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Research Paper

Modelled performance of energy saving air treatment devices to mitigate heat stress for confined livestock buildings in Central Europe

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ABSTRACT

Intensive pig and poultry production are predominantly performed in confined livestock buildings which are equipped with mechanical ventilation systems. The frequency of heat stress will increase due to climate change. Heat stress events are accompanied by performance depressions (e.g. daily weight gain, egg production, mortality, feed conversion rate). Consequently, appropriate air treatment devices can become necessary to optimise the indoor climate of confined livestock buildings because of a high inlet air temperature. In this study, we analysed the effects of three energy saving air treatment devices: (1) earth-air heat exchanger, (2) direct evaporative cooling by cooling pads, and (3) indirect evaporative cooling systems which combine evaporative cooling (e.g. by cooling pads) with a subsequent heat recovery system. All systems are compared to a reference ventilation system without air treatment, which is today's typical housing system. The results show that the earth-air heat exchanger (1) is the most efficient air treatment device. It eliminates heat stress and can also be used during wintertime to increase the inlet air temperature. The two adiabatic cooling systems (2) and (3) can reduce heat stress by about 90%. Cooling pads can lead to a high relative humidity of the inlet air between 75% and 100%, which can

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cause problems inside the livestock buildings, e.g. increasing the moisture content of the bedding material. The indirect cooling device can avoid this disadvantage at the expense of a reduced temperature reduction of the inlet air temperature and higher investment costs.
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1. Introduction

The majority of pigs and poultry in mid-latitudes are kept in confined livestock buildings (Robinson et al., 2011). A strong positive relationship between intensive livestock production and national gross domestic product has been shown worldwide (Gilbert et al., 2015). Intensive production systems in confined livestock buildings can be found predominantly in mid-latitudes (Robinson et al., 2011). Due to the high animal density, confined livestock buildings are usually equipped with mechanical ventilation systems fulfilling two major goals (1) to provide sufficient air quality during the winter season in combination with an inside air temperature close to the thermoneutral zone of the animals despite restricted ventilation rates, (2) to remove the sensible heat of the animals by high ventilation rates to avoid high indoor air temperatures during the summer season. As the optimum environmental temperature range for most farm animals lies considerably above the annual mean temperature, which is ~5 °C in North Europe (about 60°N) and ~15 °C in South Europe (about 40°N), the design of the buildings and the ventilation systems is aligned today to guarantee the lower limit of the thermal neutral zone of the animals. Therefore they are frequently termed *warm confinement livestock buildings* (Zulovich, 1993).

For pig and poultry farmers, economic losses can result from a reduction of feed efficiency, growth rate, egg and meat production, semen quality as well as infertility, reproductive disorders or increasing death rates (Dittrich, Wreford, Topp, Eory, & Moran, 2017; Fouad et al., 2016; Renaudeau et al., 2012; St-Pierre, Cobanov, & Schnitkey, 2003).

Miller, Howden, and Jones (2010) listed climate control measures inside buildings through natural air conditioning (i.e. evaporative cooling) as one out of several potential climate change adaptation options for intensive livestock production. In our model calculations, comparable adaptation measures of the ventilation system in confined livestock buildings are evaluated with respect to their potential to avoid or reduce heat stress for pigs and poultry.

Air treatment devices modify the thermodynamic properties of the inlet air. In principle, there are few technical constraints to guarantee a certain indoor climate, expressed by temperature and humidity. Limitations result from economic constraints due to high investment, energy and maintenance costs. Therefore we selected only systems which do not need supplementary energy for cooling and/or dehumidification: (1) earth-air heat exchange tubes (Bisoniya, Kumar, & Baredar, 2014; Tzaferis, Liparakis, Santamouris, & Argiriou, 1992), (2) direct evaporative cooling devices i.e. cooling pads (Renaudeau et al., 2012; Valiño, Perdigones, Iglesias, & García, 2010; Xuan, Xiao, Niu, Huang, & Wang, 2012), and (3) indirect evaporative cooling systems which combine evaporative cooling (e.g. by cooling pads) with a subsequent heat recovery system (Heidarinejad, Bozorgmehr, Delfani, & Esmaeelian, 2009; Sax, van Caenegem, & Schick, 2012; Struck, Külpmann, Sax, & Hartmann, 2014; van Caenegem, Sax, & Schick, 2012).

For each air treatment device, we model the inlet air condition behind the air treatment device in comparison to a ventilation system without air treatment as a reference. Interactions between animals, ventilation system and building are neglected to concentrate on the features of the air treatment devices. The efficacy of such devices can be evaluated by the thermodynamic state functions for sensible and latent heat (air temperature and an appropriate humidity measure like vapour pressure). However, these parameters have a limited relevance in relation to farm animals if considered separately, because thermoregulation is related to both of these two variables. Therefore, several established indices (for example the temperature humidity index THI) are selected to quantify the heat stress of animals. These parameters combine air temperature and humidity in one measure, taking into account the restrictions of the heat release of the animals, which vary with animal species (e.g. capability of sweating). The performance of each air treatment device is described by the exceedance of heat stress parameters beyond a preselected threshold value in terms of frequency of exceedance, area of exceedance (analogue to degree-hours) and frequency of consecutive exceedance of a certain duration.

The goal of the paper is to estimate their appropriateness by establishing whether the selected energy saving air treatment devices are effective to reduce heat stress in confined livestock buildings for the climatic situation in Central Europe.

2. Materials and methods

2.1. *Meteorological data*

For the calculation of the inlet air temperature which is modified by air treatment devices, meteorological data are needed on an hourly basis (air temperature and relative humidity). The Austrian Meteorological Service ZAMG (Zentralanstalt für Meteorologie und Geodynamik) provided measurements for the weather station close to the city of Wels (48.16°N, 14.07°E) for the time period 1981 to 2010. Following the climate classification of Köppen and Geiger (c.f. Kottek, Grieser, Beck, Rudolf, & Rubel, 2006), the station is located within class Cfb (warm temperate, fully humid, warm summers) which is representative for large areas in Central Europe excluding the Alps. The annual mean temperature is 8.8 °C and the mean annual precipitation amount is 979 mm year^{-1}. For the whole area of Upper Austria in the future a

mean increase of temperature is expected with values of ~+1.4 °C (~± 0.5°) by the middle of the century. The number of hot days (daily maximum temperature ≥ 30 °C) is expected to increase in this region to between 4.7 and 5.0 days year^{-1} in the middle of the century compared to a mean value of 3.3 hot days year^{-1} for the reference period of 1971–2000 (Chimani et al., 2016).

