

Feasibility assessment of recovering waste heat from enclosed cattle barns for forage drying systems

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ABSTRACT

Enclosed barns, designed for improving animal welfare, are energy-intensive facilities. The demand for cooling capacity to maintain animal thermal comfort escalates during the summer, and year after year, global warming creates increasingly challenging conditions for temperature control. Drying forage constitutes a second significant energy-intensive process, requiring a large amount of thermal power. This work focuses on evaluating the economic and energetic benefits of integrating the forage drying process with a low-temperature waste heat recovery system in a dairy farm. In this context, the optimal solution lies in exploiting the waste heat generated by the enclosed barn chiller as an energy source to aid forage drying, thereby maximizing energy efficiency. The system outlined here uses a hydronic system to extract excess heat from the barn and convey it to the drying process, reducing overall expenses. The results of this comprehensive technical and economic feasibility study demonstrate the viability of implementing such a heat recovery system in the facility. Notably, recovering thermal power from the barn chiller allows for saving about 30 % of the fuel dedicated to the dryer burner (between 3.5 and 5.5 kWh per day per cow depending on the ambient conditions), guaranteeing a short payback period. Moreover, the carbon footprint of the farm can be drastically reduced, cutting more than 250 kg of CO₂-equivalent emissions per cow each year.

1. Introduction

Energy transition requires sectors responsible for a significant share of greenhouse gas emissions, such as breeding and dairy farms, to optimize their equipment and energy management practices in order to meet financial and renewable criteria [1–5]. Waste heat recovery is a powerful method to increase the efficiency of a process and save fuel, expenses and emissions [6–9]. Waste heat is the thermal energy produced during various industrial processes that is typically released into the atmosphere without being utilized [10]. Though the application of the recovery method differs for each case, the main principle entails using a specific vector to recover (at least partially) the excess heat and utilize it in a specific user (which can also be the main process) [11]. The potentialities of this field of knowledge are significant; in Europe alone, the overall technically-available waste heat is comparable to the industrial energy demand of Italy in 2021: about 300 TWh per year

[12,13]. This energy originates from various industries and, in particular, in sectors such as metal, food and chemicals. Papapetrou et al. [13] stated that most of the waste heat is produced at medium–low temperatures (up to 300 °C). The lower the temperature of the heat generated, the more challenging the recovery, due to thermodynamic factors. For this reason, this temperature range represents both the greatest opportunity for energy recovery and the most significant challenge from a technical point of view. Literature shows that certain fluids suit the purpose of recovering waste heat better than others, depending on the application [14]. One common application of waste heat is for drying processes. However, when the temperatures are suitable, another potential use is in organic Rankine cycles for electricity production [15–18]. Studying and modeling these kinds of applications is crucial for promoting a virtuous use of energy. This approach cannot only increase a company's current revenues but also reduce their overall emissions. In Italy, there are hundreds of companies that, due to obsolete equipment, unavoidably waste significant fractions of their primary energy [19],

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Nomenclature

<i>ACH</i>	Air change per hour. [h^{-1}]	<i>NTU</i>	Number of transfer units. [-]
<i>A</i>	Total surface of heat exchanger. [m^2]	<i>Nu</i>	Nusselt number. [-]
<i>A_{pipes}</i>	Total surface of pipings. [m^2]	<i>P_{condenser}</i>	Thermal power available to the condenser. [kW]
<i>C</i>	Heat capacity ratio. [-]	<i>P_{electric}</i>	Electric power of chiller compressor. [kW]
<i>C_{cold}</i>	Heat capacity of cold fluid. [W K^{-1}]	<i>P_{cooling}</i>	Power absorbed at the evaporator. [kW]
<i>C_{max}</i>	Maximum heat capacity. [W K^{-1}]	<i>P_{amb}</i>	Ambient pressure. [Pa]
<i>C_{min}</i>	Minimum heat capacity. [W K^{-1}]	<i>Pr</i>	Prandtl number. [-]
<i>C_{hot}</i>	Heat capacity of the hot fluid. [W K^{-1}]	<i>p_{sat}</i>	Saturation pressure. [Pa]
<i>CO₂</i>	Carbon Dioxide	<i>Q_{air}</i>	Heat load due to air entering the barn. [kWh]
<i>c_p</i>	Air specific heat at constant pressure. [$\text{kJ kg}^{-1}\text{K}^{-1}$]	<i>Q_{cond}</i>	Heat load due to humidity condensation. [kWh]
<i>c_{p,mix}</i>	Specific heat of the thermal fluid mixture. [$\text{kJ kg}^{-1}\text{K}^{-1}$]	<i>Q_d</i>	Daily heat request for alfalfa drying. [kWh]
<i>c_{s,a}</i>	Specific heat of alfalfa. [$\text{kJ kg}^{-1}\text{K}^{-1}$]	<i>Q_{d,m}</i>	Monthly heat request for alfalfa drying. [kWh]
<i>c_{s,H2O}</i>	Specific heat of liquid water. [$\text{kJ kg}^{-1}\text{K}^{-1}$]	<i>Q_{ext}</i>	Heat load due to external contributions. [kWh]
ΔT_a	Temperature increase of alfalfa. [K]	<i>Q_{loss}</i>	Piping heat losses. [W]
ΔT_{H2O}	Temperature increase of forage moisture. [K]	<i>Q_{met}</i>	Heat load due to cattle metabolic heat rate. [kWh]
ΔT_{mix}	Temperature drop of the mixture due to heat losses. [K]	<i>Q_{tot}</i>	Total heat load of the barn. [kWh]
$\Delta T_{mix-soil}$	Temperature difference between soil and thermal fluid. [K]	<i>R_{ext}</i>	Thermal external resistance. [K W^{-1}]
ΔT_{sol}	Fictitious increment of external temperature due to solar irradiation. [K]	<i>RH</i>	Relative humidity. [-]
<i>EER</i>	Energy efficiency ratio. [-]	<i>Re</i>	Reynolds number. [-]
ϵ_{HX}	Heat exchanger effectiveness. [-]	<i>R_{tot}</i>	Total Thermal external resistance of pipings. [K W^{-1}]
<i>F</i>	View factor. [-]	<i>SC</i>	Starting capital. [€]
<i>f</i>	Friction factor. [-]	<i>S_e</i>	Specific external surface area of the barn. [m^2]
<i>h_e</i>	External convective heat transfer coefficient. [$\text{W m}^{-2}\text{K}^{-1}$]	σ_0	Stefan-Boltzmann constant. [$\text{W m}^{-2}\text{K}^{-4}$]
<i>h_{ev}</i>	Latent heat of vaporization of water. [kJ kg^{-1}]	<i>SPB</i>	Simple payback period. [years]
<i>h_i</i>	Internal convective heat transfer coefficient of heat. [$\text{W m}^{-2}\text{K}^{-1}$]	<i>T_{av}</i>	Average temperature between the air temperature and the barn external surface temperature. [K]
<i>h_{rad}</i>	Radiation heat transfer coefficient. [$\text{W m}^{-2}\text{K}^{-1}$]	<i>T_{cold in}</i>	Inlet temperature of cold fluid in the heat exchanger. [K]
<i>HHV_{fuel}</i>	Higher heating value of fuel. [J kg^{-1}]	<i>T_{cold out}</i>	Outlet temperature of cold fluid in the heat exchanger. [K]
<i>I</i>	Total irradiation. [W m^{-2}]	<i>T_e</i>	Temperature outside the barn. [K]
<i>IRR</i>	Internal rate of return. [-]	<i>T_{hot in}</i>	Inlet temperature of hot fluid in the heat exchanger. [K]
<i>l</i>	Barn wall thickness. [m]	<i>T_{hot out}</i>	Outlet temperature of hot fluid in the heat exchanger. [K]
<i>m_{H2O}</i>	Mass of water that needs to be condensed. [kg]	<i>T_i</i>	Temperature inside the barn. [K]
<i>m_a</i>	Daily mass of alfalfa. [kg]	<i>t</i>	Time span considered. [s]
\dot{m}_{air}	Mass flow rate of air used to dry. [kg s^{-1}]	<i>U</i>	Global heat transfer coefficient of the walls. [$\text{W m}^{-2}\text{K}^{-1}$]
\dot{m}_{mix}	Mass flow rate of hydronic fluid mixture. [kg s^{-1}]	<i>U_{pipes}</i>	Global heat transfer coefficient of the piping. [$\text{W m}^{-2}\text{K}^{-1}$]
<i>m</i>	Hourly mass of fuel consumed in the standard drying process. [kg]	<i>v</i>	Wind velocity at 10 m from the ground. [m s^{-1}]
<i>m_{fuelsaved}</i>	Hourly mass of saved fuel [kg]	ω_e	Humidity ratios outside the barn. [$\text{kg}_{\text{H}_2\text{O}} \text{kg}_{\text{Dry air}}^{-1}$]
		ω_i	Humidity ratios inside the barn. [$\text{kg}_{\text{H}_2\text{O}} \text{kg}_{\text{Dry air}}^{-1}$]
		<i>YCF</i>	Yearly cash flow. [€ year ⁻¹]
		λ	Barn wall thermal conductivity. [$\text{W m}^{-1}\text{K}^{-1}$]
		ρ_{air}	Air density. [kg m^{-3}]

and hence, contribute to the overall carbon footprint of products and processes. With a basic retrofit campaign, it would be possible to recover a substantial portion of that energy and exploit it more efficiently. This work focuses on finding the best way to exploit waste heat in an energy-intensive dairy farm and then evaluating it through a techno-economic approach. The win-win solution proposed in this work aims to increase the productivity and profitability of a dairy farm while mitigating thermal stress issues in cattle.

