert range (T_DH, 75–78), thereof 5% and 3% were dangerous (T_DH, 75–83) or to be labelled as emergency (T_DH, > 84), while with fogging only 0.8% of the values reached an alert range.

An influence of evaporative indoor air cooling on the average weight gain ΔW in g d⁻¹ pig⁻¹ was mainly found during July and August 2003, i.e. during the two measuring sections MS 2 and MS 3 in fattening period FP 2 (summer fattening period; Table 3). The average value of ΔW per measuring section during all fattening periods (FP 1–4) was 909 g d⁻¹ pig⁻¹ in MS 2 (fattening day 29–49) and 832 g d⁻¹ pig⁻¹ in MS 3 (fattening day 50–70). In comparison, ΔW in summer 2003 was reduced by about 47 g d⁻¹ pig⁻¹ in MS 2 considering ventilation with cooling (Fogging 3) and by about
134 g d⁻¹ pig⁻¹ by applying ventilation without cooling (Reference, Table 3). In the subsequent measuring section (MS 3), ΔW was reduced by about 46 g d⁻¹ pig⁻¹ without cooling — compared to the average value of ΔW in MS 3 — and was about 38 g d⁻¹ pig⁻¹ higher than the average value of ΔW in MS 3 when applying ventilation with cooling (Fogging 1). No reduction of ΔW was found at MS 1 and MS 4 in FP 2, during which cooling was applied in both compartments (Table 3). Although the compartments comprised different feeding systems, the courses of Ti and ΔW from MS 1 to MS 4 per compartment indicate a clear influence of Ti and evaporative indoor air cooling on ΔW. According to investigations of Nienaber et al. (1996), Quiniou et al. (2000), and Huynh (2005), the reduction in ΔW was presumably caused by a lowered voluntary feed intake during heat stress periods. Nevertheless, the average value of ΔW in FP 2, 830 g d⁻¹ pig⁻¹, did not differ significantly from the other fattening periods. Consequently, depressions during hot periods without indoor air cooling were probably compensated afterwards. Considering FP 1 to FP 4, ΔW was on average about 13 g d⁻¹ pig⁻¹ lower for ventilation without cooling than for ventilation strategies with cooling but did not differ significantly.

3.4. Influence of temperature and humidity conditions on evaporation rate

The general course of the evaporation process was separated into: (1) a dynamic increasing part, subsequent to the start of the fogging duty cycle; (2) steady-state conditions, featuring a constant evaporation rate; and (3) a dynamic decreasing part after the duty cycle stopped (Fig. 3). The information about the fogging rate and the model parameters, estimated separately for each of the three data sets by Eq. (7), was utilised for simulating the mean course of the evaporation rate Xe,sim. The model parameters as well as their identification criteria are listed in Table 4. The course of Xe,sim and the mean course of Xe per day period were recalculated into evaporation rate in ml min⁻¹ for presentation together with the average water supply in ml min⁻¹ in Fig. 3. The evaporation rate is shown in relation to the average length of the fogging duty cycle. In general, an average water supply of 885 ml min⁻¹ means full water supply during the respective time interval at all recorded fogging events [Fig. 3(a)], while frequent interruptions of fogging cycles were characterised by a lower average water supply, especially towards the end of the standard fogging duty cycle of 4 min [Fig. 3(b)]. No steady-state conditions were reached during the applied short duty cycles. Therefore, only the dynamic increasing course of the evaporation rate after the start and the dynamic decreasing course after the end of the duty cycles is demonstrated in Fig. 3. The fogging duty cycles featured nearly constant durations of 4 min during the hot and dry indoor conditions in August 2003 [Fig. 3(a)] but were frequently interrupted during the moderate indoor conditions in June/July 2004 [Fig. 3(b)]. These interruptions were mainly triggered by exceeding Hmax (80%). Mean values of Xe and Xe,sim and the root-mean-squared error RMSE between them, according to Eq. (10), are listed in Table 5.

Considering the fixed pressure and water supply in this investigation, the absolute amount of evaporating water, as well as the characteristics of the evaporation course, were largely influenced by the respective data set, and hence by Ti and H1 (Table 5). As indicated theoretically by Haas (2002), both played a crucial role in the evaporation of the water droplets because of the influence on the vapour pressure gradient between the droplet surface and the ambient air. The influence of the ventilation rate on the model accuracy was negligible, and consequently was not verified as an important input variable. The evaporative fraction f during steady state was about 100% for hot and dry indoor conditions but dropped down to 89% for moderate indoor conditions and to 65% for cold and humid indoor conditions [Fig. 3(a)–(c)]. The time constant tgc increased from 65 to 112 s when comparing mean indoor temperatures of 28 °C and 13 °C (Table 5).