2.2. Energy saving air treatment devices

The model calculations were performed for the 30-year time series of outside temperature and water vapour pressure on an hourly basis. The goal of the investigation was the analysis of the inlet air conditions in relation to the reference system. For each air treatment device, the inlet air temperature and water vapour pressure were calculated. The inlet air conditions are identical to the outside conditions for the reference system.

2.2.1. Earth-air heat exchanger EAHE

EAHEs utilise earth for heat storage. Outside air flows through tubes with a diameter D in the range between 0.1 and 1.0 m and a length L between 20 m and 200 m, buried at a depth z between 1 and 3 m. EAHEs are well-investigated and practically tested energy saving air treatment devices. The performance, i.e. air temperature and humidity at the end of the tubes, depends on the soil temperature T_s, the outside air temperature and humidity, the thermal features of the soil and the geometry of the tubes (Bisoniya et al., 2014; Ozgener, 2011; Tzaferis et al., 1992). All calculations are performed with the *number of transfer units* NTU method.

The soil temperature T_s as a function of depth z (m) and time t (DOY) is calculated by the well-known solution of the parabolic Fourier heat flow equation using a harmonic boundary condition with a period $\tau = 365$ d by

$$T_s(z,t) = T_m + A_0 \exp(-\gamma z)\sin\left(\frac{2\pi}{\tau}(t - t_o) - \gamma z - \frac{\pi}{2}\right)$$

The mean temperature T_m, the amplitude A_0 and the time lag t_0 are assessed by fitting the annual time course of the air temperature to the equation for the soil temperature above at the surface z = 0 m. The thermal property of the soil is expressed by the parameter $\gamma = \sqrt{\pi/\alpha_s\ \tau}$, which depends on the thermal diffusivity of the soil α_s and the period τ.

The geometry of the EAHE is characterised by the length of the tube L (m), the diameter of the pipes D (m), the mass flow rate m (kg s^{-1}), the thermal conductivity of the soil and the tubes λ_s and λ_t (W K^{-1} m^{-1}), respectively.

The temperature at the end of the tube T_{EAHE} is calculated by the outside air temperature $T_{a,out}$ and the soil temperature T_s by

$$T_{EAHE} = T_s + (T_{a,out} - T_s)\exp(-NTU)$$

with the model parameter $NTU = U\,A/c$ (–) using the overall heat transfer coefficient U (W m^{-2} K^{-1}), the surface area of the tubes $A = \pi DL$ (m^2) and the heat capacity rate $c = mc_a$ with the mass flow rate m (kg s^{-1}) and the specific heat of air $c_a = 1.006$ kJ kg^{-1} K^{-1}. U is calculated by the convective heat resistance R_c, the thermal resistance due to the conductive heat transfer through the wall of the tubes R_t with a wall thickness of r_2 and the thermal resistance due to the conductive heat transfer R_s between the tubes' outer surface and the undisturbed soil. The overall heat transfer resistance is calculated as the sum of all three resistances $R = R_c + R_s + R_t$. U is calculated by $U = 1/R$. R_c is assessed by $R_c = D/Nu\ \lambda_a$, with the Nusselt number Nu and the thermal conductivity of the air λ_a. For the calculation of Nu we use the equation by ASHRAE (2009b) $Nu = 0.023\ Re^{0.8}\ Pr^{0.4}$ with the Reynolds number $Re = v_t\ D/\nu_a$ and the Prandtl number $Pr = \nu_a/\alpha_a$, respectively, with the kinematic viscosity of air $\nu_a = 1.511\cdot10^{-5}$ m^2 s^{-1}, the thermal diffusivity of air $\alpha_a = 2.14\cdot10^{-5}$ m^2 s^{-1} and the air velocity inside the tube v_t (ms^{-1}). R_t is calculated by r_2, the inner radius r_1 and the thermal conductivity of the tube λ_t by $R_t = \ln[(r_1 + r_2)/r_1)]/2\pi\lambda_t$. R_s is calculated by $R_s = \ln[(r_1 + r_2 + r_3)/(r_1 + r_2)]/2\pi\lambda_s$ with the thermal conductivity of soil λ_s, the diameter of the soil annulus r_3 which is assumed equal to r_1.

Beside the sensible heat modification due to the EAHE also condensation can take place inside the tubes (Cucumo, Cucumo, Montoro, & Vulcano, 2008), if the outside humidity, described as mixing (humidity) ratio W_{out} (ASHRAE, 2009c), is higher than the mixing ratio in saturated conditions at the end of the tubes $W_{EAHE,S}$. The mixing ratio at the end of the tubes W_{EAHE} is calculated for $W_{out} > W_{EAHE,S}$ according to

$$W_{EAHE} = e^{-NTU}(W_{out} - W_{out,S}) + C_W(W_{out,S} - W_{EAHE,S}) + W_{out,S}$$

The constant C_W is a function of NTU calculated by

$$C_W = \frac{2(1 - e^{-NTU})}{NTU^2} - \frac{2\ e^{-NTU}}{NTU} - 1$$

which gives $C_W = -0.5424$.

For $W_{out} \le W_{EAHE,S}$ W_{EAHE} is the same as the mixing ratio of the outdoor air $W_{EAHE} = W_{out}$.

In Table 1 the selected parameters are summarised for a typical EAHE for livestock buildings. The resulting number of transfer units gives $NTU = 1.08$.

2.2.2. Direct evaporative cooling: cooling pads

In confined livestock buildings, direct evaporative cooling devices are in use to convert sensible heat (temperature) via evaporation of water into latent heat (humidity) with the major goal to reduce the inlet air temperature. We assume cellulose pads as a matrix to increase the wet surface. The efficacy of the cooling pads η_{CP}, also called wet bulb depression efficacy (ASHRAE, 2009a) is expressed by

$$\eta_{CP} = \frac{T_{out} - T_{CP}}{T_{out} - T_{out,WB}}100\%$$

with the outside air temperature (dry bulb) entering the cooling pads T_{out}, the air temperature leaving the cooling pads, entering the livestock building as inlet air T_{CP}, and the wet bulb temperature of the outside air $T_{out,WB}$. The efficacy η_{CP} is derived from empirical data of the producers. For the calculation we assumed $\eta_{CP} = 80\%$.

2.2.3. Indirect evaporative cooling: cooling pads combined with a regenerative heat exchanger

Indirect evaporative cooling systems result in a reduction of the inlet air temperature by evaporation without

Table 1 – Parameters for the earth-air heat exchanger EAHE, selected for the investigation. The parameters for air are selected and calculated for a temperature of 20 °C (Gröber, Erk, & Grigull, 1961). The thermal parameters for the soil are taken from Schauberger (1981).