The food sector has traditionally prioritized food quality and safety, but now it must also address sustainability and energy security challenges. It is crucial to focus on improving energy performance in the cattle sector as it stands out as one of the most energy-intensive industries, responsible for approximately 4.3 million tons of equivalent CO₂. The dairy industry currently contributes approximately 4.0 % to the total global anthropogenic greenhouse gas emissions and the demand for milk is expected to increase in the next years [20–23]. Dairy product manufacturers are constantly seeking sustainable processing methods to implement [24].

The present study has been applied to a specific location within one of the five Italian provinces on the south bank of Po' River, where Parmigiano-Reggiano cheese is produced [2]. This location is

strategically important for the dairy industry, yet it experiences significant summer heat. Modena, the central province of the selected area, is classified as Cfa (temperate, no dry season, hot summer) according to the Köppen and Geiger system [25]. It is well known that it is possible to increase the productivity of a dairy farm by decreasing the thermal stress on cattle [26–28]. Heat stress results in significant losses in milk production and leads to other severe issues such as fertility problems and cardio-vascular illnesses as described by [29–31]. In fact, hyperthermia could lead to cows' death if they fail to maintain thermoneutrality. This problem may occur especially in high-producing dairy cows, which may experience heat stress earlier than others [32]. The most widespread cooling systems in this field consist of fans with fogging systems, fans with sprinkler systems, and destratifiers. However, they are often insufficient to maintain proper animal welfare conditions during the hotter months [33]. Furthermore, the use of liquid water for heat abatement is not risk-free. It is necessary to avoid water stagnation that may cause slipping and hoof problems. Additionally, the water that does not vaporize becomes contaminated with manure and, therefore, needs to be properly managed [34]. Consequently, a possible solution to improve cattle thermal comfort is to establish an enclosed barn to control the thermo-hygrometric conditions of the indoor environment [35].

An air conditioning system can be used [36]. This eventually leads to the need for a water-cooled chiller composed of an inverse thermodynamic cycle machine with a compressor, an expansion valve, and two heat exchangers: the evaporator (to refrigerate the barn air) and the condenser (to dissipate the heat removed from the barn).

The air conditioning system is a costly option. Therefore, identifying a use for the waste heat available can be decisive in the application of such a housing strategy. The selected user to exploit the thermal power in this agro-industrial environment is a forage dryer used to reduce the moisture content of the forage coming from the crops that will compose the food rations for the dairy cattle.

Drying requires a significant amount of thermal energy to vaporize water, commonly generated by the combustion of fossil fuels [37,38], but it is an indispensable process in agriculture production. Reducing alfalfa (*Medicago sativa*) moisture content below 15 % facilitates long-term storage and allows the maintenance of a high-quality product, reducing carbohydrate and protein content loss [39–42]. On the other hand, both the enclosed barn with full air conditioning and the forage dryer are two extremely energy-intensive solutions. Therefore, finding an efficient way to use these two assets would improve energy usage in the farm and reduce overall expenses. To the best of the authors' knowledge, the possible utilization of the waste heat of an enclosed cattle barn has never been reported. The current work presents a model of an optimized hydronic system designed to recover the waste heat from the conditioning unit. The basic principle is to exploit the dissipated heat from the condenser of the chiller unit of the enclosed barn to save a considerable amount of fuel during the drying of the forage. This is implemented using a fluid to convey the accumulated heat, passing through the piping circuit to the final user.

2. Materials & methods

2.1. Heat load assessing

The first step was the evaluation of the heat request of the dryer and the thermal load that needed to be removed from the enclosed barn through the chiller, in order to assess if they were commensurable.

The following calculations were normalized with reference to a single lactating cow so that the results can be scaled for different farm sizes.

2.1.1. Dryer energy request

The considered hydronic system allows for the partial recovery of heat from the condenser of the chiller, which is used to preheat the air in the burner of a dryer. There are various types of dryers that can be classified based on their heat source and method of providing heat, such as heat pump drying systems, microwave drying, solar drying, dryers that use fuel combustion to heat the air, and a hybrid dryer when more than one heat source is employed for the scope (e.g. solar and liquid petroleum gas) [43–48]. The basic mechanism of the latter type of dryer is simple: a fan directs ambient air into the burner, where air participates in the combustion process and reaches the desired drying temperature for the product that is flowing on a conveyor belt [49,50]. Liquid petroleum gas (LPG), diesel and natural gas are currently the primary sources of energy for heated-air drying [51]. For the evaluated scenario, a dryer with a natural gas burner was considered due to its widespread use, especially in Italy, where the natural gas network is very extensive [52].

The thermodynamic model of the dryer was considered as a control volume with inputs (air, wet forage), power intake (amount of heat produced by the burner and recovered from the hydronic system) and output (air, dry forage) (Fig. 1).

By increasing the air temperature before entering the burner, it was possible to decrease the enthalpy leap required by the burner and, hence, reduce the amount of fuel necessary for combustion.

To calculate the energy demand of the dryer, the amount of forage

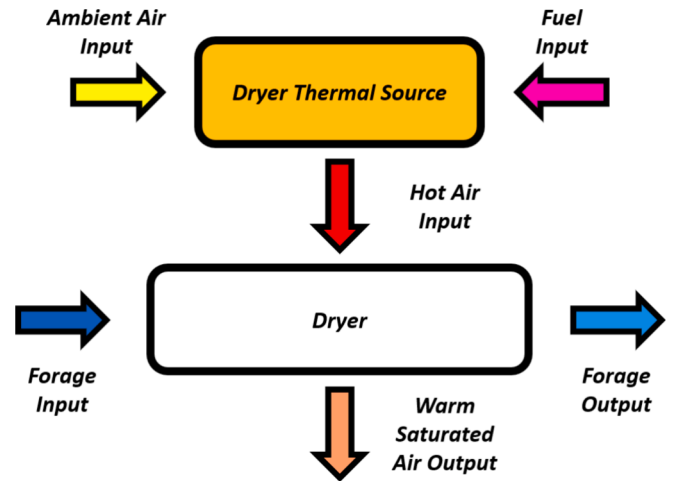


Fig. 1. Continuous drying process scheme.

used for feeding the cattle was considered. A total mixed rations (TMR) feeding technique is employed in Parmigiano-Reggiano cheese production farms. It consists primarily of alfalfa and grass [53]. The proper dehydration method for alfalfa involves the utilization of a horizontal dryer where the air is heated up to 130 °C and the highest temperature reached by alfalfa is 60 °C [40]. In this work, a daily dryer output of 12 kg cow⁻¹ (m_a) of dried alfalfa was considered. This value aligns with findings from a previous study [40] that providing 12 kg of alfalfa in the ration induces a positive effect on milk production. The specific heat considered for alfalfa $c_{s,a}$ at 0 % moisture content was 1000 J kg⁻¹ K⁻¹ [54]. Assuming an initial temperature of 15 °C for both water and alfalfa, and a final drying temperature of 60 °C, the temperature variation of both was set to 45 °C ($\Delta T_a = \Delta T_{H_2O} = 45$ °C). The initial moisture content of alfalfa was assumed to be 80 % on dry basis. Following these considerations, the daily heat request for drying the alfalfa Q_d was calculated with Eq. (1).

$$Q_d = m_a \cdot c_{s,a} \cdot \Delta T_a + 0.8 \cdot m_a \cdot c_{s,H_2O} \cdot \Delta T_{H_2O} + 0.8 \cdot m_a \cdot h_{ev} \quad (1)$$

Where h_{ev} is the water latent heat of vaporization. Assuming complete dehydration and considering only the specific heat of liquid water, rather than that of vapor (which has a lower value), leads to an overestimation of the energy requirement. However, since dryers do not operate at 100 % efficiency, this overestimation can be useful for practical purposes. The annual energy requirement of the dryer can be calculated by multiplying the daily requirement by the 365 days in the year. Considering that alfalfa cutting and harvesting take place between early May and late September [55], it is reasonable to assume that the utilization of the dryer is concentrated in these five months. Consequently, the monthly heat requirement for drying $Q_{d,m}$, is equal to:

$$Q_{d,m} = Q_d \cdot 365/5 \quad (2)$$

In the same way, it is possible to obtain the hourly heat requirement by dividing the annual heat demand by the number of hours the dryer is used in the five months considered. One possible assumption is to consider 12 h per day, which corresponds to a total of 1836 working hours per year. It is then clear that the higher the temperature of the inlet air, the greater the reduction in fuel consumption. Theoretically, if the inlet temperature approaches the drying temperature, the enthalpy leap required by the burner tends to zero. Therefore, it would be possible to turn off the burner and save 100 % of the fuel.

This model considers the energy balance between inputs and outputs and does not depend on the actual combustion kinetics.

2.1.2. Barn heat load

A barn with an external surface S_e of 18.4 m² cow⁻¹ was considered.

