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Jalel Labidi

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Integration of a solar thermal system in a dairy process

José Antonio Quijera, María González Alriols [1], Jalel Labidi*

Chemical and Environmental Engineering Department, University of the Basque Country, Donostia-San Sebastián, Spain

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ABSTRACT

Any analysis of the current energy world scenario draws on the combination of energy efficiency improvement and the use of renewable-type energies. The industrial use of renewable energies is not still well established as they present several problems that generate insecurity in this sector. Some of the renewable energy resources work intermittently (like the sun or wind) and the energy they provide is, often, of low intensity. Solar thermal technology has been successfully introduced in domestic applications and buildings. Many industrial processes work in temperature intervals where solar thermal technology would be able to supply an important amount of the total energy input at an acceptable price. Based on mathematical modeling, this work evaluates the viability of integrating a solar thermal system to the conventional energy structure of a dairy plant in the Atlantic side of Spain. Pinch methodology has been used to develop the integration of the solar subsystem in the energy installation of the plant. In order to determine the potential of the solar thermal energy, several hypotheses and scenarios were analyzed, based on real cases of the productive process. As a result, it could be stated that the solar thermal energy potential for the studied industrial process, operating at low and middle temperatures, was considerable, and must be taken into account as an energy option.

1. Introduction

Nowadays, some of the more significant environmental problems have their origin in an inappropriate policy and management of the energy resources. Particularly, fossil fuels are in the spotlight of the energy future as they are non-renewable resources that could be combined with and/or substituted, by other kind of environmental, economical and socially sustainable resources [1]. Any analysis of the current energy world scenario draws on the combination of energy efficiency improvement and the development and implementation of renewable-type energies with their associated technologies.

It does not seem to be feasible to overcome the current energy crisis working only in one of these directions. Technologies to achieve levels of energy efficiency are being implemented in many processes and industries. The utilization of renewable energy in industrial activities has not achieved considerable success for the moment [2]. However, its utilization in some sectors is well consolidated. For example, wood drying consumes, normally, wood waste (biomass) generated in the same industrial plant.

As an example of this type of technology applied to drying processes, it can be mentioned the work of Gustafsson [3] about the optimization of drying kiln operation (firing wood cheeps and using electricity) in Sweden. Although combustion is the most extended energetic use of biomass, other technologies, as the gasification is attracting more and more interest [4]. Photovoltaic technology has been industrially implemented to provide electricity. As an example, Bazen and Brown expose the experience of a poultry activity in United States, where the plant is electrically feed by photovoltaic cells [5]. The industrial use of renewable energies is not still well established as they present several problems (like intermittency and low intensity) that generate insecurity in this sector [6].

Solar thermal technology has been successfully introduced in domestic applications and buildings. Nevertheless, at industrial scale, there is yet a considerable and not enough explored field of development and implementation of this technology. Some installations have been done, for example in dairy factories. Schnitzer and Gwehenberger [7] present an interesting example in Austria, which was utilized by the authors to introduce some appropriated tools for its integration in the energy structure of the plants. The case presented by Atkins et al. [8] exposed the utilization of solar thermal technology, integrated by pinch analysis, in a dairy milk powder plant of New Zealand. Furthermore, Schweiger et al. [9] analyzed the viability of this energy technology in Spain and Portugal with some

* Corresponding author. Tel.: +34 943017178; fax: +34 943017140.
E-mail addresses: maria.gonzalez@ehu.es (M.G. Alriols), jalel.labidi@ehu.es (J. Labidi).
[1] Tel.: +34 943018740; fax: +34 943017140.

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examples in milk industry, brewing, malting, textile and paper industries, and other processes, in which solar energy supplied different amounts of energy for the studied processes. The European Solar Thermal Industry Federation presented some European examples of thermosolar application with high efficiencies, based on the production of hot water, steam and cold for washing, drying, distillation and other chemical processes [10]. Nowadays, the improvements in solar technology allow its implementation in unfavorable geographical and climatic locations [11].

Many industrial processes work in temperature intervals where solar thermal technology would be able to supply an important amount of the total energy input at an acceptable price. Advanced flat-plate collectors and evacuated tube collectors are often enough to produce water, steam and cold, at thermal levels, more or less, between 60 and 90 °C, or 90–160 °C, respectively. The last mentioned technology can generate energy from sun in climatic conditions where the diffuse component of the total radiation is the main contribution [12]. The Basque Country, and specially its north side (like all the north face of Spain and the Atlantic front-side of Europe), is a region where, statistically, beam radiation is lower than diffuse sun radiation all the year. Industrial use of solar thermal energy is not normally being considered in this area due to climatic and meteorological conditions [13]. Starting from this point, this analysis deals with the possibility of supplying solar thermal energy, in the form of hot water, to an industrial dairy process, reducing its natural gas consumption, in a zone where diffuse radiation is the main component of sun radiation.

2. The process

The dairy plant is located in the Atlantic side of the Basque Country (latitude = 43° N, longitude = 002° W). The plant receives every day about 20 000 L of cow milk from farmers from the nearby. About 15% of the raw milk goes to yoghurt production, an 80% is utilized to elaborate cheese, and the remaining quantity is used for the production of non-fermented milky drinks. When milk arrives to the factory, it is conserved in cooled tanks at 4 °C. The productive process through dairy chain begins at the morning with the pasteurization of the milk, in a pasteurizer (plate heat exchanger) working in continuous at 95 °C and 15 s of retention (HTST – High Temperature/Short Time- technology). Pasteurization of all the milk takes place from 08:00 h to 13:00 h. Connected to the pasteurizer, there is an economizer (plate heat exchanger) where outgoing milk exchanges heat with incoming one. Pasteurized milk is cooled in a refrigerating unit from 42 to 34 °C. From here, pasteurized milk is send to the yoghurt fermentation unit kept at 42 °C and operating form 08:00 h to 12:00 h. This operation takes place in an electrically heated and closed vessel for 3 h. Milk (and same other ingredients) is distributed in plastic recipients that are closed and introduced into the incubator. At the same time, production of non-fermented milky drinks is carried out (pasteurized milk with whey and some other components, like fruits). Some time later, the biggest portion of pasteurized milk is sent to two cheese tanks. Coagulation of milk takes ~~places~~ from 09:00 h to 15:30 h, at a temperature of 34–35 °C. Elaborated yoghurt, cheese and drinks are stored in cooled rooms at 4–10 °C (according to the needs of each product), waiting for commercialization.

Pasteurization is the most energy intensive operation of the production process. Tap water is heated in a gas boiler, from 12.2 °C (annual average temperature) to 95.0 °C, by natural gas. Tap water stored in a 50 m^3 capacity tank is used as cooling utility in different heat exchangers and basic operations. Water from the pasteurization unit is send to this tank at 78.0 °C, stored and later distributed to other exchangers and operation units when hot water is required during daily production. Once the productive time is finished, the water tank is emptied and cleaned, as well as the rest of the plant equipment. A big amount of exceeding water is used in cleaning operation of the plant and the equipment. Fig. 1 shows a diagram of the current dairy plant (temperatures are given as annual average values). Mass and energy flows have been established based on real operating conditions of the plant. Table 1 presents the sequence of mass flows.