Parameter	
Length of the tubes L	35 m
Diameter of the tubes $D = 2\ r_1$	0.2 m
Volume flow V	320 $m^3\ h^{-1}$
Kinematic viscosity of air ν	$1.511 \cdot 10^{-5}\ m^2\ s^{-1}$
Thermal conductivity of air λ_a	$2.57 \cdot 10^{-2}\ W\ m^{-1}\ K^{-1}$
Thermal diffusivity of air α_a	$2.14 \cdot 10^{-5}\ m^2\ s^{-1}$
Heat capacity of air c_a	1.006 $kJ\ kg^{-1}\ K^{-1}$
Wall thickness of the tubes r_2	5 mm
Thermal conductivity of the tubes (PVC) λ_t	0.19 $W\ m^{-1}\ K^{-1}$
Radius of the disturbed soil annulus r_3	0.1 m
Thermal conductivity of the soil λ_s	2.32 $W\ m^{-1}\ K^{-1}$
Thermal diffusivity of the soil α_s	$5.46 \cdot 10^{-7}\ m^2\ s^{-1}$

humidification. The outside air is cooled using direct evaporative cooling. Then this evaporatively cooled secondary air cools the outside air in a conventional air-to-air heat exchanger. We select a cooling pad CP and a downstream heat exchanger HE with a constant sensible efficacy of $\eta_{HE} = 65\%$ (ASHRAE, 2008). The temperature after the heat exchanger T_{CPHE} can be calculated by

$$\eta_{HE} = \frac{T_{out} - T_{CPHE}}{T_{out} - T_{CP}} 100\%$$

The overall efficacy of the combined system η_{CPHE} is given by

$$\eta_{CPHE} = \frac{T_{out} - T_{CPHE}}{T_{out} - T_{out,WB}} 100\%$$

2.3. *Parameters to quantify heat stress for farm animals*

For farm animals, heat stress can be quantified by the following parameters and related threshold values (1) air temperature (dry bulb) T, (2) temperature-humidity index THI, (3) specific enthalpy H, and (4) apparent equivalent temperature EAT (Table 2), which is equivalent to the specific enthalpy. For all these parameters, a related threshold value X has to be defined (see Table 2, values in bold). For a time series with the length t and n equidistant observations of a selected parameter x, the intervals τ can be defined by the exceedance of a certain threshold X. The first measure is the exceedance frequency $P_X = prob\{x|x > X\}$ (h a^{-1}). The second measure gives the absolute frequency N_X of consecutive time periods above a certain threshold X. The last one describes the exceedance area A_X calculated according to Thiers and Peuportier (2008) by

$$A_X = \sum_i \begin{vmatrix} x_i - X & \text{for} & x_i > X \\ 0 & \text{for} & x_i \le X \end{vmatrix}$$

The area above the threshold X is defined by analogy to degree-days (Gosling et al., 2013), except using the selected parameter x on an hourly basis instead of daily mean values. All parameters are calculated as yearly mean values over the 30 year period.

In Fig. 1, the selected heat stress parameters and the process temperatures for the direct and indirect evaporative cooling are summarised. For direct cooling by cooling pads, the change of temperature is an adiabatic process which means that the reduction of the air temperature runs parallel to the lines of a constant specific enthalpy. The increase of latent heat (humidity) is equivalent to the decrease of sensible heat. For the indirect cooling by the CPHE, the latent heat (humidity) does not change due to the cooling process. The inlet air humidity of the CPHE is identical to the outside air humidity.

3. Results

The hourly pairs of outside air temperature and water vapour pressure for Wels are shown in Fig. 2 for the entire period 1981 to 2010 to depict the climatic situation. The relative humidity is shown by the series of curves in 10% steps. For temperatures $T_{out} > 20$ °C, cases with saturation occur very infrequently (Fig. 3a). The exceedance of the threshold values X of the four heat stress parameters by air temperature and water vapour pressure is shown by the coloured lines.

For the successive figures we use the outside temperature of $T_{out} = 20$ °C as a cut off to show only those atmospheric conditions which can contribute to heat stress inside livestock buildings (summer dataset). The dataset with T_{out} >20 °C covers 11% of the 30-year time series.

The pairs of air temperature and water vapour pressure of the inlet air of the ventilation system are shown in Fig. 3. The exceedance of the four heat stress parameters is summarised in Table 3.

If the ventilation system is operated without air treatment (w/o AT), the inlet temperature and humidity are identical to the outside air. This is shown in Fig. 3a. The situation without air treatment is selected as the reference to describe the efficacy of the investigated energy saving air treatment systems. The frequency of exceedance of the threshold air temperature $X_T = 25$ °C and enthalpy $X_H = 55$ kJ kg^{-1} is approximately in the same range, $P_T = 275$ h a^{-1} or 3.1% and $P_H = 293$ h a^{-1} or 3.3%, respectively. The number of exceedance periods with a duration of $\tau = 12$ h is $N_T = 2$, and $N_H = 8$ times per year. This number describes the quantity of half-days above the threshold values.

The most effective air treatment device is the earth-air heat exchanger EAHE, shown in Fig. 3b. All states of the outside temperature above 20 °C are cooled below a temperature which would cause heat stress. Most of the inlet relative humidity values lie above 40%. This shift to higher relative humidity can be explained by the predominant sensible cooling. The mean temperature is reduced from 23.7 °C for the reference system without air treatment to 13.3 °C for the EAHE with a mean reduction of 10.4 K. Due to condensation inside the tubes, the vapour pressure is reduced from 16.0 to 14.2 hPa. The efficacy of the cooling power of the EAHE is also demonstrated by the fact that all exceedance parameters are zero.

The performance of the cooling pads CP as a direct adiabatic cooling system is shown in Fig. 3c. CP reduces the mean inlet air temperature by 5.2 K and increases water vapour pressure

Table 2 – Overview of heat stress parameters and the related threshold values X (upper limit for certain physiological reactions) for pigs and poultry used to evaluate the inlet air temperature T (°C), specific enthalpy H (kJ kg^{-1}), various temperature-humidity indices THI (–) and apparent equivalent temperature AET (°C). The selected parameters are highlighted in bold.