For instance, considering a cattle count of 400 heads, with an external surface of $18.4 \text{ m}^2 \text{ cow}^{-1}$, it was possible to design a barn with a floor area of $14.7 \text{ m}^2 \text{ cow}^{-1}$, and a height between 4 and 7.4 m. This value aligns with the prescription of $12 \text{ m}^2 \text{ cow}^{-1}$ from the Directorate-General for Agriculture of Emilia Romagna [56]. The corresponding total volume used for the calculation was $V = 83.6 \text{ m}^3 \text{ cow}^{-1}$.

In Table 1, the thermal properties and physical parameters of the barns are summarized.

To obtain a realistic estimation of the heat load of the barn, hourly climatic conditions recorded by the weather stations near Modena city were downloaded from a local environmental monitoring agency website (Arpae) [57]. Data collection spanned from 00:00 am on August 15, 2018, to 00:00 am on August 15, 2022, including air temperature at 2 m from the ground T_e , relative humidity at 2 m from the ground RH_e , wind velocity at 10 m from the ground v and total irradiation I . 8761 values for each of these parameters were used for the hourly heat loads calculation over the four years considered. In total, more than 140000 recorded data from the weather stations were used to obtain the most accurate and updated estimation possible.

The total hourly heat load Q_{tot} was calculated as the sum of: the external heat load Q_{ext} , the sensible heat Q_{air} of the air entering the barn, the condensation enthalpy Q_{cond} of the water that needs to be condensed to reach the selected thermo-hygrometric condition, and the metabolic heat production of the cows Q_{met} , as summarized in Eq. (3).

$$Q_{tot} = Q_{ext} + Q_{air} + Q_{met} + Q_{cond} \quad (3)$$

The quantification of Q_{ext} , Q_{air} , Q_{cond} , and Q_{met} is provided in Appendix A.

2.2. Heat recovery system

Regarding the chiller unit, it was possible to estimate the available power to the condenser ($P_{condenser}$) and evaluate the potential benefit of recovering this heat (Fig. 2).

$$P_{condenser} = P_{cooling} + P_{electric} \quad (4)$$

Where [58]:

$$P_{electric} = P_{cooling}/EER \quad (5)$$

Where EER is the energy efficiency ratio of the chiller. The data related to the chiller unit was derived from the previous heat load evaluation. The computation and modeling needed to use these values as a reference. The equipment to build the heat recovery system was chosen “off the shelf” to ensure the analysis was as realistic and accurate as possible. The selected chiller was the Carrier 30XWHP 1314B. Under the considered terms of use it is able to provide 1014 kW net cooling power with an EER of 2.46. Such a system can effectively cool an enclosed barn with approximately 400 lactating cows during the hottest months of the year. This means that the theoretically available thermal power at the condenser is approximately 1400 kW during peak load. From here, it was then necessary to assess the best way to exploit this significant amount of power to save fuel, money and eventually CO_2 emissions.

Table 1

Thermal properties and parameters of the barn.

Parameter	Value
External surface emissivity ϵ	0.6
External surface absorptivity α	0.6
External surface reflectivity ρ	0.4
Wall thermal conductivity λ	$0.1 \text{ W m}^{-1} \text{ }^\circ\text{C}^{-1}$
Wall thickness l	0.1 m
Internal convective heat transfer coefficient h_i	$10 \text{ W m}^{-2} \text{ }^\circ\text{C}^{-1}$
Barn internal temperature T_i	$24 \text{ }^\circ\text{C}$
Internal relative humidity RH_i	50 %
Air change per hour ACH	2 h^{-1}

For optimal chiller performance, the manufacturer specifies a specific temperature reduction for the chiller water as it passes through the condenser, as shown in Table 2. For this reason, it is necessary to place a second heat exchanger that heats the main heat transfer fluid to decouple the two circuits and guarantee reliable behavior of the one dedicated to the chiller.

Fig. 3 presents a potential layout of the part dedicated to heat recovery in the chiller condenser. By using a plate heat exchanger with double-sided liquid thermal exchange, it is possible to transfer a considerable portion of waste heat from the water chiller to the hydronic thermal fluid mixture. The hydronic thermal fluid considered most suitable for this operation is a 50 % by volume mixture of propylene glycol and water, referred to hereafter as the *glycol mix* [59]. The thermal properties and the choice of this fluid are described in Appendix B. Depending on the specific needs of the thermal user, it is possible to direct the required amount of chiller water into the heat exchanger using the flow selector 3-way valve. This enables the redirection of a specific amount of chiller water in the heat exchanger according to the instantaneous demand of energy. The final hydronic system will be composed as follows: a heat exchanger between chiller water and thermal fluid → piping → a heat exchanger between thermal fluid and inlet air in the dryer burner. To illustrate the actual layout of the system, a simple Piping and Instrumentation Diagram (P&ID) was developed. As shown in Fig. 3, the circuit connects the chiller condenser to the burner of the dryer.

In Fig. 4., a Sankey diagram represents the energy fluxes of the system, displaying the heat flux distribution throughout the entire system. The condenser heat is composed of the cooling requirements (which include cattle metabolic heat and barn external contributions) and the electric power requirement for the compressor. This thermal power is recovered by the hydronic circuit to be used in the dryer, in combination with the heat provided by the dryer’s natural gas burner. Finally, the overall heat losses of the system are depicted.

2.3. Technical model

2.3.1. Heat exchange computation

The effectiveness of all the heat exchangers was assessed through the application of the number of transfer units method. For this purpose, it was necessary to calculate Reynolds (Re), Prandtl (Pr) and Nusselt (Nu) numbers to evaluate the convective coefficient. In particular, the last number must be calculated using strict correlations intended for very specific considerations. Given this, the Gnielinski formula was used [60,61]:

$$Nu = \frac{f}{8} \cdot (Re - 1000) \cdot \frac{Pr}{1 + 12.7 \cdot \frac{f^{0.5}}{8} \cdot (Pr^{\frac{2}{3}} - 1)} \quad (6)$$

Where f is the friction factor, set to 0.05 [62].

From here, it is possible to compute the convective coefficient, which is necessary to calculate the overall heat transfer coefficient U . Assuming a steady-state of the system, and excluding the fouling (which in this case can be assumed negligible as all the fluids were very clean and the system was equipped with filters), an acceptable approximation of this coefficient is calculated with Eq. (A.1). The NTU method then employs the thermal capacities of both the fluids involved to assess the final temperatures of the hot and cold fluid (C_{min} and C_{max}). The fundamental equations that govern the heat exchange phase are as follows:

$$C = \frac{C_{min}}{C_{max}} \quad (7)$$

$$NTU = \frac{U \cdot A}{C_{min}} \quad (8)$$

The actual effectiveness of plate heat exchangers can be calculated with the following equations [60,62]:

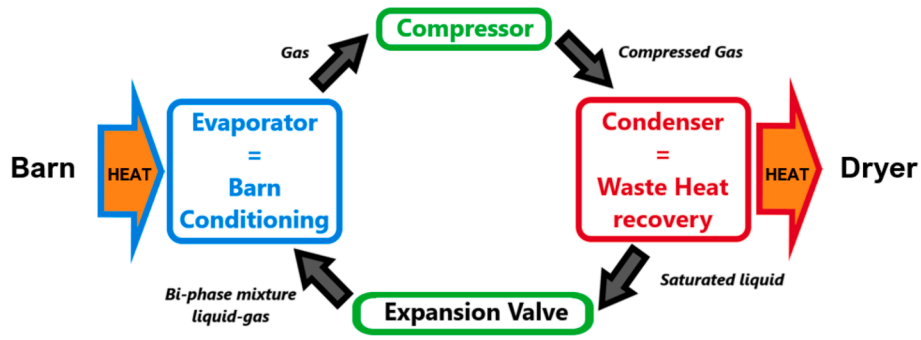


Fig. 2. Chiller configuration.

Table 2
Chiller water circuit characteristics.

Parameter	Value
Chiller water mass flow rate	67.5 kg s ⁻¹
Chiller water maximum inlet temperature	62 °C
Chiller water minimum outlet temperature	57 °C
Chiller water maximum temperature decrease	5 °C
Chiller water specific heat	4.1851 kJ kg ⁻¹ K ⁻¹
Chiller condenser waste heat	1412 kW

$$\epsilon_{HX} = 1 - \exp \left(\frac{\exp(-C \cdot NTU^{0.78}) - 1}{C \cdot NTU^{-0.22}} \right) \quad (9)$$

The computed effectiveness is then utilized to determine the outlet temperature for each thermal exchange.

$$\epsilon_{HX} = C_{hot} \cdot \frac{(T_{hot_{in}} - T_{hot_{out}})}{C_{min} \cdot (T_{hot_{in}} - T_{cold_{in}})} = C_{cold} \cdot \frac{(T_{cold_{out}} - T_{cold_{in}})}{C_{min} \cdot (T_{hot_{in}} - T_{cold_{in}})} \quad (10)$$

Changing the air temperature at the dryer inlet caused the glycol mix temperature to slightly vary in various locations of the hydronic circuit, namely the air-glycol mix heat exchanger and the glycol mix-chiller water heat exchanger. For this reason, an iteration process was necessary to

ensure a proper temperature trend along the circuit. Moreover, the temperature drop along the circuit was evaluated to take into account the thermal losses \dot{Q}_{loss} that were evaluated as:

$$\dot{Q}_{loss} = U_{pipes} \cdot A_{pipes} \cdot \Delta T_{mix-soil} \quad (11)$$

