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Enclosed Barn Conditioning System: A Comparison Between Traditional and Indirect Evaporative Cooling

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ABSTRACT

Enclosed barns can be an effective farming system, providing proper animal welfare conditions to the cattle, with also positive effect on dairy productivity. On the other hand, the air conditioning process can be extremely energy-intensive and expensive, especially during summer months. This is not only due to the high air temperature and the solar irradiance, but also because of the metabolic heat produced by the cattle. The climatic condition considered for this study is the temperate-no dry season-hot summer, typical of northern Italy, where more than 80% of Italian milk is produced. Two conditioning strategies have been evaluated: the traditional chiller with a reverse thermodynamic cycle and an indirect evaporative cooling system based on the Maisotsenko cycle. The two systems were analytically compared considering an enclosed barn with a population of 500 lactating cows, highlighting the pros and cons of each systems. Traditional reverse cycle is capable of consistently reaching the chosen thermo-hygrometric conditions, ensuring the highest dairy productivity. However, this comes at the expense of significant energy expenditure. On the other hand, indirect evaporative cooling is not always able to reach the planned conditions, but it guarantees a significantly lower energy expenditure.

1. INTRODUCTION

The dairy industry nowadays requires a full commitment to the cattle welfare in order to get a more comfortable environment. Improving animal wellness has various advantages, in fact it increases the production of milk and it reduces the cost of healthcare. In particular, heat stress results in fertility problems, and cardio-vascular illnesses and other severe issues with also the consequence of losses in milk production [1, 2]. One potential strategy to mitigate heat stress entails implementing barn conditioning measures to create a more favorable environment for the housed cows. In this work, a case study is presented in which 500 dairy cows were considered from the point of view of energy balance and the enclosed barn has been modelled to assess the required cooling power to neutralize. After the energy requirements were calculated, it was possible to develop a technical plan to sustain this cooling net power. Thermal load depends on the location of the facility, and therefore to conduct a meaningful analysis it is necessary to focus on a specific location. The location considered for this study is the Parmigiano-Reggiano production area on the southern bank of the Po River [3]. In this area, summer heat is quite intense and can be classified as Cfa (temperate, no dry season, hot summer) according to Köppen and Geiger system [4].

Two cooling strategies were evaluated outlining the advantages and drawbacks of each. The traditional vapor-compression cycle system and the indirect evaporative cooling system. The first one reliably achieves the desired thermo-hygrometric conditions, optimizing dairy productivity but requiring substantial energy consumption. Conversely, indirect evaporative cooling (based on the Maisotsenko cycle) may not consistently meet the target conditions but offers a considerable reduction in energy usage, as it does not require a compressor in the process [5]. The energy efficiency of this process makes it particularly appealing for making the dairy industry more sustainable, as CO_2 emission reduction is crucial for all sectors [6, 7].

The indirect evaporative cooling is mainly based on two parameters [8]:

-The relative humidity of the atmospheric air,

-The dry bulb temperature of the atmospheric air.

To exploit sensible heat of the air for water evaporation, higher atmospheric temperatures are advantageous. Optimal conditions entail elevated dry bulb air temperatures coupled with minimal relative humidity. As mentioned above, it is possible to state that the arid conditions are the perfect environment for this kind of system, with extremely high temperature and very low humidity. Modena has a medium-high dry bulb temperature and medium relative humidity during the summer and being classified as Cfa [4]. The indirect evaporative cooling can be a very effective way to cool down a stream of air using just water and a little amount of power to feed the main blowers [8, 9]. This ends with reducing the sensible heat of the main air flow without adding water directly to it, so eventually the final absolute moisture at the end of the process is actually the same, while the relative moisture ratio has changed accordingly with the temperature leap. This enables a very efficient system with high coefficient of performance (COP) (beyond 5-6) with a relatively low consumption of power (to sustain the blowers) and water (to feed the main evaporative mechanism) [10]. Despite these pros, the most relevant cons of this solution are that it is highly dependent on the external temperature and moisture ambient conditions.

2. MATERIALS AND METHODS

2.1 Barn heat load assessing

Firstly, the hypothetical heat load of a barn (Figure 1) sized to accommodate a population of 500 lactating cows was assessed.

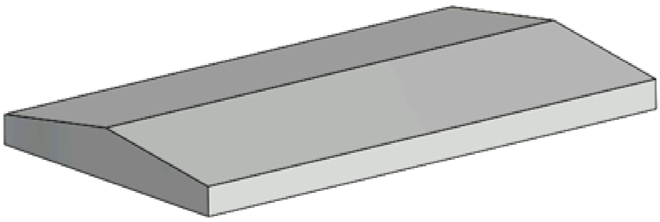

Figure 1. Simplified layout of the barn

It was assumed that the structure had an overall external surface area S_{barn} of 8,900 m^2 (including the roof and lateral walls) and a total volume V_{barn} of 41,700 m^3. Table 1 shows the building thermal properties considered for the calculation.