Heat stress parameter	Animal Species/Threshold X/Reference
Air temperature T (°C)	**X_T = 25°C optimum (pig and poultry)** (Liang et al., 2014; Ratschow & Schulte-Sutrum, 2008)
	X_T = 28°C recommended maximum (pig)
Specific Enthalpy H (kJ kg^{-1})	**X_H = 55 kJ kg^{-1} mild heat stress (pig)**
	X_H = 65 kJ kg^{-1} severe heat stress (pig) (Ratschow & Schulte-Sutrum, 2008; Rodrigues, da Silva, Vieira, & Nascimento, 2011)
	X_H = 67 kJ kg^{-1} (poultry)
Temperature Humidity Index	
THI_{Pig1} = 0.72 T_{DB} + 0.72 T_{WB} + 40.6	(Lucas, Randall, & Meneses, 2000; NWSCR, 1976)
	$X_{THI\ Pig1}$ = 75 alert situation (pig),
	$X_{THI\ Pig1}$ = 83 dangerous (pig)
THI_{Pig2} = 1.36 T_{DB} + 0.45 T_{WB} + 32	(Fehr, Priddy, McNeill, & Overhults, 1983)
	$X_{THI\ Pig2}$ = 85 mild heat stress (little effects) (pig)
THI_{Pig3} = T_{DB} + 0.36 T_{Dew} + 41.2	(Huhnke, McCowan, Meraz, Harp, & Payton, 2004; Roller & Goldman, 1969)
	$X_{THI\ Pig3}$ = 79 dangerous (pig)
	$X_{THI\ Pig3}$ = 84 emergency (pig)
THI_{Pig4} = 0.75 T_{DB} + 0.25 T_{WB}	3040 kg (pig) (Roller & Goldman, 1969) cited in Hahn, Gaughan, Mader, and Eigenberg (2009)
THI_{Pig5} = 0.65 T_{DB} + 0.35 T_{WB}	2030 kg (pig) (Ingram, 1965) cited in Hahn et al. (2009)
THI_{Pig6} = 1.17 T_{DB} + 0.63 T_{WB} + 32	(Tao & Xin, 2003b) (poultry)
THI_{Poul1} = 1.35 T_{DB} + 0.45 T_{WB} + 32	Egg production (J. Zulovich & DeShazer, 1990)
THI_{Poul2} = 0.85 T_{DB} + 0.15 T_{WB}	(Ravagnolo, Misztal, & Hoogenboom, 2000) (based on the heat index of the NOAA)
THI_{NOAA} = 0.81 T_{DB} + 46.3 +	Thresholds (St-Pierre et al., 2003)
R/100 (T_{DB} − 14.3)	$X_{THI\ NOAA}$ = 72 economic impact (fattening pig)
	$X_{THI\ NOAA}$ = 74 economic impact (sow)
	$X_{THI\ NOAA}$ = 78 economic impact (broiler)
	$X_{THI\ NOAA}$ = 70 economic impact (layer)
Apparent equivalent temperature AET	
$AET = T_{DB} + r_0\ W/c_A$	(Mitchell & Kettlewell, 1998)
	X_{AET} = 50°C mild heat stress (broiler) (~ H = 50 kJ kg^{-1})
	X_{AET} = 70°C severe heat stress (broiler) (~ H = 70 kJ kg^{-1})
	X_{AET} = 90°C extreme heat stress (broiler) (~ H = 90 kJ kg^{-1})

Dry bulb temperature T_{DB} (= air temperature T), wet bulb temperature T_{WB}, dew point temperature T_{Dew} relative humidity R (%), mixing ratio W, latent heat of evaporation r_0 = 2500 kJ kg^{-1}, heat capacity of dry air c_A = 1.006 kJ kg^{-1}.k^{-1}.

by 3.1 hPa, which corresponds to the psychometric constant. All parameters of the exceedance of the threshold of enthalpy X_H = 55 kJ kg^{-1} are equal to the reference system without air treatment. This is caused by adiabatic cooling, which means that the thermodynamic state of cooling is changed by the constant enthalpy. All values of the outside air move along the isopleths of the enthalpy. After cooling pad treatment, the inlet air shows a relative humidity between 70 and 100%. With respect to the threshold for air temperature X_T = 25 °C, the CP are very efficient with an exceedance of P_T = 3 h a^{-1}. A similar performance can be found for the THI for pigs THI_{Pig1} and poultry THI_{NOAA} with $P_{THI\ Pig1}$ = 4 h a^{-1} and $P_{THI\ NOAA}$ = 1 h a^{-1}.

The efficacy of the indirect adiabatic cooling system by CP and a heat exchanger is shown in Fig. 3d with a mean reduction of the inlet air temperature by 3.4 K. The vapour pressure is identical with the outside air. Only for an inlet air temperature of about 20 °C can the inlet humidity reach saturation. For higher temperatures as well as lower temperatures, the relative humidity is below the saturation of 100%. The frequencies of exceedance of the threshold for air temperature X_T = 25 °C and enthalpy X_H = 55 kJ kg^{-1} are reduced by 85% to P_T = 40 h a^{-1} and by 39% to P_H = 178 h a^{-1}, respectively, compared to the reference device. The number of periods with a duration of τ = 6 h is reduced by 88% to N_T = 3 times per year. Periods with a duration of τ = 12 h are totally eliminated.

The influence of the outside relative humidity on the cooling process of the CP and the CPHE are shown in Fig. 4. The grey lines, parallel to the 1:1 line, indicate the observed temperature depression. For the CP, the reduction of air temperature is distinctly higher, compared to the CPHE. If the two adiabatic systems are evaluated by the temperature-humidity indices (Table 3), the performance is nearly identical. The CP shows a better cooling performance with a lower inlet air temperature, whereas the reduction of the air temperature by the CPHE is lower, but the relative humidity does not increase so much. Due to the concept of the CPHE, the vapour pressure of the inlet air is the same for the outside air.

For CP, the frequency of exceedance of the threshold of the temperature humidity index $X_{THI\ Pig1}$ = 75 is reduced by 96% and 98% compared to the reference system from $P_{THI\ Pig1}$ = 89 h a^{-1} and $P_{THI\ NOAA}$ = 29 h a^{-1} to $P_{THI\ Pig1}$ = 4 h a^{-1} and $P_{THI\ NOAA}$ = 1 h a^{-1}, respectively. For CPHE, the reduction of the THI exceedance is 92% and 97%, respectively.

