Optimal evaporative cooling system configuration for livestock building

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Abstract
The efficiency of ventilation and evaporative cooling systems usually depend on the system configuration, air distribution and cattle layout. From a thermo-fluid dynamics point of view, local recirculation zones could be present, reducing significantly the advantages of evaporative cooling. Furthermore, while for hot and dry climates such advantages are in general quite evident, in Italy the temperate and moderately humid summers make uncertain the cost-benefits ratio.
Previous results show that such systems are convenient even for temperate climates. In order to get an efficient implementation an optimal location of EC panels is required. From this point of view a double target is required: i) to reduce the indoor dry bulb temperature; ii) to keep low the increasing humidity due to the direct evaporative cooling effect.

Keywords*:* CFD, dairy cattle comfort, THI.

Introduction
The thermal exchange between cattle-breeding body surface and the surrounding environment influences growing, birth-rate and in general all the animal activity. As an example, in the case of dairy cattle the thermo hygrometric stress can cause a significant reduction of milk production (Armstrong,1994; Collier et al., 1982; Ravagnolo, 2000; Ray et al., 1992). In this direction, an important parameter to investigate in order to asses the discomfort is the Temperature Humidity Index (*THI*) (Thom, 1958; Buffington et al.,1981; Igono et al.1992, Bohmanova et al.,2007)

The bovine stress conditions can be reduced by using different methods: either increasing the animals thermal dissipation, and/or obtaining an improvement of the environmental conditions.

In case of dairy cattle the first experiences in the 1940s demonstrate that cooling obtained by direct water spraying on bovines increase milk production. Other studies show that wetting and shadowing cattle result in a significant reduction of respiration rate and body surface temperature (Seat & Miller, 1948), although the presence of the drawback of an increased percentage of possible cattle injuries due to the slippery floor.

The indoor environmental conditions can be improved by using traditional HVAC systems, with remarkable operating costs. In this framework, the use of Evaporative Cooling systems (EC) .In case of dairy cattle the first experiences in the 1940s demonstrate that cooling obtained by direct water spraying on bovines increase milk production. Other studies showed that wetting and shadowing cattle result in a significant reduction of respiration rate and body surface temperature (Seat & Miller, 1948), although the presence of the drawback of an increased percentage of possible cattle injuries due to the slippery floor.

The indoor environmental conditions can be improved by using traditional HVAC systems, with remarkable operating costs. In this framework, the use of Evaporative Cooling systems (EC) (Wang, 2001) could be more suitable.

In this paper the optimal layout of such panels is investigated by using a 3-D numerical simulations based on the CFD approach, considering a complete thermo-dynamical model including air buoyancy effects due to the natural convection, fan forced convection, sensible heat and humidity production due to the animals and the external solar radiation at the Sicilian latitude and in general at the temperate climate zones.

Materials and methods

The simulated domain has the following dimensions: 12,0 m (front width), 6,4 m (length of represented stable zone), 6,70 m (maximum height).

Perimetrical walls are made by 20 cm brickwork covered by 1,00 cm plaster. Four windows are present in the computational volume. Simple 3,00 mm glass is considered for each window, having a 2,25 m^2 surface area. The roof is composed by rain coated wood with air hollow space, insulation and plaster.

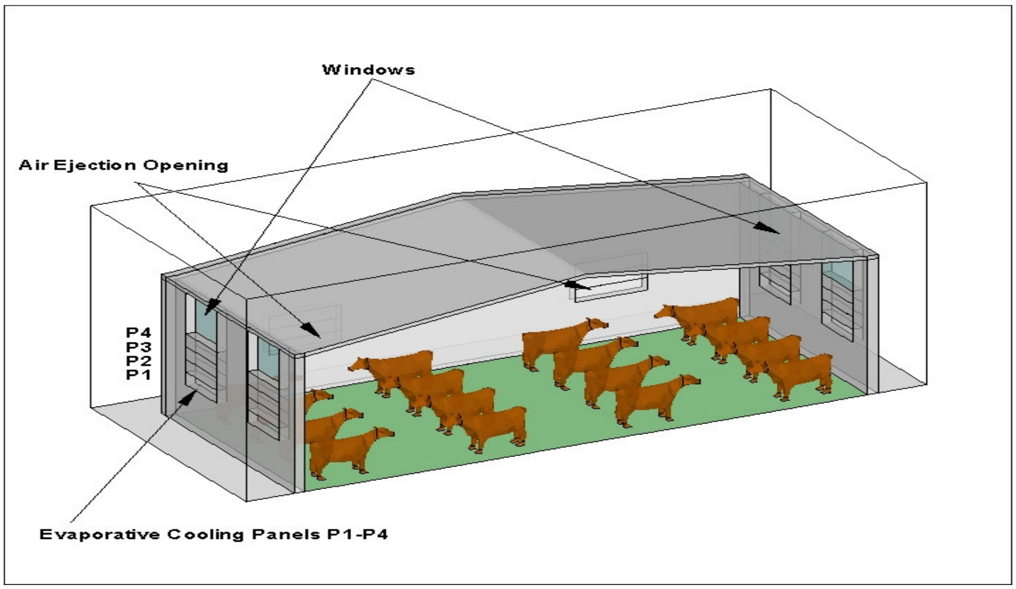

Figure 1. Simulation domain

<u>*Numerical methods*</u>

The Navier-Stokes equations for a compressible fluid were solved by computing the unsteady discrete solution of continuity, momentum and energy equations:

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \mathrm{v}) = 0 \tag{1}$$

$$\frac{\partial (\rho \mathrm{v})}{\partial t} + \nabla \cdot (\rho \mathrm{v} \times \mathrm{v}) = -\nabla p + \nabla \cdot (\overline{\overline{\tau}}) + \rho g \tag{2}$$

$$\frac{\partial (\rho E)}{\partial t} + \nabla \cdot (\mathrm{v}(\rho E + p)) = \nabla \cdot (k \nabla T + \overline{\overline{\tau}} \cdot \mathrm{v}) \tag{3}$$

where: E is the total energy ($E = e + v^2/2$), v is the air velocity, T is the temperature, $e = C_v\Delta T$ is the internal energy, p is the static pressure, $\bar{\bar{\tau}}$ is the strain tensor, ρg is the gravitational force and Cv the constant volume specific heat.

Air buoyancy effects due to the natural convection were modelled by the Boussinesq approximation , where in the momentum equation the density ρ is substituted by the term:

$$\rho(T) = \rho_0(1-\beta(T-T_0)) \tag{4}$$

being β the air thermal expansion coefficient, and ρ_0 the density at the reference temperature T_0 (288 K).

A CFD commercial code was used, based on control volumes approach for the balance equations in conservative form with space and time second order accuracy; the computational domain was suitably discretized in about 700˙000 control volumes.

Turbulence was modelled by using a realizable $k-\varepsilon$ model, using for the dissipation rate ε a transport equation derived from an exact equation for the transport of the mean-square vorticity.

A progressive variable time step in the range from 0,1 to 1 s was used for a global time simulation of 1 hour. In order to get shorter simulation times only a quarter of the whole livestock volume was modelled imposing a transversal symmetry condition. The reference stable was located at 37°N with a South exposed front .

The dynamic nature of the external cooling operating load was taken into account: following the hourly outdoor air dry bulb temperature of TRY (Test Reference Year, 1985); in particular, considering the Italian climatic area of the south Tyrrhenian Sea coast, the considered time interval is 12:00 ÷ 13:00 of the warmest summer day at this latitude (July 21th), and using a numerical time-varying solar ray tracing algorithm for the daily solar radiation (Fluent Inc. 2006).

The supply airflow rate $\dot{V}$ (19 m^3/s) of the evaporative cooling system was designed by using the following expression [20]:

$$\dot{V} = \frac{\dot{Q}_{sen} \cdot v_s}{c_p\left(t_{in} - t_s\right)} \tag{5}$$

where: $\dot{Q}_{sen}$ = 58,5 kW is the maximum seasonal sensible cooling load.

The outdoor rate was calculated according to the CLTD method (Cooling Load Temperature Difference, ASHRAE 2005), while the indoor one, due to the cattle presence, was computed by considering the metabolic load for each bovine equal to 914 W for 600 kg of body weight at 26°C (Yeck et al, 1959), , where v_s is the specific volume of the supply air, C_p is the constant pressure specific heat, t_{in} = 26°C is the design indoor air temperature and t_s = 23,5°C is the design supply air temperature.

The latter was obtained using a direct evaporative cooler starting from a design dry bulb temperature and relative humidity respectively of 32°C and 45%.

<u>*Thermal model*</u>

The thermal load for the cattle was computed using a formulation based on the model developed by McGovern and Bruce (2000).

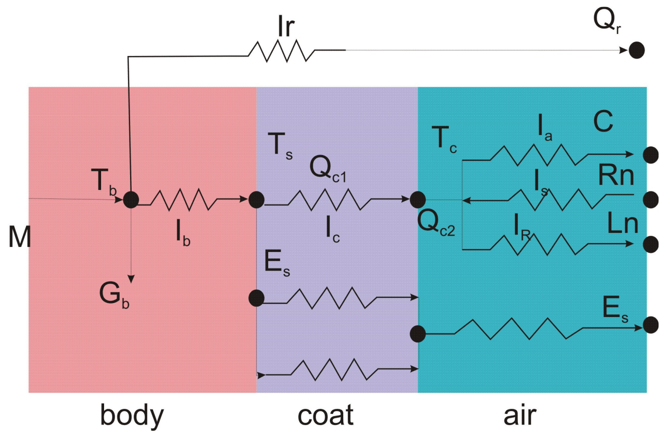

Figure 2. Schematic diagram of modelled heat flows following McGovern and Bruce (2000)