It is important to emphasize the big amount of water (before being heated at the boiler and used at the pasteurizer), which is not reutilized at any other operation. This hot water is discharged every day at the end of the productive journey, losing its heat content. As previously mentioned, hot water for plant basic operations is produced in a gas boiler. Table 2 presents the heat consumption of the main operations. The current operation requires an annual amount of energy of 1584 MWh/year.

The reutilization of hot water from pasteurization in cleaning and other operations like milk coagulation, would reduce the energy consumption in 789.1 kWh/batch. This reduction can be achieved due to the inclusion of the tank of water from pasteurization. The saving of energy could reach the value of 680 kWh/batch.

3. Results

3.1. Energy analysis of the process and opportunities for optimization. Scenario 1

Pinch analysis is a good and well-tested tool which permits to understand the performance of heat exchange network, explaining the way that heat and cold flow through the plant and identifying operations where energy can be recovered or better employed. Pinch analysis has been applied successfully for processes operating at low temperatures (like dairy activities) [7,8]. In order to consider the viability of a solar thermal installation, it is necessary to evaluate the efficiency of the current energy system and to visualize any opportunity for improvement. Table 3 presents the streams database at dairy industry. To carry out the pinch analysis, some assumptions have been made:

- The water storage tank mitigates time phase lags between operations that take place at different moments of the day, connecting them and distributing hot water when it is needed.
- During operation, dead times are not significant in comparison to activity times; in practices, time event chart and equipment occupancy chart result coincident.

In order to understand any relation between streams and their temporal connections, initially, all hot (*H*) and cold (*C*) streams were considered. This includes the cleaning hot stream (*12H*), and the surplus water (*13H*) that is evacuated every day from the water tank to the sewer system (at 25 °C). Pinch temperature was found to be 60.5 °C for ΔT_{min} = 10 °C. Minimum energy requirements for heating and cooling were 268 kW and 260 kW, respectively. These results were obtained considering an idealized situation, with maximum heat transfer between streams (maximum heat recovery, *MHR*), assuming that all the streams were coincident in time, and exploiting the cleaning and excess hot water streams (*12H* and *13H*). In fact, this configuration is not realistic with the actual design of the process since the heat from cleaning stream is very difficult to recover. Presently, the excess hot water is not reutilized and its thermal value is lost. So, it would be more realistic not to consider both streams for the analysis. This way, the graphic suggest a threshold problem, a special situation where no cold water would be needed and only hot water would be required. Heat

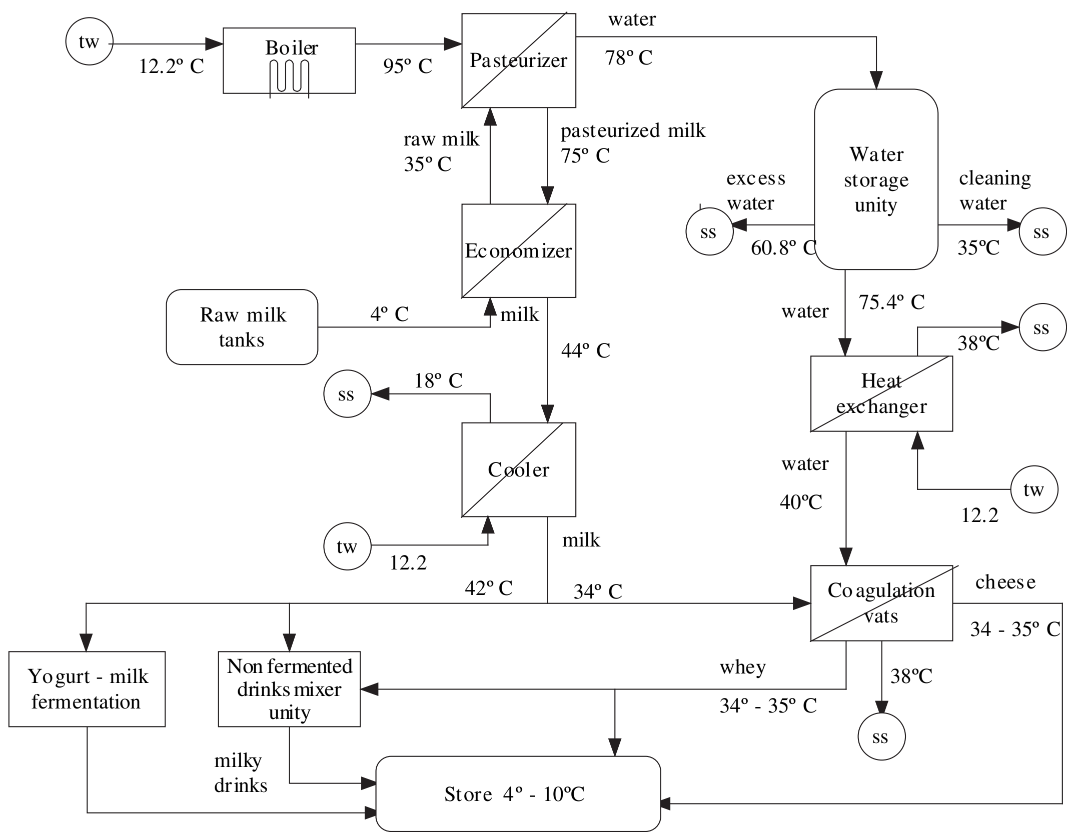

Fig. 1. Scheme of the dairy facility (tw: tap water; ss: sewage system).

demand was then established in 867 kW, in agreement with the mass and heat balances results (880 kW).

At last, it is very interesting to compare the shift grand composite curves (*SGCC*) of the threshold problem, with an intermediate situation, in which, the impossibility of recovering any energy from the cleaning water stream afterwards its use was assumed, but, at the same time, considering the opportunity of reutilizing the stored hot water from the tank instead of getting rid of it every day.

Fig. 2a and b show the *SGCC* for both situations. Fig. 2a presents the threshold, where only heat from the boiler is needed. On the other hand, the presence of a big pocket of energy (hot water) stands out. The area of this heat pocket quantifies the daily energy loss and the opportunity of assigning this heat to other uses and operations in the plant, under a temperature of 57.4 °C.

As a threshold problem, the integration of a solar thermal system would be, theoretically, suitable at temperatures higher than 9.0 °C, but in practice, it would be more suitable higher than 57.4 °C (below this value, there is a considerable amount of hot water in the tank).

In addition to the results of the pinch analysis, it should be taken into account other extern variables which could affect the energy performance of the process, like temperature variability through the year. While incoming water temperature goes down in winter, the boiler heat demand increases, and the opposite happens in summer. Required heat from the boiler fluctuates from 914 kW in January, to 840 kW in July (results of the energy balance; annual average of 880 kW, and 4401 kWh/batch).

Table 1
Data base of mass flows in the dairy process.