4. Discussion

The increase in the frequency of hot days and heatwaves in the last decades (Anders, Stagl, Auer, & Pavlik, 2014; Martin; Beniston et al., 2007; M.; Beniston, Stoffel, & Guillet, 2017;

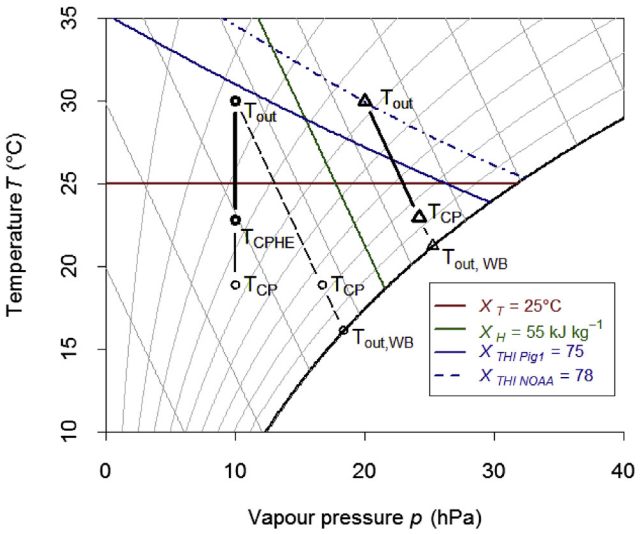

Fig. 1 – Mollier diagram with the process temperatures and vapour pressure for cooling pads CP as a direct evaporative cooling device (open triangles; with an outdoor temperature of $T_{out} = 30$ °C, vapour pressure $p_{out} = 20$ hPa and the inlet air temperature after the cooling pad T_{CP} for an efficacy $\eta_{CP} = 80\%$) and the combination of a cooling pad with a heat exchanger CPHE for an indirect evaporative cooling device (open circles with a dashed line for the cooling pads) with an outdoor temperature of $T_{out} = 30$ °C, vapour pressure $p_{out} = 10$ hPa and a temperature after the cooling pad T_{CP} with a efficacy $\eta_{CP} = 80\%$. The process temperatures and vapour pressure for the heat exchanger (solid line) with the air inlet temperature after the heat exchanger T_{CPHE} with the efficacy $\eta_{HE} = 65\%$. The courses of the four heat stress parameters (temperature T, specific enthalpy H and two temperature-humidity indices THI_{Pig1} and THI_{NOAA}) are given for the selected thresholds.

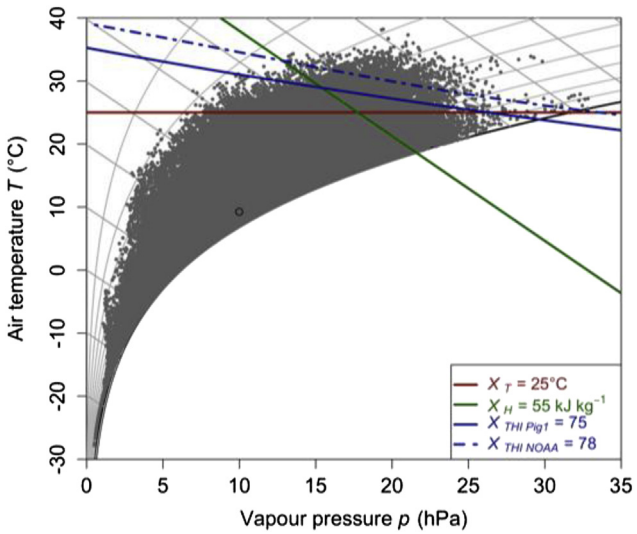

Fig. 2 – Air temperature T and vapour pressure p for Wels over the period 1981 to 2010 as hourly data, depicted in a Mollier diagram (a rotated psychrometric chart). The relative humidity is shown by the series of curves in 10% steps. The selected thresholds of the four heat stress parameters are depicted (temperature $X_T = 25$ °C, specific enthalpy $X_H = 55$ kJ kg^{-1}, temperature humidity index THI for pigs $X_{THI\ Pig1} = 75$, and temperature humidity index for poultry $X_{THI\ NOAA} = 78$). The mean value of air temperature T and vapour pressure p is tagged by an open circle.

Della-Marta & Beniston, 2008; Orth, Zscheischler, & Seneviratne, 2016) due to climate change results in a growing interest to avoid heat stress for farm animals. One of the measures to alleviate heat stress of farm animals in confined livestock buildings is the use of cooling systems to reduce the inlet air temperature. We analyse three air treatment devices that are compatible with current livestock housing systems. Despite the need for substantial capital investment – which is generally expected for confined livestock in the future (Olesen & Bindi, 2002) – they can be seen as incremental adaptation measures to avoid disruptions of the current livestock systems in the future (Kates, Travis, & Wilbanks, 2012).

The three investigated systems are evaluated by well-accepted heat stress parameters for pigs and poultry. A wide variety of heat stress parameters (Commission for Thermal Physiology of the International Union of Physiological Sciences, 2001; Gosling et al., 2013) are used to describe the influence of the thermal environment on humans (Epstein & Moran, 2006) and animals (Fournel, Rousseau, & Laberge, 2017; Hahn et al., 2009). The majority has been developed for cattle, which are kept predominantly on grassland. This means that these heat stress parameters are used to describe the conditions outside and not the indoor situation in confined buildings. Most of these parameters combine air temperature and humidity (e.g. wet bulb temperature, dew point temperature) by a linear model, called temperature humidity index THI. The coefficients for these models are derived by physiological reactions like core body temperature, skin temperature, respiratory rate, or the performance of farm animals (Roller & Goldman, 1969). Beside such combined parameters (temperature and humidity), some studies rely solely on air temperature (Liang et al., 2014) and thereby neglect the decisive role of humidity on livestock well-being. For the threshold air temperature of $X_T = 25$ °C, which is used to identify heat stress in this article, a thermal situation with a low humidity would be evaluated in the same way as a very humid situation (80–100%) and this is not reflecting empirical data. Within the thermoneutral zone humidity can be neglected, but it becomes relevant under heat stress conditions (Wegner, Lambertz, Das, Reiner, & Gauly, 2016.

Some of these heat stress parameters use thermodynamic variables like the specific enthalpy H (Moura, Nääs, Silva, Sevegnani, & Corria, 1997; Ratschow & Schulte-Sutrum, 2008, p. 346; Rodrigues et al., 2011) or the (apparent) equivalent temperature AET (Mitchell & Kettlewell, 1998). To include the convective heat release of broilers in a better way, the ambient air velocity is included to the THI and called temperature humidity velocity index THVI (Tao & Xin, 2003a). The THVI was not included in this investigation because we evaluated the inlet air conditions and not the indoor air. To evaluate heat stress of animals, threshold values for these indices have to be