Where U_{pipes} is the overall heat transfer coefficient of the pipes, A_{pipes} is the piping area and $\Delta T_{mix-soil}$ is the temperature difference between the glycol mix and the soil. Soil temperature was set to 10 °C for the considered timespan [63]. U_{pipes} was calculated as:

$$U_{pipes} = \frac{1}{R_{pipes} \cdot A_{pipes}} \quad (12)$$

Next, the overall thermal resistance of the piping R_{pipes} was evaluated to consider the resistance due to convective in the glycol mix and the conduction through the first layer of the piping, insulation, and soil. Finally, it was possible to calculate the mixture temperature drop ΔT_{mix} knowing the thermal losses along circuit and the thermal capacity C_{mix} of the fluid:

$$\Delta T_{mix} = \frac{\dot{Q}_{loss}}{C_{mix}} = \frac{\dot{Q}_{loss}}{\dot{m}_{mix} \cdot c_{p-mix}(T)} \quad (13)$$

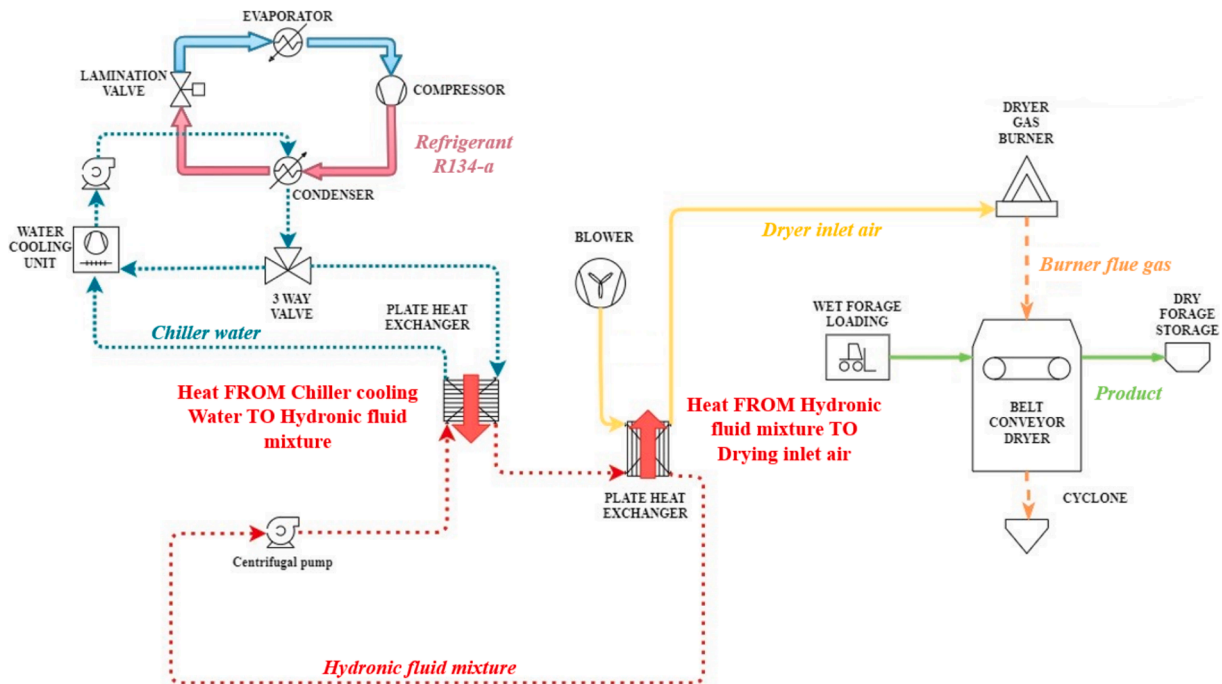


Fig. 3. P&ID of the hydronic system.

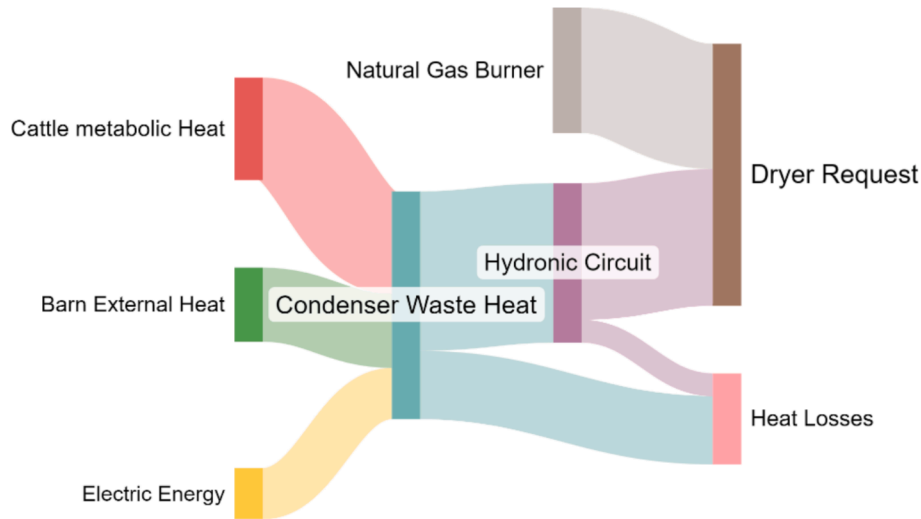


Fig. 4. Sankey diagram of heat recovery overview.

2.3.2. CO₂ emissions computation

To calculate the CO₂ emission, it was necessary to evaluate the fuel consumption of the system. The hourly mass consumption m_{fuel} was calculated as:

$$m_{fuel} = \frac{Q_d}{\eta_{comb} \cdot HHV_{fuel}} = \frac{\dot{m}_{air} \cdot (T_{drying} - T_{ambient}) \cdot c_{p-air} \cdot t}{HHV_{fuel}} \quad (14)$$

Where the air flow \dot{m}_{air} was set to 5 kg s⁻¹ to meet the requirements of a dryer capable of satisfying the thermal needs of 400 heads. This cattle number was also used in the economic assessment because it would be meaningless to estimate a budget for a single cow. HHV is the gross calorific value of natural gas (54 MJ kg⁻¹ [64]), η_{comb} is the combustion efficiency of the burner and t are the 3600 s of the considered hour. For each of the 1836 h within the considered period (from May 1st to September 30th), the average hourly temperature (air temperature at 2 m from the ground) was used as the input air temperature $T_{ambient}$ for the model. In each hour, the average temperature of the four years was employed. T_{drying} is 130 °C as specified in 2.1.1.

It was possible to assess the hourly amount of fuel saved $m_{fuelsaved}$ by calculating the heat recovered from the hydronic circuit:

$$m_{fuelsaved} = \frac{Q_{recovered}}{\eta_{comb} \cdot HHV_{fuel}} = \frac{\dot{m}_{air} \cdot (T_{pre-heated} - T_{ambient}) \cdot c_{p-air} \cdot t}{HHV_{fuel}} \quad (15)$$

Once the amount of fuel saved was known, according to the reduced enthalpy leap in the forage dryer, the actual avoided emissions were evaluated as:

$$CO_{2,avoided} = k \cdot m \quad (16)$$

Where k is the global warming potential measured in kg of equivalent CO₂ for each kg of used fuel (5.6 kg_{CO2} per kg of natural gas in this case) [65,66].

2.4. Economic assessments

In agro-industrial decision-making processes, it is necessary to assess feasibility not solely from a technical standpoint but also from an economic perspective. Hence, to complete the system assessment, an investment analysis was carried out based on the quotes provided by the main suppliers considering the thermal needs of a barn with 400 heads. The capital budgeting analysis methods used are as follows:

- Yearly Cash Flow (YCF): used to assess the overall fuel-saving cash flow related to the use of the aforementioned system

- Simple Payback period (SPB): used to assess the payback period
- Net Present Value (NPV): used to assess the investment value trend over the first 10 years of cash flow. It allows for understanding the influence of interest rates i and cash flow on the overall value of the investment.
- Internal Rate of Return (IRR): used to compute the discount rate that is deemed convenient for a 10-year investment, it represents the discount rate that makes the Net Present Value equal to zero in the selected timespan.

$$YCF = \dot{m}_{fuelsaved} \cdot time_{drying} \cdot Price_{fuel} - Power_{pumping} \cdot time_{drying} \cdot Price_{electricity} \quad (17)$$

$$SPB = \frac{SC}{YCF} \quad (18)$$

$$NPV = SC - \left[\frac{YCF_1}{1 + IRR} + \frac{YCF_2}{(1 + IRR)^2} + \dots + \frac{YCF_n}{(1 + IRR)^n} \right] \quad (19)$$

Where SC is the starting capital. A sensitivity analysis was conducted to assess the range of financial profitability based on realistic market cost fluctuations. The price of saved fuel was assumed to range between 0.0470 € kWh⁻¹ and 0.1339 € kWh⁻¹. These values correspond to the semi-annual natural gas prices for non-household consumers with an energy consumption between 1000 GJ and 9999 GJ between 2019 and 2023 in Italy [67]. The starting capital is considered as the sum of various components. Sensitivity analysis on these components and the annual electricity price was performed considering a possible ± 10 % variation in the costs summarized in Table 3 [68]:

Then, an IRR was evaluated to understand the actual profitability margin of the investment operation, calculated as follows.