Operation	Δt (h)	Flow type	m (kg/batch)
Pre–treatment of milk and pasteurization	5.0	Raw milk	20,600
		Pasteurized milk	20,497
		Material losses at pre–treatment	103
Yogurt elaboration	3.0	Milk to fermentation unity	3075
		Produced yoghurt	2982
		Material losses at fermentation	92.2
Cheese production	4.5	Milk to cheese elaboration unity	16,398
		Produced cheese	1804
		Whey	14,594
Non-fermented milky drinks	1.0	Produced milky drinks	1751
Cleaning of installation	2.0	Water for cleaning operations	16,000
Extern utilities of heating and cooling	5.0	Water at boiler	45,701
		Water for cooling operations	37,540
Excess of water	9.5	Non reutilized water from pasteurization	22,158

3.2. Optimization of the process for solar thermal energy implementation. Scenario 2

Two options for the improvement of the current process energy efficiency were considered: the first one, deals with the possibility of supplying energy to the fermentation unit by the hot water accumulated in the tank and the second option consist of the reuse

Table 2
Energy utilization at the current dairy activity.

Operation	Δt (h)	Hot water demand (kg/batch)	$T_1 \rightarrow T_2$ (°C)	Heat flow (kW)	Heat load to process (kWh/batch)
Pasteurization	5.0	45,701	95.0 → 78.0	−181	−904
Milk coagulation	4.5	7543	40.0 → 38.0	−7.8	−35.1
Cleaning	2.0	16,000	65.0 → 25.0	−377	−754
Boiler	5.0	45,701	12.2 → 95.0	−880	−4401

Table 3
Streams data base at dairy industry.

Stream		t_1 (h)	t_2 (h)	m (kg/s)	CP (kW/°C)	T_1 (°C)	T_2 (°C)	$\dot{Q}$ (kW)	Q (kWh)
Number	Type								
1	C	08:00	13:00	2.5	10.6	12.2	95.0	880	4401
2	C	08:00	13:00	1.1	4.4	4.0	35.0	136	6780
3	H	08:00	13:00	1.1	4.4	75.0	44.0	−136	−680
4	C	08:00	13:00	1.1	4.4	35.0	75.0	176	877
5	H	08:00	13:00	2.5	10.6	95.0	78.0	−181	−904
6	H	08:00	13:00	1.0	3.8	44.0	36.0	−30.3	−151
7	C	08:00	13:00	1.0	4.1	12.2	18.0	23.9	119
8	H	11:00	15:30	0.5	1.9	73.3	40.0	−65.0	−292
9	C	11:00	15:30	0.6	2.6	12.2	38.0	68.2	307
10	C	11:00	15:30	1.0	3.9	34.0	35.0	3.9	17.5
11	H	11:00	15:30	0.9	3.9	40.0	38.0	−7.8	−35.1
12	H	16:00	18:00	2.2	9.3	65.5	25.0	−377.0	−754
13	H	18:00	20:00	3.1	12.9	62.5	25.0	−481.9	−964

of the daily excess of water in the process the following. The non-reutilized water stream volume was 22 158 L/day and its temperature at the end of the working day was 62.4 °C (at 18:00 h; annual average temperature).

3.2.1. *Energy for milk fermentation*

Fermentation unit consists of an electric closed incubator. The average actual consumption of electricity is 4.0 kW (12.0 kWh/batch). A flow of 115 kg/h (344 kg/batch) is required to replace the electricity consumption by hot water coming from the tank. This flow is needed from 09:00 to 12:00 h, at a temperature of 76.4 °C (at the beginning of the operation), and it means a. requirement of 2.2 kW (6.6 kWh/batch), in the form of water from pasteurization. When water is needed for fermentation, the water reserve is big enough to supply the demanded quantity to this unit as well as to the rest of the process operations.

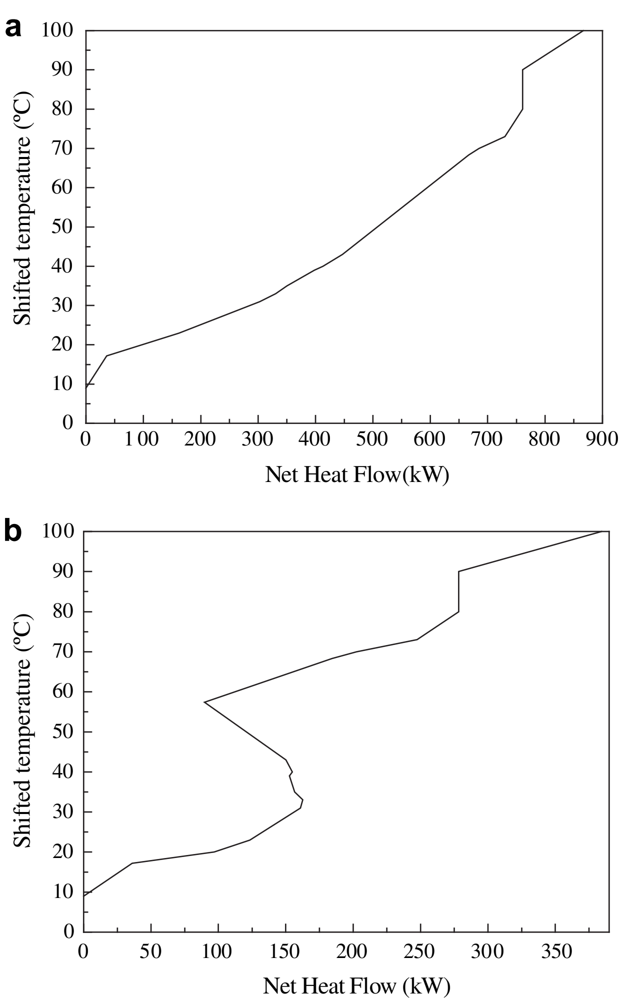

Fig. 2. a) Grand composite curve: streams number *12C* and *13C* are not considered for heat transfer. b) Grand composite curve: stream number *13C* is considered for heat transfer.

3.2.2. *Complete reuse of pasteurization hot water in the process*

Considering the volume of water assigned to a by-jacket fermentation unit, at the end of the working day, there would be a volume of 21 814 L of hot water stored in the tank that, at the beginning of the following working day, new production time, would have a temperature around 40.6 °C. In order to study the possibility of using this water from day to day, two options were investigated:

a) Direct feeding: in this option, all the stored water is fed into the boiler. This water can be used from 08:00 h until 10:23 h (from 40.6 °C to 37.0 °C). After this moment, fresh water is fed to the boiler at an annual average temperature of 12.2 °C. The energy requirement of the boiler would be 740 kW (3699 kWh/batch). This configuration would require high operation flexibility and the boiler would have to present a fast response to temperature changes (from 37.0 °C to 12.2 °C).
b) Mixed feeding: during the boiler operation time (5 h), it would be possible to mix hot water from the tank with fresh water, giving a mixture at 23.9 °C. For this option, the heat flow required would be 756 kW (3780 kWh/batch).

The energy consumption would be 1.85% lower in the direct feeding option. Nevertheless, the mixed feed would be more suitable under a technical point of view.

On the basis of this choice, the mass and energy balances were established considering the reutilization of all the water of the tank in the same day (this includes the supply to the fermentation unit), and following day to feed the boiler. The energy balance showed that the annual average heat flow was 756 kW (3781 kWh/batch), improving the energy efficiency of the plant in a 14.10% and providing a daily saving of 622 kWh/batch, with high stability of the working process and low investment.