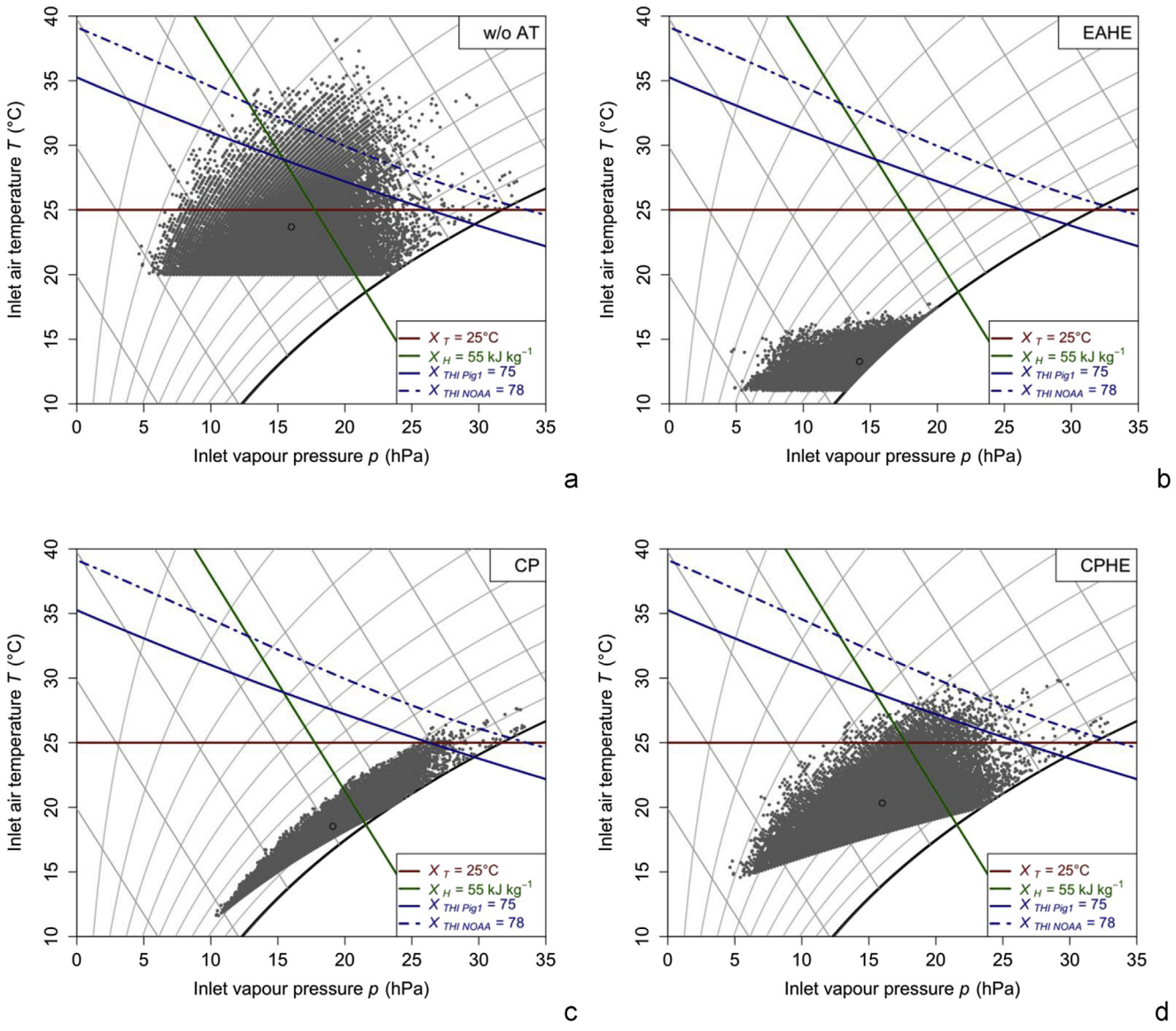

Fig. 3 – Air temperature and vapour pressure of the inlet air for summer conditions (T_{out} > 20 °C). (a) without air treatment w/o AT, (b) earth-air heat exchanger EAHE, (c) direct evaporative cooling by cooling pads CP, and (d) indirect evaporative cooling by cooling pads and a heat exchanger CPHE. For the evaluation of the inlet air, the four thresholds of the heat stress parameters are depicted (temperature $X_T = 25$ °C, specific enthalpy $X_H = 55$ kJ kg^{-1}, temperature humidity index THI for pigs $X_{THI\ Pig1} = 75$, and temperature humidity index for poultry $X_{THI\ NOAA} = 78$). The mean value of inlet air temperature T and inlet vapour pressure p is tagged by an open circle.

used. Table 2 reviews heat stress parameters from the scientific literature. We evaluated the hygro-thermal properties of the inlet air by a threshold for air temperature X_T, one for the specific enthalpy X_H, and thresholds for THI for pigs $X_{THI\ Pig1}$ and for broilers $X_{THI\ NOAA}$. This set of indicators covers the range of possible impacts of indoor air conditions on confined pig and poultry production and thereby balances the uncertainties of single indicators.

There is a strong influence of the selected heat stress parameter on the evaluation of the performance of an air treatment device. In Fig. 1, the slope of the selected heat stress parameters as well as the behaviour of the cooling pads CP and the CPHE are shown. For example, the effect of the adiabatic process of the cooling pads cannot be depicted by using the specific enthalpy X_H, because the slope of the cooling process and the slope of specific enthalpy are parallel. This means that the hygro-thermal situation of the animals is not improved by the cooling pads, using the enthalpy as an indicator. The most effective change of the hygro-thermal situation of the inlet air for livestock well-being is indicated if the air temperature is selected to quantify the performance because the slope of the air temperature (horizontal red line) and the adiabatic process shows the highest difference. This means that the selection of a certain heat stress parameter determines the calculated performance of the air preparation device.

Table 3 – The exceedances of threshold values X of the four heat stress parameters for the inlet air without air treatment, earth-air heat exchanger, direct evaporative cooling by cooling pads, and indirect evaporative cooling by cooling pads and a heat exchanger. The exceedance is described by the frequency P_x (h a^{-1}), the area above the threshold A_X and the number N_x of a certain duration of τ = 6/12/18 h of consecutive exceedances of the thresholds.

Air treatment devices	Heat stress parameters			
	Temperature T	Enthalpy H	THI_{Pig1}	THI_{NOAA}
Threshold X	25 °C	55 kJ kg^{-1}	75	78
Without Air treatment w/o AT				
Frequency P_X (h a^{-1})	275	293	89	29
Area A_X	710	1313	179	39
Duration N_X τ = 6/12/18 h	26/2/0	20/8/1	7/0/0	2/0/0
Earth-air heat exchanger EAHE				
Frequency P_X (h a^{-1})	0	0	0	0
Area A_X	0	0	0	0
Duration N_X τ = 6/12/18 h	0/0/0	0/0/0	0/0/0	0/0/0
Cooling pads CP				
Frequency P_X (h a^{-1})	3	293	4	1
Area A_X	7	1313	4	1
Duration N_X τ = 6/12/18 h	0/0/0	30/8/2	0/0/0	0/0/0
Cooling pads and heat exchanger CPHE				
Frequency P_X (h a^{-1})	40	178	7	1
Area A_X	44	645	8	1
Duration N_X τ = 6/12/18 h	3/0/0	10/4/1	1/0/0	0/0/0

The slopes of the THI (blue lines in Figs. 1 and 2, see also CIGR (1992)) are in between the two rather extreme indicators, i.e. specific enthalpy and air temperature. The slopes of both THI should reflect the capability of the animals to release sensible and latent heat as discussed by Beckett (1965) for pigs. Therefore it can be assumed that the slope of the temperature-humidity indices should be influenced by air temperature. For air temperatures close to body temperature, the slope of the THI curve should be steep, because evaporation dominates over sensible heat release. For lower air temperatures the sensible heat release becomes more important, which leads to a flatter slope.