Table 1. Barn thermal properties

Property	Value
External surface emissivity ε	0.6
External surface absorptivity α	0.6
External surface reflectivity ρ	0.4
Wall thermal conductivity λ	0.1 W m^{-1} °C^{-1}
Wall thickness l	0.1 m
Internal convective heat transfer coefficient h_i	10 W m^{-2} °C^{-1}

The floor was assumed to be adiabatic. The hourly climatic conditions recorded by the weather stations near the city of Modena (one of the 5 province of Parmigiano Reggiano production) were downloaded from the Environmental Agency (ARPAE) of the Emilia-Romagna Region website (https://simc.arpae.it/dext3r). The data used refers to the period between 12:00 AM on August 15, 2018, and 12:00 AM on August 15, 2022, including air temperature T_{amb}, relative humidity φ_{amb} [%], wind velocity v, and total irradiation I. This data was used for the hourly heat loads computation for a time-span of four years, from August 15, 2018, to August 15, 2022.

2.1.1 Vapor-compression energy calculation

To calculate the electrical energy requirement of a vapor-compression chiller, it is necessary to establish the thermo-hygrometric conditions inside the barn. Specifically, a temperature of 25°C (T_{barn}) and a relative humidity of 50% (φ_{amb}) were chosen. The number of air changes per hour n_a was fixed at two. With this information, it is possible to calculate the heat load Q_{barn} that needs to be removed by the chiller to reach the desired conditions, which is the sum of: external heat load Q_{out} due to radiation and convection with external walls and roof, the sensible heat of the air introduced into barn Q_s, the condensation enthalpy of the water that needs to be condensed Q_w, and the metabolic heat produced by the cows Q_m as indicated in Eq. (1).

$$Q_{barn} = Q_{out} + Q_s + Q_m + Q_w \tag{1}$$

To calculate Q_{out}, the sol-air temperature [11] was taken into account with Eq. (2).

$$Q_{out} = S_{barn} \cdot U \cdot (T_{amb} + \Delta T_{sol} - 25°C) \cdot t \tag{2}$$

U is the overall heat transfer coefficient, t is the time step considered (1 hour), and ΔT_{sol} is the temperature increment due to solar irradiation. The overall heat transfer coefficient was calculated as:

$$U = \frac{1}{\frac{1}{h_{barn}} + \frac{l}{\lambda} + \frac{1}{h_{out}}} \tag{3}$$

h_{out} stands for the external convective heat transfer coefficient, and it was estimated as [12]:

$$h_{out} = 4 + 4v \tag{4}$$

ΔT_{sol} was calculated as:

$$\Delta T_{sol} = R_{out} \cdot F \cdot \alpha \cdot I \tag{5}$$

F is the view factor that was considered 1 for the whole building, R_{out} is the external resistance calculated as:

$$R_{out} = \frac{1}{h_{out} + h_r} \tag{6}$$

and h_r is the radiation heat transfer coefficient, calculated as:

$$h_r = \varepsilon \cdot 4 \cdot \sigma_0 \cdot T_{s-a}^3 \tag{7}$$

σ_0 is the Stefan-Boltzmann constant and T_{s-a} is the average temperature between the air temperature and the external surface temperature, assumed in this work 5°C higher than the air temperature. Q_s was calculated as:

$$Q_s = n_a \cdot V_{barn} \cdot \rho_{air} \cdot c_{p_{air}} \cdot (T_{out} - T_{barn}) \tag{8}$$

ρ_{air} is the air density and c_{pair} is the air specific heat at constant pressure. V_{barn} is the total volume of the barn. Q_w was calculated with Eq. (9).

$$Q_w = m_w \cdot 2501.3\ kJ/kg_{barn} \tag{9}$$

m_w is the mass of condensed water removed from the air entering the building, computed with Eq. (10).

$$m_w = (\omega_{out} - \omega_{barn}) \cdot n_a \cdot V_{barn} \cdot \rho_{air} \tag{10}$$

ω_{out} and ω_{barn} are the humidity ratios of the ambient and inside the barn (corresponding to 50% at 25°C). They were calculated as [13]:

$$\omega = 0.622 \cdot \frac{\varphi \cdot p_{sat}}{p_{amb} - \varphi \cdot p_{sat}} \tag{11}$$

p_{amb} is the ambient pressure (101,325 Pa), and p_{sat} is the saturation pressure at the specific temperature considered (25°C indoor, T_{amb} outdoor). When the m_w results negative for a specific condition, it was assumed zero and no water addition was considered. The hourly Q_m was assumed 0.9 kWh for each cow [1, 14, 15]. Q_{barn} was calculated for 8760 hours for each of the four years considered. Then, the electric consumption was calculated by dividing the total heat load by the energy efficiency ratio of the system, namely 2.5.