To reinforce the viability of utilizing some hot from the stored water from one day to another, Fig. 3 exposes the thermal interactions between water needed in operations and the evolution of the temperature of the water stored inside the tank through the time. As can be seen, the use of the energy contained in the water from pasteurization in other operations of the process (fermentation,

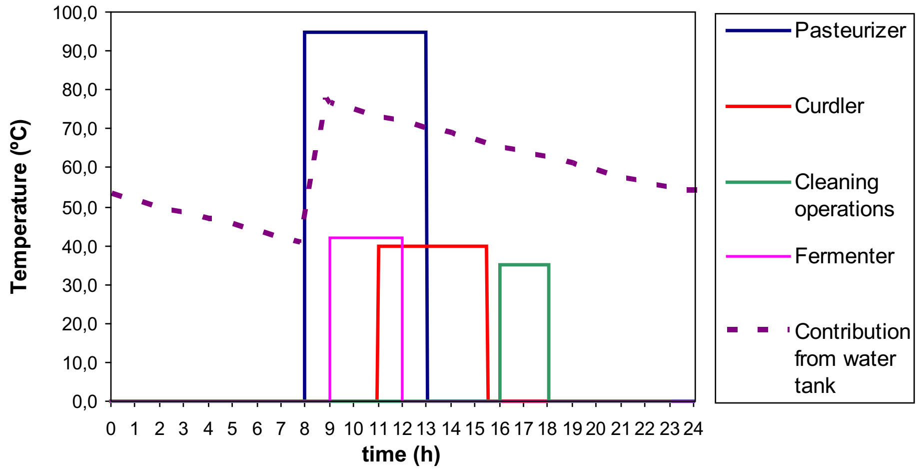

Fig. 3. Evolution of the water temperature inside the tank while its hot water is used in the operations of the process.

coagulation and cleaning at the same day, and pasteurization next day), has been found to improve the plant energy efficiency as well as to reduce the water consumption. In fact, the implementation of those adjustments in the working design, could reduce the energy demand (hot water) from 1584 MWh/year to 1361 MWh/year, which entails the saving of 223 MWh/year.

3.3. Viability of a solar thermal system. Scenario 3

3.3.1. Solar radiation in the zone

Once the energy needs of the original and the optimized processes were determined, the next step was to consider the viability of incorporating the solar thermal energy in the process. According to the data published by the Basque Entity of Energy (EVE) [13], the solar radiation received in the zone where the dairy plant is located is between 1500 and 1700 h/year. In terms of a surface of 1 m^2 surface, oriented to the south and tilted 43°, total irradiation varied from the annual minimum of December, 2248 $kWh/m^2 \cdot day$, to the maximum of July, 5353 $kWh/m^2 \cdot day$ [14]. The accumulation of total radiation at the end of the year was 1494 $MWh/m^2 \cdot year$.

A key characteristic of the solar radiation in this area is the importance of the diffuse irradiation in comparison with the beam irradiation. Along the year, the diffuse irradiation contributes more than beam one does to the total amount of radiation. This fact must be taken into account for the selection of the appropriate solar technology. It is also important to consider the heat requirement pattern in the process and the solar radiation distribution in the area throughout the hours. Considering the heat flow demand at each operation unit throughout a day and examining the total solar radiation (in a daily annual average), received on a horizontal arbitrary surface of 1000 m^2, it is possible to correlate, at the same time, both solar energy demand and solar contribution; the bigger the overlap between energy requirement and the total irradiance, the better convergence between provision and demand. Fig. 4 displays the degree of overlapping, as a function of the total irradiance. The water deposit in the optimized system, which would help to minimize phase lags, would need to be adapted to the solar thermal energy.

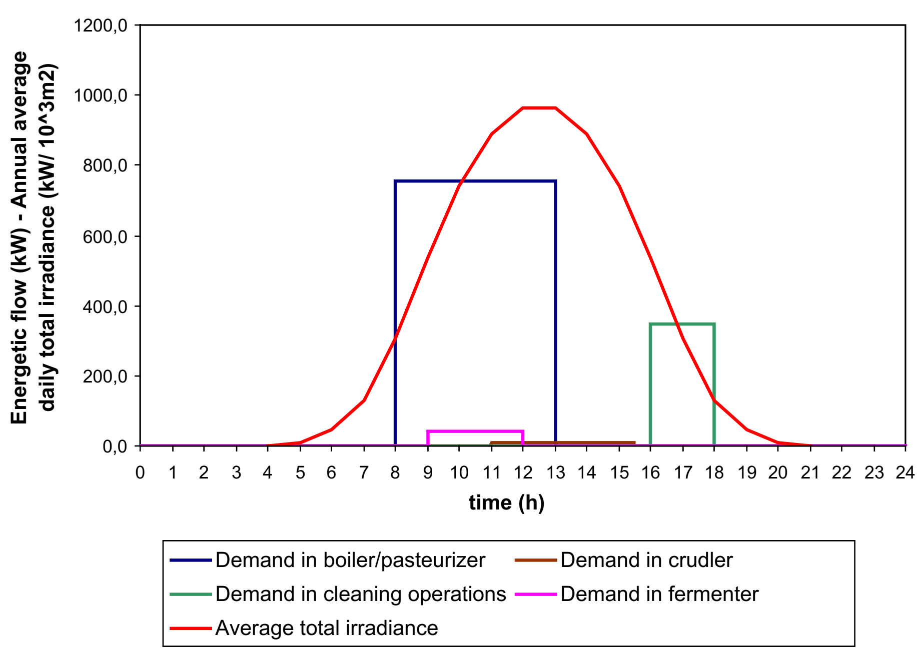

Fig. 4. Daily overlap between times of heat demand and solar irradiance.

Other interesting aspect would be to associate the boiler energy requirements along the year with the total solar irradiation received on a horizontal surface of 1000 m^2. Fig. 5 shows the overlap between the current energy supplied by the boiler and the total solar radiation received on the exposed area. The Radiation received on the surface was not able to fit the energy needs all the time. However, it was found to be substantial enough to ensure the demand of more than a half of the days of a year. Moreover, this contribution was greater than the requirement most of the year (the surface received more energy than the process needs). A high degree of convergence was found between insolation time and the process demand of hot water, as well as a notorious opportunity to provide much of the energy needed to sustain the process through a solar field.

3.3.2. Outline of the suitable solar system

As diffuse irradiation was found to be quantitatively more important than beam irradiation, it was necessary to select a solar collecting technology which focused on this radiation type. Evacuated tube heat pipe collectors do an appropriate use of solar diffuse irradiance and, also, they work optimally in situations where medium temperatures are needed. They usually present high enough optical gain values while their thermal loss factors (linear and square) are usually smaller than conventional or advanced flat-plate collectors ones. Furthermore, their performance efficiency is acceptable in zones with high ambient humidity and low winter temperatures. They are usually more expensive [15]. Some specifications of the selected commercial collector are exposed in Table 4. Equation (1) gives the characteristic equation of the chosen commercial collector:

$$\eta = 0.81 - 1.20 \cdot \frac{T_c - T_a}{G} - 0.007 \cdot \frac{(T_c - T_b)^2}{G} \qquad (1)$$

From this equation, it is easy to understand that its efficiency is higher when total irradiance increases, but it falls as the difference between average collector-temperature and the effluent temperature of primary fluid rises.