With the four selected heat stress parameters, we describe the heat load for pigs and poultry by the exceedance of preselected thresholds. The exceedance is quantified by the frequency of the exceedance (in hours per year), the mean number of consecutive exceeding events per year with durations of 6–18 h and the area under the curve, i.e. the combined level and duration of exceedances for all four heat stress parameters. Hence, the latter takes into account not only the frequency but also the absolute value of the exceedance of the heat stress above the selected threshold. Our procedure of three exceedance estimates acknowledges empirical proofs that the combined duration and magnitude

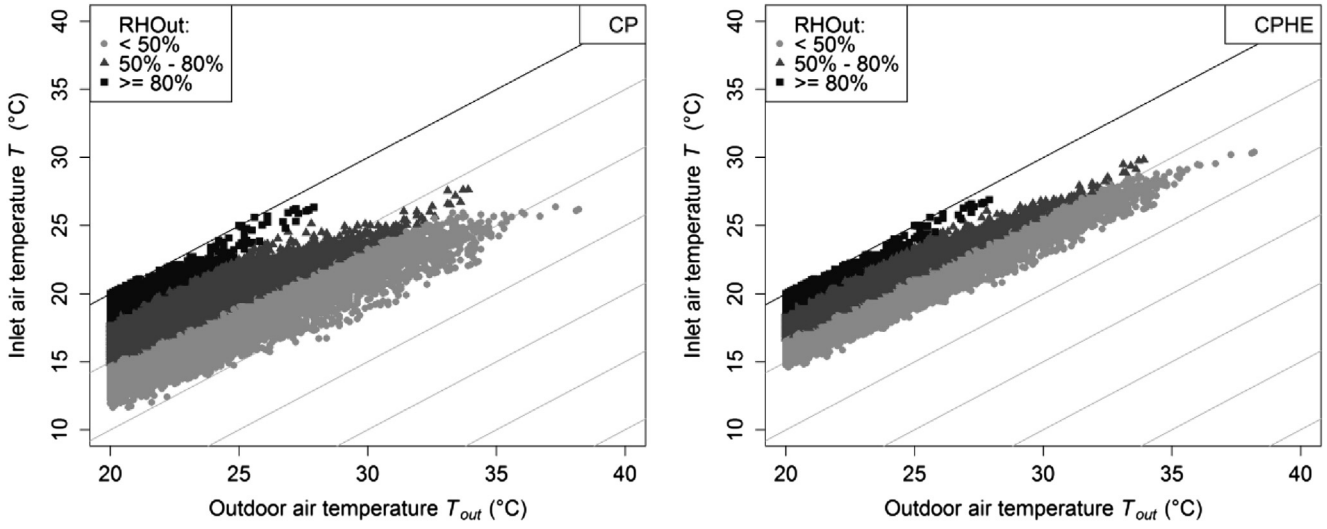

Fig. 4 – Inlet air temperature as a function of the outdoor air temperature for a low (rH_{out} < 50%), medium (50% ≤ rH_{out} < 80%) and high relative humidity rH_{out} (rH_{out} > 80%) for (left) direct evaporative cooling by cooling pads CP and (right) indirect evaporative cooling by cooling pads and a heat exchanger CPHE. The grey lines show the temperature depression of the inlet air by steps of 5 K for summer conditions (T_{out} >20 °C).

of exceedance events is more important than their mere frequency.

The calculations are performed with a 30-year data set of hourly values, which corresponds to the length of a climatic period and acknowledges inter- and intra-annual variability. Comparable studies were done for CP at 4 sites in Portugal by the use of hourly data for a two year period (Lucas et al., 2000) and for 17 sites in Oklahoma over 7 years (Huhnke et al., 2004). Valiño et al. (2010) calculated the efficacy of CP for livestock buildings for six European cities (Madrid, Athens, Milan; Stuttgart, London, and De Bilt) by the construction of average days for each month on the basis of climate change scenarios.

The additional energy needs as well as the heating and cooling performance of the investigated systems were not taken into account because this depends strongly on the ventilation system and its control unit and the livestock (Krommweh, Rösmann, & Büscher, 2014), e.g. a higher pressure drop of an air treatment device could be compensated by a lower ventilation flow rate due to a lower inlet air temperature during summertime to achieve the aspired indoor optimum temperature (Hessel, Zurhake, & Van den Weghe, 2011).

4.1. *Earth-air heat exchanger*

Earth-air heat exchangers EAHE have been used since the 1960s (Ozgener, 2011; Scott, 1965). Due to the energy crisis in the 1970s, several EAHE systems were installed for livestock buildings (Deglin, van Caenegem, & Dehon, 1999; Krommweh et al., 2014; Müller, Stollberg, & Venzlaff, 2005; MWPS-34, 1990; Schauberger, Axmann, & Keck, 1980; Schauberger & Keck, 1984). Several mathematical models (Bisoniya et al., 2014; Tzaferis et al., 1992) are in use to calculate the air temperature and humidity after EAHE treatment. We selected a model based on the number of the transfer units *NTU* (Mihalakakou, Santamouris, Asimakopoulos, & Tselepidaki, 1995; Tzaferis et al., 1992). This model describes the modification as an asymptotic approach of the air temperature to the undisturbed soil temperature in the depth of the earth tubes. Theoretically, the air temperature can only achieve the ground temperature with tubes of infinite length. Due to the limited length of EAHE on farms, this convergence is in reality interrupted. The assumed parameters of the EAHE in this study are typical for systems used for livestock buildings (Table 1). The value of $NTU = 1.08$ lies in between the range of EAHE systems investigated by Pfafferott (2003). A major advantage of the EAHE is its applicability over the entire year with the following features: (1) effective damping of short-term temperature fluctuations (Hollmuller, 2003), (2) heating of the inlet air temperature during wintertime which increases the ventilation flow rate and the related indoor air quality, and (3) cooling during summertime (Bisoniya, 2015; Hessel & Van den Weghe, 2011; Venzlaff & Müller, 2008; van Caenegem & Deglin, 1997).