2.1.2 Indirect evaporative cooling energy consumption calculation

To evaluate the condition that can be reached inside the barn it was chosen an off-the-shelf evaporative cooling system, and the model considered was the Climate Wizard CW-80 Indirect Evaporative Cooler, able to process 20,880 m^3/h and with a power consumption of 10 kW per unit [16].

The evaluation of the CW-80 system's performance depends on a tool offered by Seeley International. This tool allows for determining the volume of air flow and its temperature upon exiting the dry channel. These determinations rely on the temperature and humidity conditions at the heat exchanger's entrance. With knowledge of the supplied temperature and volume of air to the environment needing cooling, subsequent calculations were performed to assess the cooling capacity of an M-cycle for cooling the barn.

Once the outlet temperature of the system T_{IEC_out} (corresponding to the inlet temperature in the barn) are known, it is possible to make a simplified black box calculation to calculate the final barn temperature T_f trough the following formula:

$$T_f = T_{IEC_{out}} + (Q_{out} - Q_m)/(n_a \cdot V_{barn} \cdot \rho_{air} \cdot c_{p_{air}}) \tag{12}$$

At this temperature it is also possible to calculate the final relative humidity knowing the absolute humidity at the outlet of the indirect evaporative cooling system that correspond to the ambient relative humidity. This because there is no water addition in the dry channel [17]. This calculation can be repeated at different air changes per hours, the higher the n_a, the lower the final temperature and the humidity. Knowing the nominal volume air flow of the system and its electric power consumption, it is possible to calculate the yearly electric energy consumption at different air changes per hour.

2.2 Temperature-humidity index (THI)

THI is a value that represents the combined effects of air temperature and humidity associated with the thermal stress level of the cows [18].

The formula used in this work for calculating the THI is the following [19]:

$$THI = [0.8 \cdot T_{amb}] + \left[\left(\frac{\varphi}{100}\right) \cdot (T_{amb} - 14.4)\right] + 46.4 \tag{13}$$

A THI ≤74 can be considered normal, a value between 74 < THI < 79 can be classified as alert condition, between 79 ≤ THI < 84 can be considered as a danger condition, and a THI ≥ 84 can be considered an emergency condition [20]. In this work, while the THI in case of compressed vapor chiller is technically imposed by the chosen temperature, the THI obtained with indirect cooling depends on the external conditions and on the number of air changes per hour.

3. RESULTS

3.1 Energy comparison

In Table 2, the energy consumption of various scenarios using a vapor compression chiller (C.V.) and an indirect evaporative cooling system (I.E.C.) is displayed. The comparison is made across multiple years of analysis and with an increasing number of air changes per hour for the evaporative cooling system. Along with the increase in the number of air changes per hour, the number of units (n_u) that need to be installed also varies.

Table 2. Cooling systems energy comparison

n_u	n_a	Year	1	2	3	4
1	2	C.V. Peak Power kW	576	611	524	535
		C.V. Yearly El. En. MWh	922	925	888	913
4	2	I.E.C. Peak Power kW	40	40	40	40
		I.E.C. Yearly El. En. MWh	299	306	301	298
10	2-5	I.E.C. Peak Power kW	100	100	100	100
		I.E.C. Yearly El. En. MWh	460	468	461	455
20	2-10	I.E.C. Peak Power kW	200	200	200	200
		I.E.C. Yearly El. En. MWh	537	534	534	520

As shown, the indirect evaporative cooling system consumes significantly less power than the vapor compression system at the same number of air changes per hour. The annual energy consumption of the indirect evaporative coolers is roughly half that of the vapor compression system if the number of installed indirect evaporative coolers allows for 10 air changes per hour during the most demanding part of the year.

3.2 THI comparison

In Table 3, THI (Temperature-Humidity Index) is considered. The number of hours in the year categorized as alert, dangerous, and emergency are depicted at different air changes per hour n_a.

Table 3. I.E.C. effect on THI at different n_a

n_u	n_a	Year	1	2	3	4
4	2	Alert [h]	1387	1398	1361	1273
		Danger [h]	1349	1453	1345	1321
		Emergency [h]	1267	1225	1166	1164
10	2-5	Alert [h]	1223	1088	1171	1146
		Danger [h]	293	287	224	154
		Emergency [h]	3	5	0	0
20	2-10	Alert [h]	463	467	348	289
		Danger [h]	19	13	3	1
		Emergency [h]	0	0	0	0

Among the scenarios presented, only the one with 20 machines installed and the capability to achieve 10 air changes per hour in onerous conditions has a small number of hours where the thermo-hygrometric conditions can be dangerous for the animals. However, in this configuration, several machines are not utilized for most of the year but only for a few hours, thus their economic value is not fully exploited.