The overall maximum timespan was set to 10 years, and the NPV was set to 0. This allowed for the assessment of the maximum interest rate that is feasible to adopt, ensuring that the actual years of return on investment are within the specified limit timespan.

$$SC = \left[\frac{YCF_1}{1 + IRR} + \frac{YCF_2}{(1 + IRR)^2} + \dots + \frac{YCF_n}{(1 + IRR)^n} \right] \quad (20)$$

Where SC is the starting capital. In an industrial environment, it can be said that investments of this magnitude are justified in having payback periods exceeding 5 years [69]. For this reason, a timespan of 10 years was used to comprehend the financial evolution of the investment throughout the selected amount of time. To take into account the 'time value of money' and compare the proposed solution to other investment

Table 3
Costs considered for the capital budgeting evaluation.

Item	Unit Cost	Quantity	Total Cost
Piping	10 € m ⁻¹	600 m	6,000 €
Piping insulation	20 € m ⁻¹	600 m	12,000 €
Pump (3 kW) and filters	3,500 €	1	3,500 €
Electric energy expenditure for pumping 2.9 kg s ⁻¹	0.28 € kWh ⁻¹	5400 kWh year ⁻¹	1,512 € year ⁻¹
1st liquid-liquid plate heat exchanger	20,000 €	1	13,000 €
2nd liquid-gas plate heat exchanger	20,000 €	1	20,000 €
Installation and service	18,000 €	1	25,000 €
Glycol mix	5 € kg ⁻¹	2500 kg	12,500 €
Total waste heat recovery system 1° year			93,512 €

The overall cost of the waste heat recovery system is approximately 100,000 €. All price data were acquired through commercial research from the main manufacturer of the various components.

opportunities, three discount rates were considered: 0 %, 5 % and 10 % [70].

3. Results

3.1. Heat load Assessing

The heat load analysis showed that the yearly dryer request per cow is about 2672 kWh. The heat load of the barn amounts to (5130 ± 110) kWh per cow, and the waste heat available at the condenser is given by the sum of the heat load of the barn and the electrical power of the chiller, resulting in (7170 ± 160) kWh per cow. The uncertainty represents the standard deviation over the four years analyzed. However, as discussed in the Material and Methods section, only the heat available in the months when the dryer is in operation can be exploited. In Fig. 5, it is possible to see the comparison between the heat available at the condenser of the conditioning system and the monthly dryer requests (based on the actual food needs of the cattle). The error bars represent the standard deviation for each month considered.

As seen in Fig. 5., the availability of waste heat from the condenser is considerably higher than the actual demand for the dryer. Therefore, recovering this thermal energy is effective and can significantly reduce

the fuel consumption for the drying process. Moreover, the waste heat is, to a certain extent, directly proportional to the actual wellness of the cattle. By lowering the internal temperature of the barn, the thermal comfort increases. Although this leads to an increase in the actual energy consumption of the chiller, the availability of waste heat from the condenser also rises, making the recovery even more convenient. For a more cautionary analysis, the dryer heat requirement was also compared to the heat available at the condenser during the 12 central hours of the day, in particular, from 7 a.m. to 7p.m. This is because it is more likely that this kind of equipment is not operated continuously during the day but only during the working hours of manpower.

Furthermore, when solar irradiance and air temperature are lower, cattle can graze outside the barn without the risk of heat stress due to ambient conditions, thereby also reducing the energy costs associated with conditioning the barn.

By excluding the nighttime hours and considering the temperature range during the period 7 a.m. to 7p.m., the waste heat available at the condenser per cow amounts to (4960 ± 160) kWh every year, which is about 70 % of the heat available when considering the entire 24 h of a day. Fig. 6. depicts the heat available at the condenser of the conditioning system and the dryer request divided by month, considering only the central 12 h of the day.

3.2. Simulation results

In Table 4 the main simulation parameters are summarized.

The chiller condenser heat exchanger exhibits a low thermal capacity ratio. This is attributed to the high mass flow of cooling water and the relatively high power availability exploitable by the glycol mix. In addition, in the second heat exchanger (located at the inlet of the dryer), the thermal capacity ratio is significantly higher than in the previous one. Consequently, in alignment with the NTU theory, the final effectiveness is not as high. However, to significantly increase this parameter, a much larger heat exchanger would be necessary, and such an investment was not considered profitable given that the effectiveness parameter is not related to the dissipated energy.

The temperature path, along with thermal losses in the circuit for the two extreme cases regarding minimum (T1), maximum (T2) and mean (T3) ambient air temperature, is presented in Fig. 7.

The recovered power depends on the condenser usage conditions,

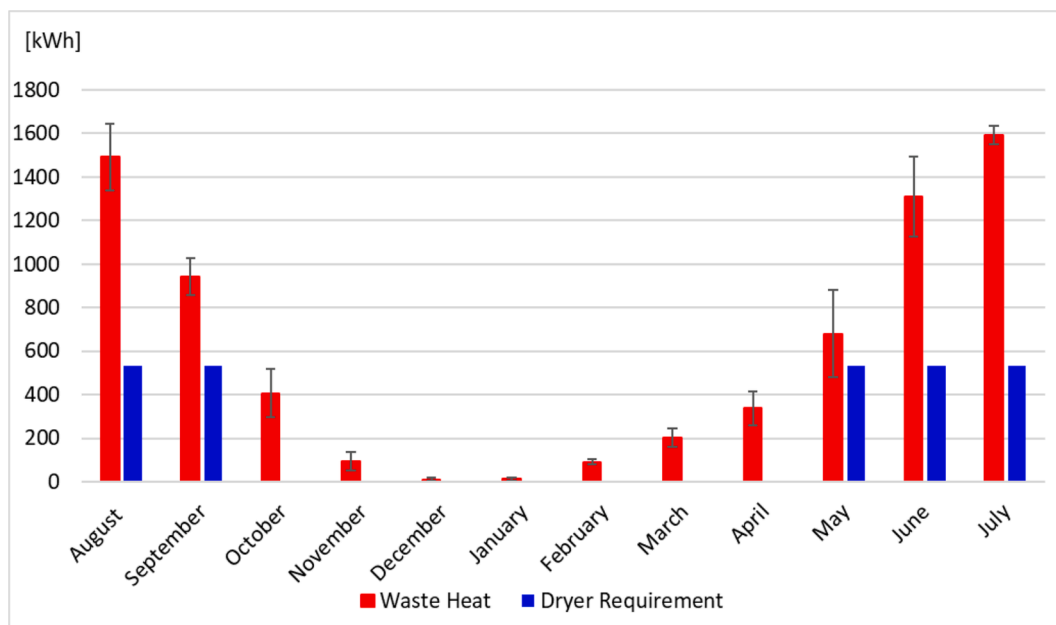


Fig. 5. Waste heat vs dryer requirement.

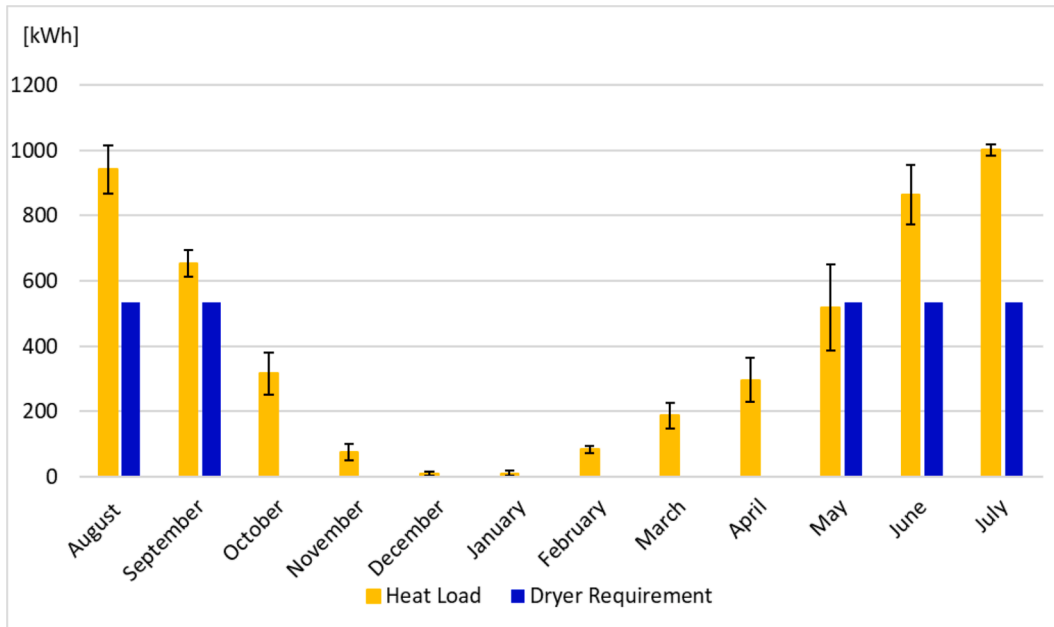


Fig. 6. Waste heat from 7 a.m. to 7p.m. vs dryer requirement.

Table 4
Simulation Outputs.