The combination of the current energy installation (based on the current natural gas boiler) and the solar thermal unit would result in a system with the following components and characteristics:

- A two-tank forced-circulation system whose primary heat transfer fluid would be a mixture of water and propylene glycol ($c_p = 3.889$ kJ/kg K) and the secondary fluid tap water from the original accumulator tank ($c_p = 4.187$ kJ/kg K).
- A solar field, consisting of high performance vacuum tube heat pipe collectors of low loss ratio. The solar energy would be exploited during the daily insolation time and, then, turned out until the next insolation period.
- A heat exchanger, with an efficiency of 0.95, which would transfer the incoming energy from the primary fluid to the secondary one.
- The auxiliary heat equipment would be the current natural gas boiler.
- Two tanks would be working in the installation; the current deposit would be assigned for pasteurization, and a second one would supply water to other operations of the dairy plant.

Before modeling the new proposed energy system, some considerations about water accumulation tank were done. The storage unit was considered as a stratified tank with three levels. To keep the water temperature at its outlet value from the pasteurizer, the boiler intervention would be required more intensively. Considering the involvement of a solar thermal system, it could provide power for more hours throughout the day (almost as many as hours of sun light are). Considering the needed water flow for other operations, there would be a deficit, daily replaced by tap water, and then, mixed with hot water returned from the pasteurizer to obtaining an average temperature of 46.1° C at the lowest level of stratification of the tank (this was the temperature of the secondary fluid for the heat exchanger). At the same time, heated water would enter to the upper level of stratification, from which it could directly be extracted and send to the boiler. From the last hour of sun lighting, hot water could be stored and used next day. At last, there would be an intermediate level where a themocline could develop.

For the new configuration, the boiler annual average demand could fall from the 3780 kWh/day required by the optimized system (based solely on the use of the boiler for 5 h/day) to 2732 kWh/day. The new design provided an average decrease of fossil fuel consumption of around 27.7%. The annual energy consumption could decrease from 1361 MWh/year for the optimized system based only on natural gas, to 984 MWh/year for the solar thermal integrated system. Fig. 6 presents the scheme of the solar-conventional energy system for the dairy plant. Table 5 presents the basic parameters adopted for the thermal feasibility study.

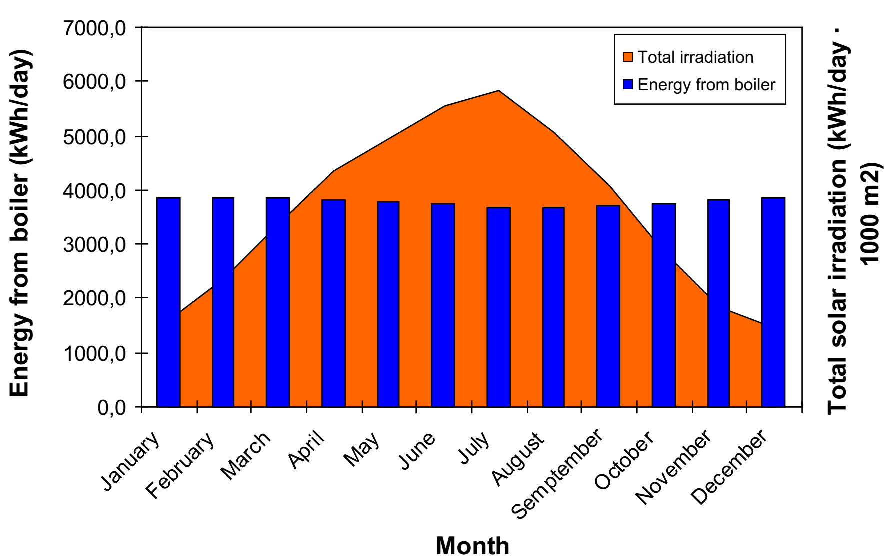

Fig. 5. Annual overlapping between demand of energy and solar irradiation a long the year.

Table 4
Some characteristic of the chosen commercial evacuated tube – heat pipe collector.

Parameter	Value	Unit
Total area of collector	4.237	m^2
Absorber area	3.000	m^2
Optical efficiency	0.81	–
Linear loss coefficient	1.20	$W/m^2\cdot K$
Square loss coefficient	0.007	$W/m^2\cdot K$
Maximum working temperature	130	°C
Specific flow	0.017–0.042	$kg/m^2\cdot s$
Primary fluid mixture	Water–propylene glycol (35%)	–
Incidence angle modifier	0.97	–

Table 5
Mass flow and temperature basic parameters for the modeling of the energy system of the dairy process.

Equipment	Parameter	Value	Unit
Gas boiler	Water flow	45,701	kg/day
	Outlet water temperature	95	°C
	Operation time	5	h
Pasteurizer	Inlet water temperature	95	°C
	Outlet water temperature	78	°C
Solar accumulator	Accumulation volume	45,701	kg
	Flow of water from pasteurizer	23,543	kg/day
	Flow of water for the boiler	9141	kg/h
	Flow of tap water	22,158	kg/day
	Outlet water temperature	95	°C
Water secondary tank	Accumulation volume	22,158	kg/day
	Inlet water temperature	78	°C

3.4. Modeling of the solar-conventional thermal system energy performance

The feasibility of the solar thermal installation for this process was evaluated for the following scenarios: a structure installation in parallel, where different variables and operating parameters were modified, and series design with two, three and four rows of collectors. Some simplifications were previously considered:

- Collector modules were oriented to the south.
- Collectors were tilted 43°.
- No shadows were taken into account.
- Physical space for collector disposal was not limited.

3.4.1. Mathematical models for the solar field performance

The basic equation to evaluate the heat amount that a collector can contribute is the characteristic equation of the device. This equation carries out the collector heat balance, where the maximum amount of heat that can be achieved from the sun, and the thermal losses, are related. The reduction of the available heat is related to the primary fluid inlet temperature to the collector, the ambient temperature, and the total irradiance. This mathematical proportion is established by two coefficients (linear and square). Furthermore, when this equation is going to be utilized to evaluate the collector performance in a real context, it is recommended to include an incidence angle modifier (Equation (2)) [16,17]:

$$\eta = \eta_o \cdot K_\theta - \frac{a_1(T_c - T_a) + a_2(T_c - T_a)^2}{G} \tag{2}$$

The efficiency given by this equation takes into account the performance of 1 m^2 of absorber area, so it is possible to establish an equation considering the net absorber area of the solar field. This equation would be appropriated to determine the total amount of heat transferred to the primary fluid in the field. It would be also advisable to include the incidence angle modifier into the equation (Equation (3)):

$$\begin{aligned} Q_{sf} &= \eta \cdot K_\theta \cdot G \cdot A_{sf} \\ &= A_{sf}\left(G \cdot \eta_o \cdot K_\theta - a_1 \cdot (T_c - T_a) - a_2 \cdot (T_c - T_a)^2\right) \end{aligned} \tag{3}$$