Table 3 considers only the case of indirect evaporative cooling. This because, regarding the vapor compression cycle, a machine capable of maintaining a constant 25°C and 50% humidity was assumed, resulting in a constant THI of 71.7. Figures 2 and 3 show the temperature and humidity trends inside and outside the barn for configurations with 5 and 10 air changes per hour, using August 15, 2021, as a representative 24-hour period, which was extremely critical in terms of heat stress.

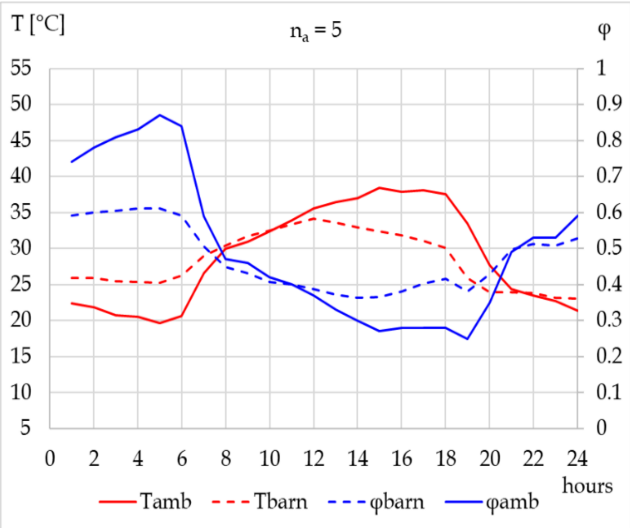

Figure 2. Temperature (T) and relative humidity (φ) inside and outside the barn with 5 air changes per hour

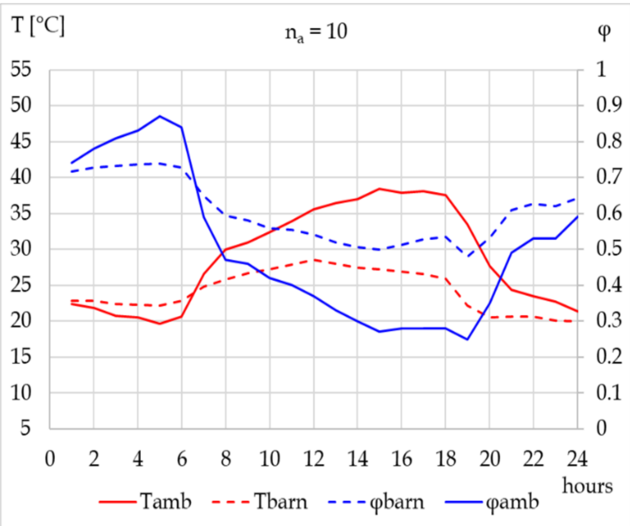

Figure 3. Temperature (T) and humidity (φ) inside and outside the barn with 10 air changes per hour

As seen in the previous figures, the temperature inside the barn can exceed the outside temperature under certain conditions, especially when there is a low number of air changes per hour. However, the situation improves drastically with 10 air changes per hour and 20 operating units. Although humidity is higher in this case, the THI is 76.3 compared to an outdoor value of 81.6. This is because temperature has a greater impact on heat stress.

4. CONCLUSIONS

The calculations presented above have delineated the pros and cons of both compressed vapor chillers and indirect evaporative cooling systems. The indirect evaporative cooling system offers advantages rooted in water evaporation and minimal energy consumption required to operate the main ventilators, resulting in remarkably low overall energy consumption. However, this system's performance is heavily reliant on external factors such as relative humidity and inlet air temperature, and it is not always able to maintain the target conditions unless an extremely high number of machines are installed, which remain unused for most of the year.

On the other hand, the traditional compression vapor chiller stands as a well-established and widely adopted solution capable of providing cooling power of up to MWs in standalone configurations. Its relatively lower dependence on external ambient conditions renders it suitable for various environments. Nonetheless, it consumes substantial energy, as its COP is not as high as that of the evaporative system.

A promising avenue for further research involves integrating and managing both systems throughout the day to minimize energy consumption. This combined approach could offer the flexibility of the traditional compression vapor cycle along with the enhanced efficiency of the indirect evaporative cooling system. Furthermore, the literature provides evidence of the potential benefits of using an indirect evaporative cooling system as a pre-cooler for a traditional chiller [20]. This strategy can potentially enhance the vapor compression cycle's performance by maintaining the condenser phase at a steady state, irrespective of external ambient conditions. Furthermore, a focus on evaluating these two systems from an economic point of view, both individually and in combination, would complete the feasibility analysis.

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