Beside earth tubes, air-flowed pebble bed systems are used, which show a similar performance (Krommweh et al., 2014; Reichel, 2017). An alternative to the EAHE with a similar performance of air conditioning is a water–air heat exchanger and the use of groundwater with a water temperature that is close to the annual mean value of the air temperature (ter Beek, 2017).

Müller et al. (2005) estimated additional investment costs for EAHE per livestock housing unit for fattening pigs in the range of 50–100 € (corrected to 2017 real values), which is substantial considering total investment costs of about 400€ per unit according to KTBL (2017) for a farm size of 960 livestock housing units. Ascione, Bellia, and Minichiello (2011) estimated a payback period of 5–9 years. An Austrian livestock facility supplier (Stalltechnik Ing. Bräuer, Behamberg, Austria) states investment costs of 40–53 € per unit for a volume flow rate of 100 $m^3\ h^{-1}$, which equates to a fattening pig place. Using pebbles instead of pipes, the investment costs can be reduced to 16 € for a volume flow rate of 100 $m^3\ h^{-1}$ (corrected to 2017 real values). Consequently, costs of this type of air preparation are in the same range as the economic losses in animal performance (Müller et al., 2005; Venzlaff & Müller, 2008).

4.2. *Direct evaporative cooling: cooling pads*

Besides fogging inside the livestock building, cooling pads are one of the most widely applied direct evaporative cooling devices. The main advantages of this system compared to fogging are: (1) that the cooling pads affect only the condition of the inlet air and neither animals nor litter, and (2) the cooling pads clean the inlet air by retaining dust that is continuously removed by the spare water (Nääs, 2006).

Huhnke et al. (2004) and Fehr et al. (1983) used a constant efficacy of $\eta_{CP} = 70\%$ and $\eta_{CP} = 80\%$, respectively, to mimic the performance of cooling pads. In our model calculations, we derived the efficacy by the use of empirical data from CP producers which result in an efficacy of $\eta_{CP} = 70\%$. Nääs (2006) reported a range of η_{CP} between 52% and 90%. Koca, Hughes, and Christianson (1991) show that the efficacy is strongly influenced by the geometry of the airflow inside the pads with variations of about ±10%.

Lucas et al. (2000) estimated that, for the climate of Portugal, the use of cooling pads with an efficacy of $\eta_{CP} = 70\%$ can eliminate heat stress during most periods. For China, the applicability of cooling pads was evaluated along the different climatic zones by means of the wet bulb temperature (Xuan et al., 2012). Results show that the cooling pads can be used below a wet bulb temperature of 28 °C. Pads made from organic material showed the best performance with an efficacy of 95%.

With respect to service life and maintenance, the water quality (salinity and hardiness) can be decisive (Al-Helal, 2003; MWPS-34, 1990). Also clogging by algae is reported, which can be avoided by frequent drying of the pads (e.g. during nighttime) and by regular cleaning (MWPS-34, 1990). The advantages and disadvantages of evaporative cooling are summarised in DEFRA (2005).

With respect to investment costs, Watt (2012) estimated a payback period for several sites in the US between 2.0 for Denver and 4.9 years for Bridgeport depending on the climate conditions of the site.

4.3. *Indirect evaporative cooling: cooling pads in combination with a heat exchanger*

Indirect evaporative cooling is a combination of evaporative cooling followed by heat exchange (De Antonellis, Joppolo, Liberati, Milani, & Molinaroli, 2016; De Antonellis, Joppolo, Liberati, Milani, & Romano, 2017; Duan et al., 2012; Heidarinejad et al., 2009; Watt, 2012) which is also suggested for confined livestock buildings (van Caenegem et al., 2012). In

agricultural engineering, these systems are widely used to improve the storing conditions of fruits and vegetables (Lal Basediya, Samuel, & Beera, 2013), but no field reports are available for livestock buildings. Therefore, we included such a system in our performance test of air treatment devices in order to conduct a first assessment of this cooling system for livestock buildings. A detailed description of the modelling and measurements of indirect cooling systems is given by Boukhanouf, Alharbi, Ibrahim, Amer, and Worall (2017) and Hasan (2012). The term indirect adiabatic cooling is sometimes misleadingly used for farm animals meaning that direct cooling means inside the building by fogging and sprinkling systems (water contact of the animals) whereas indirect cooling is related to the inlet air (Hahn, 1981).

The device we tested for its cooling performance is a combination of a cooling pad and a fixed plate heat exchanger with an efficacy of $\eta_{HE} = 80\%$ (ASHRAE, 2008). Other heat exchanger systems, like rotary energy exchangers (heat wheel) or heat pipe exchangers, are described in detail by ASHRAE (2008). These systems are also recommended for livestock buildings (MWPS-34, 1990). Such heat exchanger devices cannot only be used during summer time to reduce heat stress (in combination with cooling pads) but also during wintertime to increase the inlet air temperature. For the latter, the ventilation rate can be increased (MWPS-34, 1990) which will substantially improve the inside air quality.

5. Conclusions

Heat stress for intensive pig and poultry production will increase in the future due to climate change. Even in temperate climates such as in Central Europe, this will cause economic losses for farmers. To reduce heat stress for farm animals, three energy saving air treatment devices were investigated. The evaluation of the performance was done by parameters which are used to describe heat stress for pig and poultry.

Earth-air heat exchangers show the best performance. By the use of this system, heat stress can be totally avoided. Beside the cooling effect, the following benefits can be expected: (1) effective damping of short-term temperature fluctuations, and (2) heating of the inlet air temperature during wintertime which increases the ventilation flow rate and the related indoor air quality.

Direct evaporative cooling by cooling pads shows a high potential for reducing temperature, depending on the relative humidity of the outdoor air. Due to the adiabatic cooling, the inlet air humidity ranges between 75% and 100%, which can cause problems inside the livestock buildings by moistening bedding materials.

Indirect evaporative cooling by cooling pads in combination with a subsequent heat exchanger results in a reduction of the inlet air temperature by evaporation without humidification. The additional feature is the possible use of the heat exchanger during wintertime, to reduce the sensible heat loss of the livestock building by heating the outside air by means of the outlet air of the livestock building.

The results show clearly that already existing cooling technologies can be applied in confined livestock buildings to successfully reduce heat stress for farm animals in Central Europe. Nevertheless, economic evaluations are necessary to prove whether the biometeorological effects of the suggested devices in limiting livestock production losses justify additional investment and maintenance costs. Such assessments would require the models in this study to be linked with climate change scenarios and economic farm models.

Acknowledgements

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