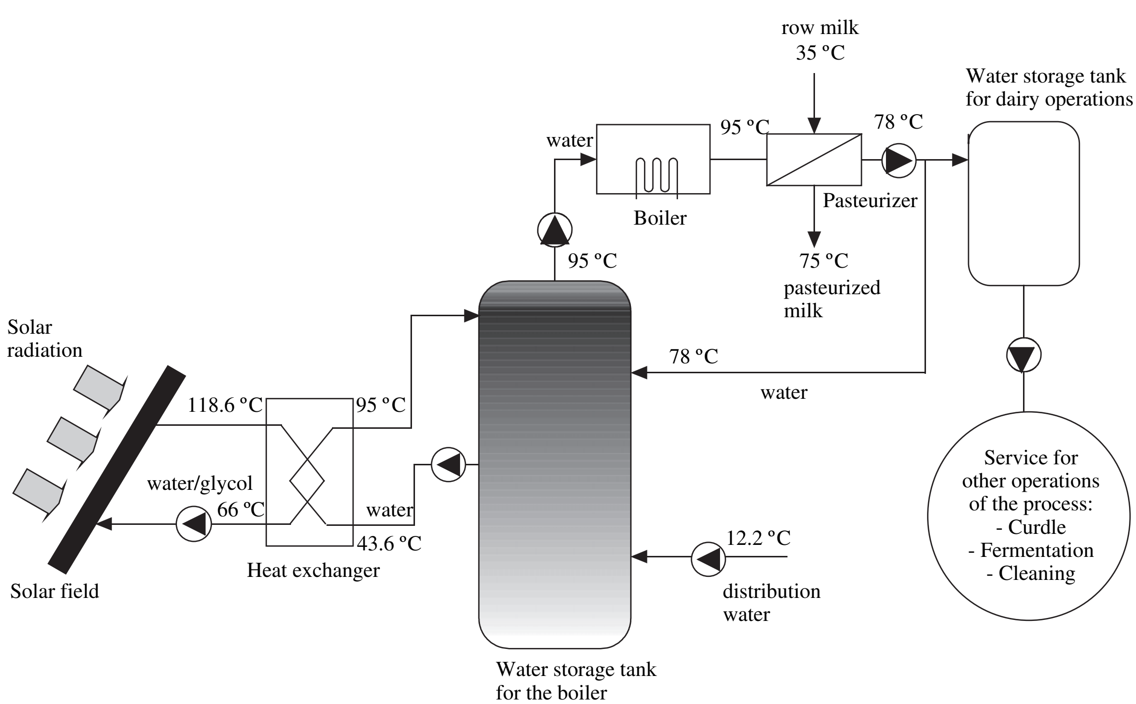

Fig. 6. Scheme of the proposed design energy contribution system for the dairy plant.

The average collector internal fluid temperature could be estimated by Equation (4), as a function of inlet and outlet temperatures of the primary fluid:

$$T_c = \frac{T_i + T_o}{2} \tag{4}$$

When series connections in the solar field are considered, the influence of primary fluid distribution should be taken into account. The most important contribution of the arrangements series to the global performance was the increment of the effluent temperature from the solar field but, at the same time, the fluid flow decreases. The flow decrease was related to the number of row of collector disposal in the series. The flow of the primary fluid which the pump should move could be calculated by Equation (5), valid for connections of the collector module arrays in parallel as in series:

$$m_{pf} = \frac{m_c \cdot A_{sf}}{N_{series}} \tag{5}$$

When series arrangements are evaluated, the variation of the value of the linear and square coefficients of thermal losses should be taken into account. The determination of the K parameter is required, which is an intermediate coefficient used to calculate the new values of η_o, a_1 and a_2 (Equation (6)) [17]:

$$K_i = \frac{A_c \cdot a_i}{m \cdot c_e} \tag{6}$$

Once this parameter has been calculated, new characteristic factors for N collector number disposed in series arrangement could be achieved (Equations (7) and (8)) [18]:

$$\eta_{oN} = \eta_o \left(\frac{1 - (1 - K_i)^N}{N \cdot K} \right) \tag{7}$$

$$a_{iN} = a_1 \left(\frac{1 - (1 - K_i)^N}{N \cdot K} \right) \tag{8}$$

An estimation of the needed absorption area of a solar field could be obtained by Equation (9) as the ratio of average value of the heat duty obtained from the primary circuit over the heat duty proportioned by 1 m^2 of absorber area:

$$A_{sf} = \frac{\sum_i Qi_{\text{c.prim.}}}{\sum_i Qi_{m^2c}} = \frac{\overline{Qi_{\text{c.prim.}}}}{\overline{Qi_{m^2c}}} \tag{9}$$

3.4.2. Arrangement in parallel

For this arrangement the following assumptions were made:

- The solar field design consisted of a single row of collectors in parallel. The T_o of the primary fluid was the same as the output of each collector's set, and the mass flow was the sum of the individual capacities of each collector.
- From the lower stratification level of the accumulator tank, water was send to the exchanger at a T which fluctuated throughout the yea, but with an annual average value of 43.6° C.
- After heat exchanging with the primary fluid, water was sent to the upper level of the tank and, then, pumped to the boiler with a T_o of 95.0° C.
- The maximum possible working T was always required for the warming of the primary fluid leaving the solar field in its way to the heat exchanger (for the specific conditions, its annual average value was 119 °C). This T was kept as T_i of the primary fluid when it arrived to the exchanger.

Table 6
Variation of the average absorber area and the solar fraction as the outlet temperature of the secondary fluid changes.

Temperature effluent of secondary fluid at heat exchanger (°C)	Annual average absorber area of the solar field (m^2)	Annual average solar fraction
95	3.330	1.00
90	3.012	0.90
80	2.375	0.70
70	1.738	0.50
60	1.101	0.30
50	465	0.10

- The T_o of the primary fluid from the exchanger was fixed at 66.0 °C (calculated by *NTU*, Number of Unities of Transference method). Also, this was the T_i of water-glycol mixture in the solar collectors. So, T_c was 92.3 °C.
- The mass flow of the secondary fluid varied along the year; its annual average value was 1.047 kg/s (a circumstance imposed by the requirement of hot water in the pasteurizer).

Initially, it was demanded to the solar subsystem to provide all the energy (in form of hot water) needed by the process; it means that solar fraction's value was 1.00 (Scenario 3). The minimum fluid flow ratio (primary/secondary) required was 1.108, corresponding to primary flow of 1.160 kg/s. To obtain a solar fraction of 1, it was necessary to have an annual average solar field (net absorber area) of 3330 m^2 obtaining a solar contribution of 2730 kWh/day. In August, it would be possible to obtain a solar supply of 2626 kWh/day, by an absorber solar field of 1149 m^2, but, in January, an absorber area of 9442 m^2 was required to generate 2821 kWh/day for the same solar fraction. The ratio between the maximum and minimum absorption surfaces for $f = 1.00$ was 10.2. To supply the process only by solar thermal energy, it would be necessary to install the maximum calculated surface. On the other hand, this installation would be oversized during 11 months of a year.