A high accuracy of the estimated model parameters, with an R² of 0.84 and 0.88 between measured and simulated data (Table 4), was achieved for the hot period in August (normal stocking rate) as well as for the empty stable and high measuring frequency in November, respectively. Both data sets featured clearly defined, non-interrupted fogging duty

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**Table 3 – Average weight gain in relation to inside temperature and measuring section (MS 1–4) throughout four fattening periods (FP 1–4) and during summer 2003 (FP 2)**

<table>
<thead>
<tr>
<th>Measuring section</th>
<th>MS 1</th>
<th>MS 2</th>
<th>MS 3</th>
<th>MS 4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fattening day</td>
<td>8–28</td>
<td>29–49</td>
<td>50–70</td>
<td>71–91</td>
</tr>
<tr>
<td><strong>Fattening period 1–4 (February 2003 until July 2004; FP 1–4)</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mean inside temperature Ti</td>
<td>C ± standard deviation</td>
<td>21.5 ± 1.3</td>
<td>21.5 ± 3.5</td>
<td>19.9 ± 3.3</td>
</tr>
<tr>
<td>Average weight gain ΔW, g d⁻¹ pig⁻¹ ± standard deviation; [C1/C2]b</td>
<td>841 ± 74</td>
<td>909 ± 66</td>
<td>832 ± 41</td>
<td>727 ± 64</td>
</tr>
<tr>
<td><strong>Summer fattening period (June 2003 until September 2003; FP 2)</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Ventilation strategya</td>
<td>Fog 2/Fog 1</td>
<td>Ref/Fog 3</td>
<td>Fog 1/Ref</td>
<td>Fog 3/Fog 2</td>
</tr>
<tr>
<td>Mean inside temperature Ti</td>
<td>C ± [C1/C2]b</td>
<td>23.2/23.0</td>
<td>27.9/25.1</td>
<td>20.7/24.1</td>
</tr>
<tr>
<td>Average weight gain ΔW, g d⁻¹ pig⁻¹; [C1/C2]b</td>
<td>923/889</td>
<td>775/862</td>
<td>870/786</td>
<td>775/766</td>
</tr>
</tbody>
</table>

a Fog 1, Fog 2, Fog 3: ventilation with evaporative indoor air cooling (Fogging 1–3): Ref: ventilation without evaporative indoor air cooling (Reference).
b C1: compartment 1 (sensor-liquid feeding); C2: compartment 2 (mash feeding, ad libitum).
cycles of 4 and 2 min, respectively. Due to the high measuring frequency of 1 s in data set 3, the duty cycles were not distributed evenly between time 0 and 2 [Fig. 3(c)] but characterised accurately the supplied water amount per time interval. In comparison, frequent interruptions of the duty cycles as well as high variations in water supply and evaporation rate when comparing the single day periods occurred during the measuring period in June/July [Fig. 3(b)]. They resulted in a lowered accuracy of the model parameters in terms of $R_T^2$ (Table 4) as well as in a high variation of $t_{63}$

### Table 4 – Model parameters $a_j$ and $b_j$ and model evaluation criteriaa at separate data setsb

<table>
<thead>
<tr>
<th>Data set</th>
<th>$a_j$ (± SE)</th>
<th>$b_0$ (± SE)</th>
<th>$b_1$ (± SE)</th>
<th>AIC</th>
<th>YIC</th>
<th>$R_T^2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>-0.4000 (± 0.0058)</td>
<td>0.2514 (± 0.0047)</td>
<td>0.3030 (± 0.0074)</td>
<td>-1.1</td>
<td>-8.7</td>
<td>0.84</td>
</tr>
<tr>
<td>2</td>
<td>-0.6486 (± 0.0046)</td>
<td>0.2244 (± 0.0035)</td>
<td>0.0892 (± 0.0050)</td>
<td>-1.9</td>
<td>-7.3</td>
<td>0.80</td>
</tr>
<tr>
<td>3</td>
<td>-0.5995 (± 0.0418)</td>
<td>0.2254 (± 0.0210)</td>
<td>0.0502 (± 0.0325)</td>
<td>-3.2</td>
<td>-3.2</td>
<td>0.89</td>
</tr>
</tbody>
</table>

Mean indoor temperature $T_i$, °C: 28, 21 and 13; Mean relative indoor humidity $H_{ir}$, %: 53, 69 and 83 (data set 1, 2 and 3, respectively).

a SE, standard error; AIC, Akaike Identification Criterion; YIC, Young Identification Criterion; $R_T^2$, coefficient of determination according to Young.

b Data set 1: August 2003; data set 2: June/July 2004; data set 3: November 2004 (cf. Table 5).