Parameter	Value
Glycol mix / Cooling water thermal capacity ratio	0.04
Glycol mix thermal capacity	10.6 kW K ⁻¹
Cooling water / Glycol mix heat exchanger effectiveness	0.9
Glycol mix / Drying air NTU	3.96
Drying air thermal capacity	5 kW K ⁻¹
Drying air / Glycol mix thermal capacity ratio	0.47
Glycol mix / Drying air heat exchanger effectiveness	0.76

which are a function of the ambient temperatures and solar radiation. In particular, it is higher when the air temperature is warmer and solar radiation is more intense. The calculations show that for the average year considered, the proposed hydronic system allows for energy

recovery ranging from 3.5 to 5.5 kWh per day per cow, with an average of 4.3 kWh, depending on climate conditions. The median value of energy recovered is 4.2 kWh, while the corresponding median temperatures in the circuits are very similar to those reported for the average. The glycol mix flow was chosen to maximize its temperature. A higher flow would decrease the maximum glycol mix temperature, with the effect of reducing also the temperature of the air upstream of the burner. However, the availability of power was limited by the lower temperature of the chiller water compared to the dryer operating temperature, which was one of the constraints of the current system. This limit can be overcome by recovering additional energy for other potential users, depending on the facility's layout (e.g. the presence of dairy boilers for pasteurization, other types of burners for cheese production, etc.). Another possible solution to increase the share of energy provided by the proposed hydronic system, compared to natural gas, is to lower the drying air temperature. In particular cases, such as Parmigiano-

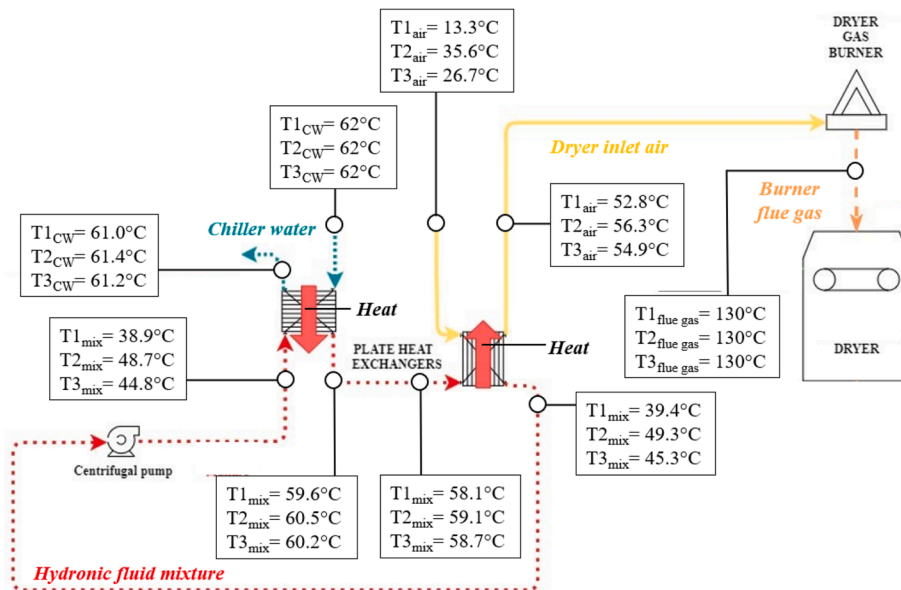


Fig. 7. System temperatures under minimum, maximum and average ambient conditions.

Reggiano cheese production, the maximum air temperature is 100 °C. Furthermore, only 2 kg per cow per day of forage dried at a temperature higher than 100 °C can be used, as allowed by the cheese production standards [71]. In this work, an air temperature of 130 °C was considered to provide a conservative analysis that can be applied to a wide range of dairy applications.

3.3. Economic results

The annual savings amounted to 27 % of the fuel (natural gas in this case) used to dry the forage. All the computations were made to establish an even drying time schedule, but it is clear that it would be possible to use the pre-heated air alone for drying and completely bypass the burner. This approach can significantly increase fuel savings (since the burner is not used, resulting in 0 kg s⁻¹ of fuel consumption) but the processing time rises drastically as well, creating issues in the modified supply chain. Furthermore, the harvesting strategy can be subjected to weather conditions, so it is not always possible to schedule it according to the drying strategy.

In this work, the specific fuel savings per cow were evaluated considering a drying temperature of 130 °C, with the drying taking place from May to September, 12 h day⁻¹.

It is relevant to note how the savings are lower in hotter months (Fig. 8.). This is because the enthalpy leap to reach the target temperature inside the dryer would be reduced, so the advantage provided by the waste heat recovery system would decrease.

The actual fuel savings range from 12,920 € to 36,810 € every year, depending on the actual price of natural gas. This results in a total savings of between 11,260 € and 35,500 €, considering the electric energy consumed to run the system. The SPB at a 0 % discount rate ranges between 2.4 and 9.1 years (Fig. 9), depending on the three different fuel prices and component cost fluctuations. Case 2 refers to a 10 % decrease in component and electric energy costs, while case 3 refers to a 10 % increase. The investment analysis clearly shows how the system can be feasible not only from the technical but also from the economic point of view. The internal rate of return is also shown in Fig. 9.

The actual savings can be different depending on:

- The actual usage of the chiller and, consequently, the effective availability of heat at the condenser.
- The actual schedule of the dryer, which can be adjusted according to the company's needs.
- The maximum temperature reachable by the glycol mix, which can be higher when exploiting other waste heat sources available on-site.

For a further capital budgeting analysis, the Net Present Value graphs were used to evaluate the financial convenience of the investment for different discount rates in Figs. 10, 11 and 12. Three specific natural gas costs were considered, corresponding to the minimum (47 € MWh⁻¹), time-weighted average (75 € MWh⁻¹), and maximum (133 € MWh⁻¹) semi-annual natural gas prices for non-household consumers with an energy consumption between 1000 GJ and 9999 GJ from 2019 to 2023 in Italy [67].

To resume and summarize the outputs of the various NPV analyses, the higher the specific fuel cost, the greater the convenience of such a system. In particular, taking into account an average price situation and a discount rate between 5 % and 10 %, the payback time amounts to around 5–7 years, which is considered feasible for this kind of investment. The other two fuels commonly used for the application, namely LPG and diesel, during the same period, showed a higher average cost, ranging between 78 and 113 € MWh⁻¹ for LPG [72–74] and between 103 and 188 € MWh⁻¹ for diesel [75–77]. In the event of a transition from these fuels to natural gas, it is also important to consider that the connection expenses for such capacity can amount to tens of thousands of euros.

3.4. Avoided CO₂ emissions

The overall avoided emissions when using the waste heat recovery system follow a similar trend to the monthly savings, aligning with the percentage of saved fuel. Considering that 5.6 kg of equivalent CO₂ is emitted for every kg of burned natural gas, it is possible to assess that the value of avoided emissions is approximately 102.6 tons of CO₂ per year for the configuration (400 cows and 18328 kg of natural gas saved), resulting in 257 kg of CO₂ avoided per cow every year.

4. Conclusions

In this study, the potential exploitation of waste heat by an energy-intensive thermal user was investigated and assessed. Specifically, a hydronic system for recovering waste heat from the air conditioning system of an enclosed barn was modeled and evaluated from both technical and economic perspectives. Initially, the amount of energy available was verified to be comparable with the energy demand of the forage dryer of 2672 kWh per cow per year. Once this assessment was completed, the amount of heat that can be recovered from the chiller condenser was quantified, and its exploitable potential was determined, resulting in (4960 ± 160) kWh per cow per year for the months and the daytime considered. The proposed solution demonstrates a significant reduction in fuel consumption for the operation of the dryer burner. The

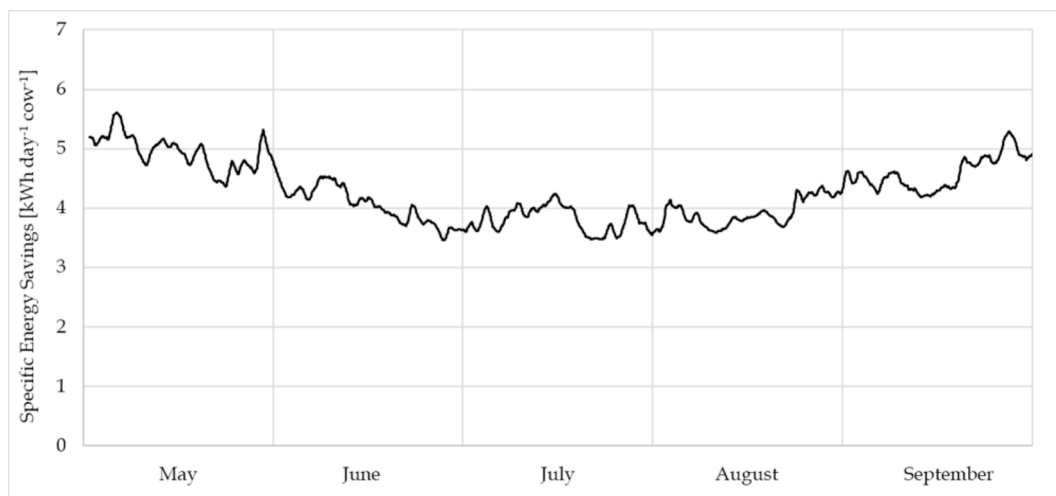


Fig. 8. Daily energy recovered over the drying months of the year.