So, it was appropriated to consider the reduction of the solar fraction by increasing the energy output of the boiler.

If the mass flow ratio and the effluent temperature of the primary fluid from the exchanger were kept at its initial value (1.108 kg/s and 66.0° C, respectively), the reduction of the T_o of the secondary fluid would oblige the conventional heating utility to cover the energy deficit. Thus, the decrease in temperature would induce a decrease of the solar fraction. Table 6 shows the evolution of the net absorber area of the solar field and the solar fraction for different values of outlet temperature of the secondary fluid, going from the exchanger in its way to the solar accumulator.

Usually, solar fields which work at low and middle temperature can not adjust their surface to the other operation variables of the

Table 7
Duties and solar field surfaces for different arrangements of collectors.

Parameter	Unit	Arrangement			
		1 p	2 s/p	3 s/p	4 s/p
Outlet temperature of primary fluid	°C	119	130	130	130
Mass flow of primary fluid	kg/s	1.160	0.478	0.32	0.24
Heat duty from primary circuit	kWh/day	2876	1369	958	719
Solar contribution	kWh/day	2733	1366	910	683
Boiler contribution	kWh/day	0	1366	1823	1842
Solar fraction		1.00	0.50	0.33	0.27
Average equivalent A_{sf} to parallel		3330	1665	1109	833
Average A_{sf} for f = 1.00		3330	6660	9991	12,334

Table 8
Data base for estimation of absorber and total area of the solar field.

Variables of the energy system of the plant	Value	Unit
T_{o-prim} at heat exchanger	66	°C
$T_{o\text{-}sec}$ at heat exchanger	95	°C
Maximum $T_{i\text{-}prim}$ at heat exchanger	130	°C
Annual average $T_{o\text{-}prim}$ from collection area	119	°C
$T_{i\text{-}sec}$ at heat exchanger	43.6	°C
Ratio of flows primary/secondary	1.108	–
Efficiency of the heat exchanger	0.95	–
Volume of water in the accumulator	45,701	kg
Operation time of the boiler	5	h
Annual average time of operation of the solar installation	12.5	kg
Energy demand to the boiler without solar contribution	2600	kWh/day
Annual average of total irradiance at 43°, south orientation	4.088	kWh/m^2·day

Table 9
Variation of the needed solar field areas (absorber and total) for different solar fractions.

Average annual solar fraction	Absorber area of the solar field (m^2)	Number of collectors	Total area of the solar field (m^2)
1.00	1939	646	2739
0.90	1745	582	2465
0.80	1551	517	2191
0.70	1357	453	1917
0.60	1164	388	1643
0.50	970	323	1369
0.40	776	259	1096
0.30	582	194	822
0.20	388	129	548
0.10	194	64.6	274
0.00	0	0	0

systems. So, a defined net absorber area is installed; this surface can contribute more or less energy in relation to the other variable of the process, and it is possible to model the production of energy as solar fraction, and to monitor their development over the year, according to the installed working absorber surface. Under the most exigent conditions with the thermal installations (T_o of primary and secondary fluid heat exchanger, 95.0 °C and 66.0 °C, respectively; primary/secondary mass flow relationship, 1.108), and for absorber areas lower than 1000 m^2, the system could not provide enough energy to reach the solar fraction of 1.00 during any period of the year. Above this surface, and obviously with more relevance as the area of the solar field increases, there would be some months in which this value would be reached and even exceeded (the solar field was able to provide more heat than required by the process at that time). In any event, December and January were notoriously unproductive, and that would require huge areas of solar collectors to supply to the process a significant amount of energy. Moreover, July, August and September were the most profitable and could bring significant amounts of energy to the process in terms of the installed surface.

3.4.3. Combination arrangements in series-parallel

As mentioned before, arrangements in series provided a higher temperature of the primary fluid but, for the same surface as in parallel, the obtained mass flow was lower, and the efficiency decreased. To evaluate the energy supply of other arrangements (2 lines, 3 lines and 4 lines in series-parallel), the flows which circulate through the series columns were recalculated according to the requirement of each arrangement. The modeling results based on these different structures are presented in Table 7.

Normally, series-parallel systems are designed with the objective of obtaining higher temperatures than only parallel arrangements. In the case of the dairy process, this advantage was not relevant because, as the temperature increased, there was a dramatic reduction of the primary fluid and, at the same time, a very significant decrease of the solar fraction. From this perspective, for the same energy efficiency, it would be more economical interesting (economically too) the option of arrangement exclusively in parallel.

3.4.4. Estimation of the net and total absorber area of the solar field

For an arrangement in parallel of all collectors of the field, it is possible to establish the absorption and total surfaces for a fixed technology and operation variables of the system. For this, it is necessary to determine the energy contribution of the primary circuit (annual average value), and the amount of energy which can provide with an absorption surface of 1.0 m^2. The relationship between these variables provides the surface required to deliver this amount of energy. From this value, the size of the field and the

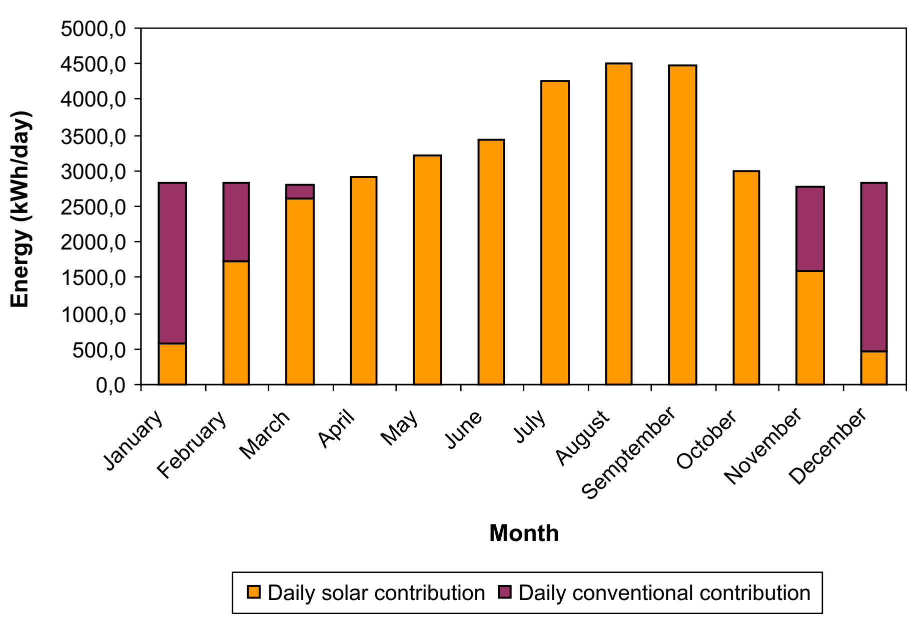

Fig. 7. Daily participation of solar and conventional duties.

Table 10
Effect of solar fraction variation (and respective solar field absorber area) on the temperature of primary and secondary fluids.