### Table 5 – Evaporation characteristics at different climatic indoor conditions

<table>
<thead>
<tr>
<th>Data set</th>
<th>August 2003</th>
<th>June/July 2004</th>
<th>November 2004</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mean indoor temperature $T_i$, °C (min–max)</td>
<td>28 (24–31)</td>
<td>21 (18–27)</td>
<td>13 (11–15)</td>
</tr>
<tr>
<td>Mean relative humidity $H_{ir}$, % (min–max)</td>
<td>53 (32–83)</td>
<td>69 (48–85)</td>
<td>83 (66–100)</td>
</tr>
<tr>
<td>Minimum–maximum ventilation rate $I$, m$^3$ h$^{-1}$</td>
<td>5300–6700</td>
<td>1900–7700</td>
<td>1900–7700</td>
</tr>
<tr>
<td>Mean evaporation rate $X_e$, g [H$_2$O] kg$^{-1}$ [dry air] min$^{-1}$</td>
<td>2.48/2.48*</td>
<td>0.95/0.94*</td>
<td>0.46/0.47*</td>
</tr>
<tr>
<td>RMSEb</td>
<td>±0.53</td>
<td>±0.35</td>
<td>±0.18</td>
</tr>
<tr>
<td>Evaporative fraction during steady state $b$, %</td>
<td>100</td>
<td>89</td>
<td>65</td>
</tr>
<tr>
<td>Time constant $t_{63}$, s</td>
<td>65</td>
<td>—</td>
<td>112</td>
</tr>
</tbody>
</table>

a Measured/simulated value.
b RMSE: root-mean-squared error between measured and simulated values [Eq. (10)].
from 98 to 243 s per day period. Nevertheless, the variation of \( \beta \) from 85% to 96% per day period in this dataset provided a good estimation of the evaporated water amount per time step. Similarly to the results found by comparing the three data sets, Luo et al. (1994) reported a clear influence of temperature and relative humidity on the time that is needed for complete evaporation of water droplets.

As the control of fogging systems is based on the knowledge about the evaporation rate (Gates et al., 1991b), the information about the evaporative fraction and the time needed to reach steady-state conditions is one of the main input factors. Commonly, \( \beta \) is assumed to be 100% if model based investigations on fogging system are performed (Panagakis et al., 1996; Arbel et al., 1999, 2003). This might be correct for hot and dry climatic outside conditions but has to be restricted for moderate climates. Thereby, non-evaporating water might lead to wet and subsequently fouled surfaces and, in consequence, increase \( \text{NH}_3 \) volatilisation and emission from the facility (Haeussermann et al., 2005). As specific characteristics of housing and fogging systems, situated in different environments, differ in the way they influence evaporation dynamics, the on-line information received from continuous on-farm measurements can be utilised as input to the control system in order to simulate the evaporation rate, and thus to optimise fogging and ventilation control.

4. Conclusions

Ventilation strategies with and without utilisation of fogging for adiabatic indoor air cooling were compared with each other. Air temperature, relative humidity, and ventilation rate were measured with a frequency of 1 min in a mechanically ventilated experimental piggery in southern Germany throughout a total of four fattening periods. Data sampling included quasi-continuous information on water consumption and energy use as well as on the average daily weight gain of the pigs.

One objective of the study was to investigate effects of the tested fogging system on the measured variables. The second objective was to evaluate evaporation characteristics such as evaporative fraction at different ambient conditions and the duration for reaching steady state. This information is crucial for an accurate control of fogging and ventilating and can be utilised either on-line or for simulating optimised control settings. Evaporation characteristics were evaluated at three separate data sets using a transfer-function model.

Main temperature effects of fogging were achieved during days with mean daily outside temperatures above 14 °C, during which water consumption averaged on 4.91 d⁻¹ pig⁻¹. They resulted in an average reduction of diurnal peaks in indoor temperature by about 4 to 5 °C, clearly exceeding the negative effects of the increased indoor air relative humidity. Relative humidity below 40% was mainly avoided when using the fogging system.

A positive effect of fogging was found on the temperature-humidity index — alert situations were reduced from 15.5% to 0.8% considering all four fattening periods — and on the average daily weight gain of the animals during hot summer conditions.

The achieved temperature reduction resulted in a reduction of the 75th percentile of the ventilation rate and thereby in a reduction of the energy use of the ventilation fan by approximately 25%.

The evaporative fraction was 100% during steady state, while 63% of steady state were reached within 65 s during warm and dry ambient conditions (28 °C, 53% relative humidity RH). It dropped to 89% and 65% for moderate (21 °C, 69% RH) and cold/humid (13 °C, 83% RH) indoor conditions, respectively, and the duration to reach steady state was nearly doubled for the latter. These values were site-specific for the investigated facility and climatic outside conditions. Nevertheless, they allow to describe general influences on the evaporation course when using a fogging system in a mechanically ventilated animal facility.

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