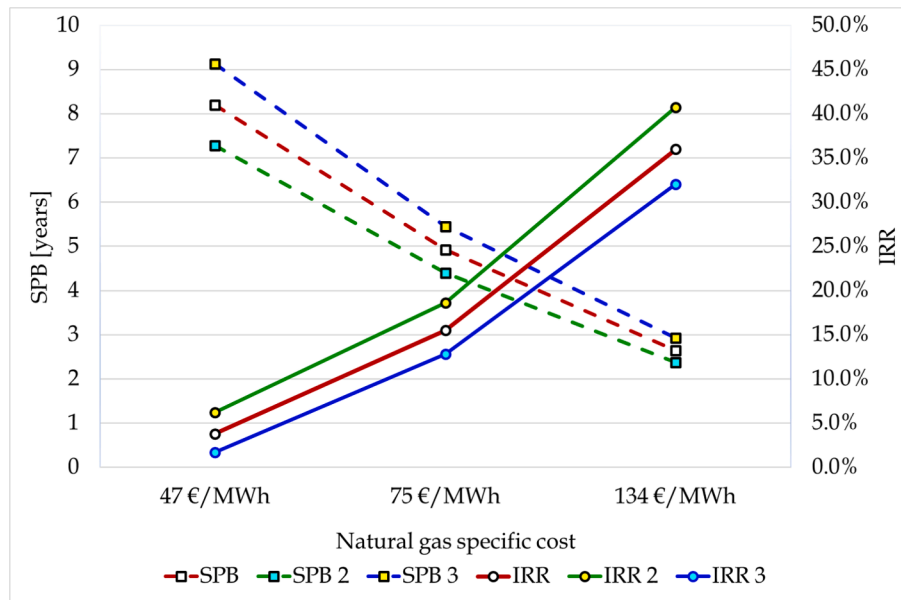


Fig. 9. Financial assessment of the investment, including sensitivity analysis on fuel and components cost.

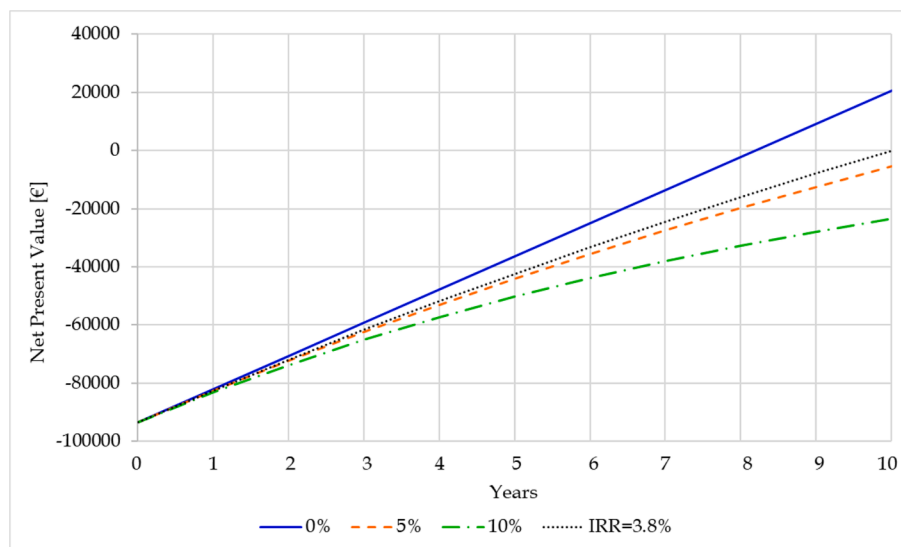


Fig. 10. Net Present Value of the investment with natural gas specific cost 47 € MWh⁻¹ and various discount rates (0%, 5%, 10%) and IRR equal to 3.8%.

calculated savings range between 11,260 € to 35,500 € every year depending on different natural gas prices (specifically, between 28.2 € cow⁻¹ and 88.8 € cow⁻¹ every year). This corresponds to a payback period of approximately 6 years, considering a discount rate of 5 % and an average natural gas price of 75 € MWh⁻¹. Furthermore, this system can have a highly positive environmental impact due to the 257 kg of equivalent CO₂ not emitted into the atmosphere every year for each cow. The synergy between barn cooling for animal welfare and fuel consumption reduction in the drying process emerges as a win-win solution that increases both productivity and energy efficiency. Moreover, incorporating such a circuit within the company's infrastructure provides a versatile way to manage waste heat. This can also be beneficial for other potential users typical in this kind of facility (e.g., pasteurization, as described by Cox et al. [78]). In this study, an investigation of the waste heat recovery system was conducted under specific climatic conditions, considering both temperature and humidity. It is essential to note that these conditions serve as a crucial framework for the analysis, providing a deep understanding of the system's performance within the

context of a specific geographical area. Future research could explore the adaptability of the waste heat recovery system across diverse climatic conditions, thereby enhancing the broader application of the study's outcomes.

Declaration of Generative AI and AI-assisted technologies in the writing process

During the preparation of this work the authors used ChatGPT-3.5 to improve language and readability. After using this tool/service, the authors reviewed and edited the content as needed and take full responsibility for the content of the publication.

CRediT authorship contribution statement

Marco Puglia: Writing – review & editing, Writing – original draft, Visualization, Validation, Supervision, Software, Methodology, Investigation, Formal analysis, Data curation, Conceptualization. **Saverio**

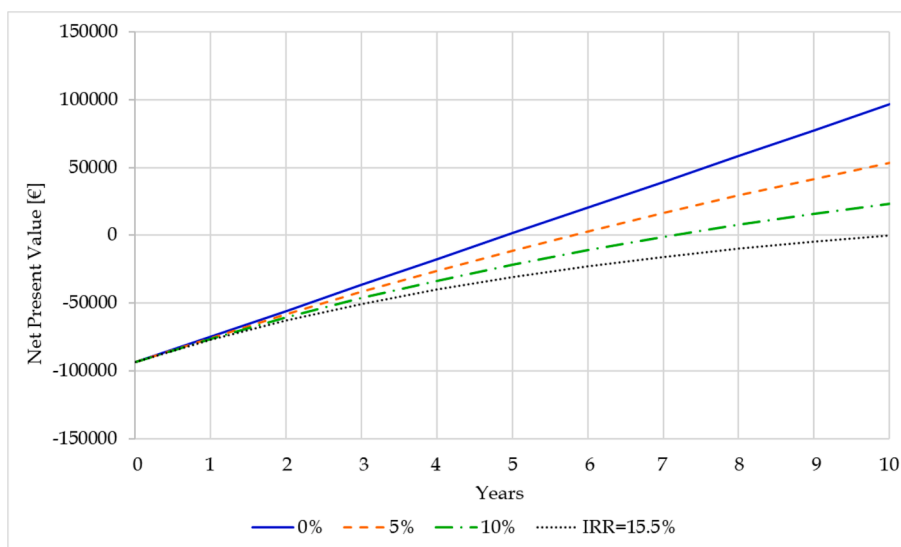


Fig. 11. Net Present Value of the investment with natural gas specific cost 75 € MWh^{-1} and various discount rates (0%, 5%, 10%) and IRR equal to 15.5%.

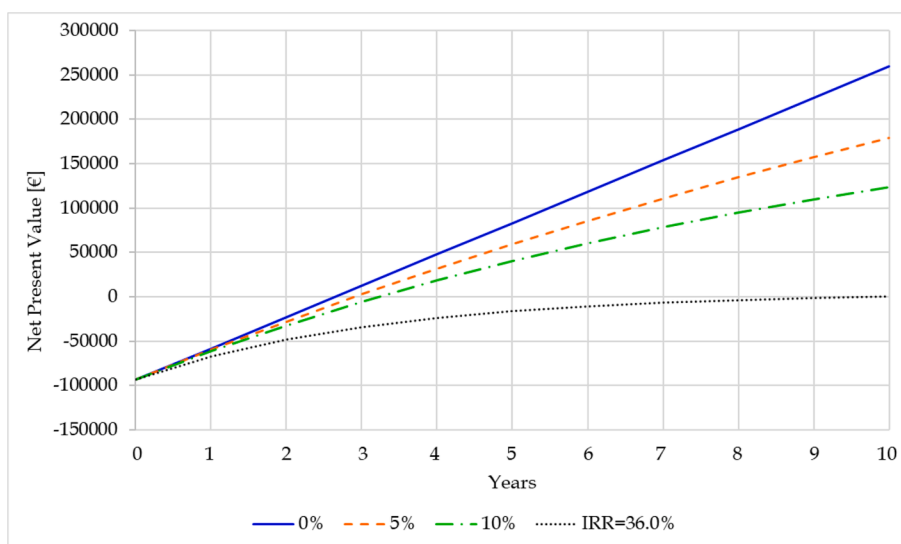


Fig. 12. Net Present Value of the investment with natural gas specific cost 133 € MWh^{-1} and various discount rates (0%, 5%, 10%) and IRR equal to 36%.