In this work the heat flows are modelled as shown in fig.1:

$$Q_b = M - Q_{r,b} - G_b \tag{6}$$

$$Q_{c,1} = Q_b - E_s \tag{7}$$

$$Q_{c,2} = C + L_n - R_n \tag{8}$$

where Q_b , $Q_{c,1}$, $Q_{c,2}$ are the heat fluxes from the body core to the skin, skin to the coat and coat to the air surrounding the simulated animal, respectively, $Q_{r,n}$ is the net heat flux from the respiration system, M is the metabolic heat production, G_b is the stored heat, E_s is the latent heat loss from the skin; C is the convective heat flux from the coat surface; L_n is the long-wave heat exchange with the surroundings and Rn is the short-wave radiation, with all variables expressed in W/m^2 of surface area of the simulated animal.

The heat flux from the body to the skin can be computed using the thermal resistance of the body tissue I_b as

$$Q_b = \frac{(T_b - T_s)}{I_b} \tag{9}$$

where Tb and Ts are the temperatures of the body and the skin of the simulated animal expressed in °C.

Combining Eqns (6) and (9) and solving for the skin temperature we obtain

$$T_s = T_b - I_b(M - Q_{r,b} - G_b) \tag{10}$$

where the normal body temperature of cattle is taken as 39 °C.

The combined effects of temperature and humidity are considered by using the following expression for the respiratory water loss ($g\ s^{-1}$), (Berman,2006):

$$Rwl = 0.41 - 0.02 * Ta + 0.0005 * Ta^2 - 0.004 * RH + 0.00004 * RH^2 \tag{11}$$

where Ta is the air temperature in °C and RH the relative humidity (%)

The THI index

The *THI* represent the stress cattle condition index in relation to the combined effect of air dry bulb temperature and humidity. For dairy cattle, the following relationship can be considered (Bohmanova et al.,2007):

$$THI = T_{db} + 0{,}36\ T_{dp} + 41{,}2 \tag{12}$$

where: T_{db} and T_{dp} [°C] are the dry bulb and the dew point temperatures of the local environment.

Results

Two different set of simulations were performed: the first one computing the THI index on a daily base, comparing results obtained with evaporative cooling panels in position P1, and evaporative cooling ceiling diffusers (not shown in Fig.1), and natural ventilation.

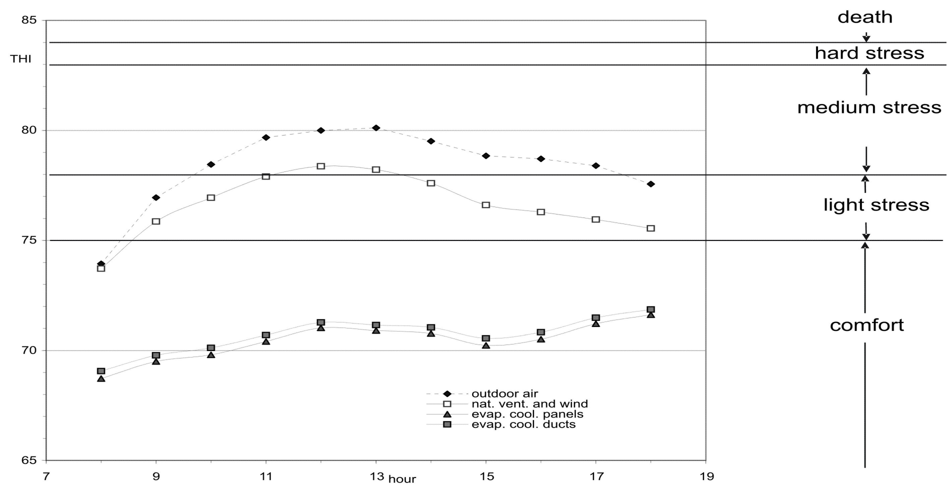

Figure 3. Simulated hourly THI

In Figure 3 the hourly average *THI* profiles, for the simulated cases, in the horizontal α-plane are reported. On the right side of the same figure the *THI* levels related to comfort conditions are shown too.

It is possible to observe that in both EC configurations the THI profiles are entirely in the comfort zone (THI=70 in the average). In the case of natural ventilation and wind a light stress is detected while for cattle without shadow the worst condition appears.

In Figure 4 the local THI index computed with EC panels in positions P1,P2,P3,P4 is shown after 10 m on a longitudinal plane positioned at 1.2 m from the pavement, highlighting the dominance of the thermal effects respect to those caused by the relative humidity, and suggesting that the best position for the EC panels is just over the bovine shoulders.

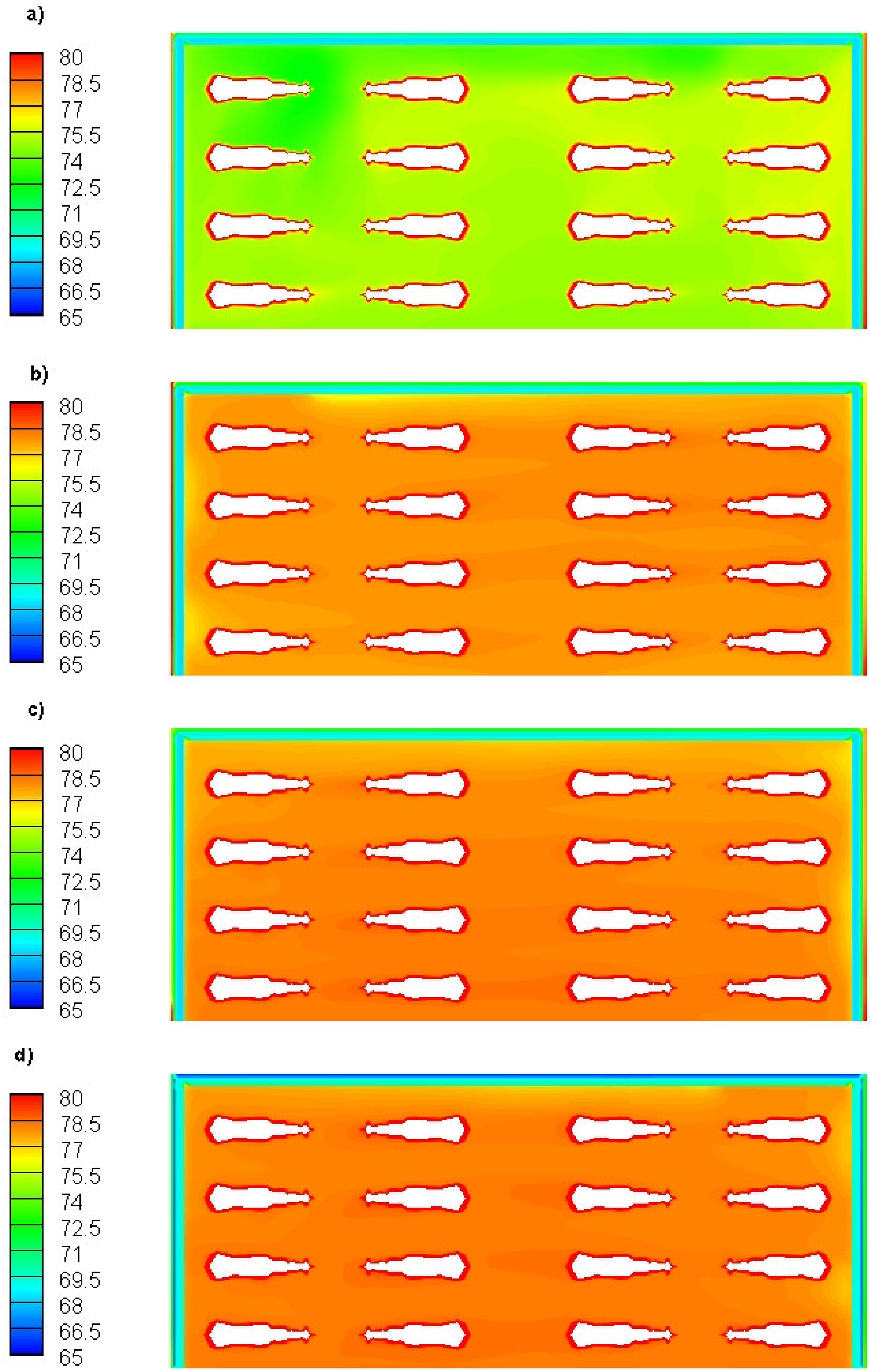

Figure 4. THI distribution for evaporative cooling panels positioned in: a) P1 b) P2 c) P3 d) P4

Conclusions

The Computational Fluid Dynamic approach provides a powerful tool to investigate local effects like temperature and relative humidity distributions, which constitutes a critical point in the use of evaporative cooling for heat stress relief for dairy cattle.

Particularly, the combined temperature and humidity, coupled with buoyancy effects seem to suggest that the best position for the EC panels seem to be just above the animal.

The stratification effects due to the metabolic heat production in the lower zone of the stable tends to contrast the cooler air descent in the P2:P4 positions, resulting in a higher THI index.

A same level collocation of EC panels should be avoided in order to reduce the air velocity disturbance on the animals, due to the presence of evaporative cooling fans.

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