Mirandola: Writing – original draft, Visualization, Software, Investigation, Formal analysis, Data curation, Conceptualization. **Nicolò Morselli:** Writing – review & editing, Software, Methodology, Investigation, Data curation. **Simone Pedrazzi:** Writing – review & editing, Supervision, Software, Investigation, Conceptualization. **Alberto Muscio:** Supervision, Resources, Methodology, Conceptualization. **Paolo Tartarini:** Supervision, Resources, Project administration. **Giulio Alessina:** Writing – review & editing, Supervision, Project administration, Funding acquisition.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Appendix A. Barn heat load calculation

Q_{ext} was calculated as the sum of the heat transferred through the barn walls from the external ambient to the barn. The radiation contribution was

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taken into account using the sol–air temperature, according to the UNI/TS 11300 Standard [79] with Eq. (A.1):

$$Q_{ext} = S_e \bullet U \bullet (T_e + \Delta T_{sol} - T_i) \bullet t \quad (\text{A.1})$$

Where U is the overall heat transfer coefficient, t is the time step considered (1 h in this case) and ΔT_{sol} is the fictitious increment of the external temperature due to solar irradiation ($T_e + \Delta T_{sol}$ is the sol–air temperature). U was calculated as:

$$U = \frac{1}{\frac{1}{h_i} + \frac{l}{\lambda} + \frac{1}{h_e}} \quad (\text{A.2})$$

Where h_e , representing the external convective heat transfer coefficient, was approximated as [80]:

$$h_e = 4 + 4v \quad (\text{A.3})$$

Where v is the wind velocity at 10 m from the ground. ΔT_{sol} was calculated for every hour as:

$$\Delta T_{sol} = R_{ext} \bullet F \bullet \alpha \bullet I \quad (\text{A.4})$$

Where F is the view factor, considered 1 for the whole building, R_{ext} is the external resistance calculated as:

$$R_{ext} = \frac{1}{h_e + h_{rad}} \quad (\text{A.5})$$

and h_{rad} is the radiation heat transfer coefficient, calculated as:

$$h_{rad} = \varepsilon \bullet 4 \bullet \sigma_0 \bullet T_{av}^3 \quad (\text{A.6})$$

σ_0 is the Stefan-Boltzmann constant and T_{av} is the average temperature between the air temperature and the external surface temperature, and it is assumed always 5 °C higher than the air temperature.

Q_{air} was calculated as:

$$Q_{air} = ACH \bullet V \bullet \rho_{air} \bullet c_p \bullet (T_e - T_i) \quad (\text{A.7})$$

Where ρ_{air} and c_p are the air density and the air-specific heat at constant pressure, respectively.

Q_{cond} was calculated with Eq. (11):

$$Q_{cond} = m_{H_2O} \bullet h_{ev} \quad (\text{A.8})$$

Where m_{H_2O} is the mass of water that needs to be condensed and removed from the air entering the barn to achieve the proper relative humidity, calculated as:

$$m_{H_2O} = (\omega_e - \omega_i) \bullet ACH \bullet V \bullet \rho_{air} \quad (\text{A.9})$$

ω_e and ω_i are the humidity ratios of the external ambient and inside the barn. Both were calculated with the following formula [81]:

$$\omega = 0.622 \bullet \frac{RH \bullet p_{sat}}{p_{amb} - RH \bullet p_{sat}} \quad (\text{A.10})$$

p_{amb} is the ambient pressure, assumed to always be 101325 Pa, and p_{sat} is the saturation pressure at the specific temperature considered (24 °C inside the barn, T_e outside the barn). When the m_{H_2O} resulted negative for specific conditions, it was assumed zero (no water addition was considered).

The hourly Q_{met} was assumed to be 0.9 kWh for each lactating cow. This value was derived from a literature review analysis. Saha et al. [82] quantify the energy of heat production along with tissue deposition as the difference between gross energy nutrient intake and the energy not excreted in feces, urine, methane or milk amounting to 15.21 Mcal day⁻¹, corresponding to an average heat power of 0.74 kW. Morris et al. [83] calculated a median heat production value of 22.1 Mcal day⁻¹ for lactating Jersey cows, namely 1.07 kW. Berman et al. [84] affirmed that the skin water loss can reach up to 1.5 kg h⁻¹, which means a heat dissipation of about 0.95 kW.

Once the hourly Q_{tot} was calculated for each of the 8760 h of the year (average value of the four years considered), it was possible to calculate both the yearly heat load and the heat load of the months when the dryer is working, namely from May to late September.

Appendix B. Heat transfer fluid

Three different fluids were evaluated for the heat recovery based on their thermal properties:

- Water: as the basic fluid, it would be suitable for this application. However, its freezing temperature can pose a risk during winter months. Furthermore, it has a low boiling temperature, which may not be problematic in this specific case, but challenges may arise if the hydronic system is integrated with other applications that release heat at higher temperatures (e.g. cheese factory). The risk is the presence of a double-phase fluid with unpredictable behavior and difficulty functioning that would compromise the integrity of the circuit [81].

- Diathermic oil: this fluid would be suitable for smaller applications because of its high viscosity (nearly two orders of magnitude higher than water) and its toxicity. A possible leak in the circuit would be critical [85].
- Mixture of water and propylene glycol (*glycol mix*): this mixture combines the thermal properties of water (high conductivity and specific heat) with an increased boiling point at atmospheric pressure (more than 150 °C) [86]. Moreover, glycol prevents the mixture from freezing, even during cold temperatures.

The mixture of propylene glycol and water (in a volume percentage of at least 25 % glycol over water) was the most suitable fluid due to its thermal properties closely resembling those of water (Table B.1.). In addition, this mixture is completely nontoxic and a possible leak is not dangerous. Moreover, the low viscosity (similar to that of water) is not too energy-demanding for the pumping unit.

Table B1

Fluids thermal properties (referred to Std. conditions) [81].

Fluid	Specific Heat [J kg ⁻¹ K ⁻¹]	Thermal Conductivity [Wm ⁻¹ K ⁻¹]	Density [kg m ⁻³]	Kinematic Viscosity [mm ² s ⁻¹]
Water	4186	0.56	1000	1.10
Diathermic oil	1800	0.12	840	70
Glycol mix 50 %	3860	0.43	1028	3.08

The *glycol mix* (50 % volume [59]) was chosen and modeled taking into account its dependence on properties with respect to temperature (Fig. B.1.) at each step of the simulation. Due to the narrow pressure range that the circuit provided (maximum 4 bar), the properties were considered only temperature-dependent: $\rho = \rho(T)$, $c_p = c_p(T)$, $\nu = \nu(T)$, $\lambda = \lambda(T)$:

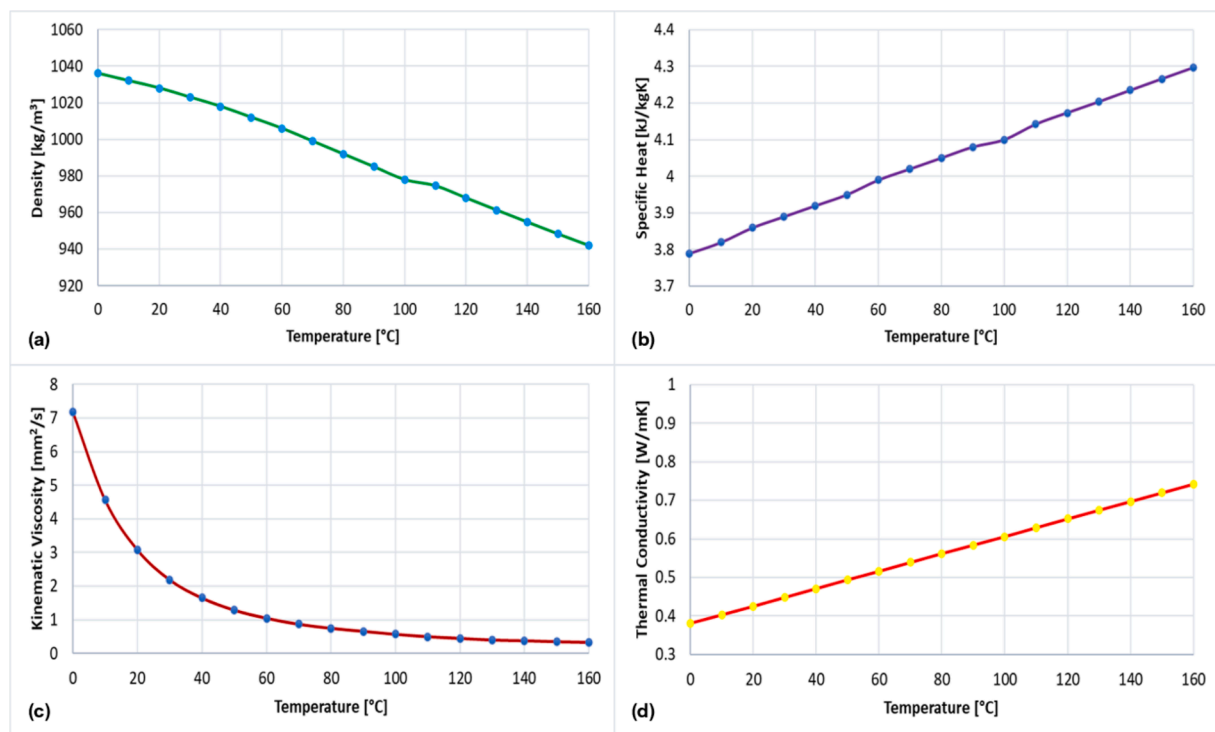


Fig. B.1. Glycol mix density (a), specific heat (b), kinematic viscosity (c), conductivity (d) trends over temperature.

Data availability

Data will be made available on request.

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