Average annual solar fraction	Absorber area of the solar field (m^2)	T_o-prim from solar field (°C)	T_o-sec from heat exchanger (°C)
1.00	1939	119	95.0
0.90	1668	114	90.0
0.71	1199	103	80.0
0.51	805	93.0	70.0
0.32	466	82.8	60.0
0.12	171	72.6	50.0
0.04	52.2	68.1	45.6
0.00	0	66.0	43.6

number of solar collectors needed are determined. Table 8 exposes the data used for the calculation.

The result of the system modeling under the above variables is presented:

- Absorber area of the solar field, A_{sf}: 1939.2 m^2.
- Average annual solar fraction achievable by the previous surface, f: 1.00.
- Number of required collectors, N_c: 646.4 collectors.
- Total area of the solar field, A_{tsf}: 2738.8 m^2.

Fig. 7 presents the heat duty month to month, that must deliver the solar subsystem and the conventional heater. As it is easily visualized, there were 7 months were the contribution of the solar surface exceed the needs in heating. The excess of heat production could be used to drive an absorption heat pump for cold production. For the same operation conditions, the required absorber surfaces for different solar fractions are reported in Table 9. In order to understand the potentiality of the new energy system, it was important to evaluate its performance when the effluent temperature of the primary fluid from the collection area decreased. At this point, the reduction of the solar fraction (and, linked to this, the diminution of the solar collector area) was considered. Logically, any diminution of the temperature of the primary fluid lead a respective reduction of the temperature of the secondary fluid when it left the heat exchanger and arrived to the accumulator (so, increasing dependence on natural gas), (see Table 10).

Finally, in order to show the viability of replacing conventional fossil fuels energy by solar thermal energy in the dairy plant, it could be mentioned that 71% of the natural gas consumption could be replaced by a total solar field area of 1694 m^2, which could save half the consumption of natural gas ($f = 0.51$), with a total area of 1136 m^2. Regarding the economy of fossil fuel consumption, the saving would depend on the installed surface, other accessory elements of the solar thermal installation, investment and an appropriate use of the resource. As a result of this analysis, it could be estimated that, a solar fractions of 0.30, 0.50 and 0.70 could reduce the natural gas consumption of the base case (1584 MWh/year) and the optimized process (1361 MWh/year) to 953 MWh/year, 689 MWh/year, and 408 MWh/year, respectively.

4. Conclusions

From this analysis, several conclusions could be depicted about the technical feasibility of integrating solar energy in the dairy process:

- The global solar radiation would be enough to meet the process energy requirements, to a greater or lesser extent, depending on the installed collector area; would be possible to achieve important average annual solar fractions by middle size solar fields.
- For more than half a year, high solar fractions (above 0.50) could be obtained with relatively small solar fields (less than 1000 m^2 of absorber area). The remaining part of the year, the radiation received would be sufficient to replace, with significant solar fractions, the energy demand of the process.
- In terms of energy contribution, during winter months, especially December and January, solar productivity was found to be very poor due to the low number of sun light hours and low received solar radiation.
- From April to October, a substantial excess of energy could be used to drive a cooling systems reducing energy consumption, and avoiding the loss of the resource.
- Given the nature of solar radiation in the area (more diffuse component that directly over the year), the most appropriate technology would be vacuum tube–heat pipe type, because the intensive use that these collectors make of diffuse radiation, and its low coefficients of linear and quadratic losses.
- In view of the fact that the collectors can deliver a primary fluid at a higher temperature (an annual average of 118.6 °C) than the pasteurization demands (95.0 °C), it would not be necessary any kind of series-parallel array of the solar field. The most appropriate design proves to be a parallel distribution of collectors, tilted toward the south an angle equal to the latitude of the place (43°).

Any decision on the appropriate sizing of the solar field should be based on economic and environmental aspects, and some other inherent aspects that may be critical for decision-making (as the maximum area available for the solar installation). The results of this analysis have demonstrated the technical feasibility of replacing thermal energy to the dairy process plant under the specific climatology where it is located, by relatively reasonable size solar fields. The analysis also showed the actual possibility of reducing the current consumption of fossil fuels and the associated CO_2 emissions.

Nomenclature

A_c	absorber area of a collector
A_{sf}	net absorber area of a solar field
A_{tsf}	total area of a solar field
a_1	liner heat loss factor
a_2	square heat loss factor
c_p	specific heat capacity
Cp	mass heat capacity
f	solar fraction
G	total or global irradiance
H	daily total or global radiation
I	hourly total or global radiation
m	mass; mass flow rate
$\dot{Q}$	heat load
Q	heat flow
T_a	ambient temperature
T_c	average collector internal fluid temperature.
T_o	temperature of outlet fluid
T_i	temperature of inlet fluid
ΔT	temperature change
$\Delta T_{\min}$	minimum variation on temperature assumed between curves at pinch analysis
t	time
Δt	time interval; time length of an operation
η	solar collector instantaneous efficiency
η_o	solar collector optical efficiency

References

[1] Eia-U.S. Department of Energy. International energy outlook 2009, Washington, http://www.eia.doe.gov/oiaf/ieo/pdf/0484(2009).pdf; 2009 (Consulted on October, 8th, 2010).
[2] EEA-European Environmental Agency. Energy and environmental report 2008, Report no 6, Copenhagen, http://www.eea.europa.eu/publications/eea_report_2008_6; 2008 (Consulted on October, 8th, 2010).
[3] Gustafsson S. Optimization of drying kiln operation. Int J Energy Res 2000;24:19–26.
[4] Higman C, van der Burgt M. Gasification. 2nd ed. Elsevier; 2008.
[5] Bazen E, Brown M. Feasibility of solar technology (photovoltaic) adoption: a case study on Tennessee's poultry industry. Renew Energ 2009;34:748–54.
[6] Philibert C. IEA-International Energy Agency, Barriers technology diffusion: the case of solar thermal technologies, www.iea.org/work/2006/cop12/solar_thermal.pdf (Consulted on October, 8th, 2010).
[7] Schnitzer H, Brunner C, Gwehenberger G. Minimizing green house emissions through the application of solar thermal energy in industrial processes. J Cleaner Prod 2007;15:1271–86.
[8] Atkins M, Walmsley M, Morrison A. Integration of solar thermal for improved energy efficiency in low-temperature-pinch industrial processes. Energy 2010;35:1867–73.
[9] Schweiger H, Mendes JF, Benz N, Hennecke K, Prieto G, Gusi M, et al. The potential of solar heat in industrial processes. A state of the art review for Spain and Portugal. Denmark: EuroSun 2000, http://www.les.ufpb.br/portal/index2.php?option=com_docman&task=doc_view&gid=157&Itemid=30; 2000 (Consulted on October, 8th, 2010).
[10] ESTIF-European Solar Thermal Industry Federation. Solar industrial process heat. State of the art, Roma, http://www.iea-shc.org/publications/downloads/shc_annual_report_2006.pdf; 2006 (Consulted on October, 8th, 2010).
[11] Vannoni C, Battisti R, Drigo SEIA-International Energy Agency. Potential for solar heat in industrial processes, http://www.iea-shc.org/publications/downloads/task33-Potential_for_Solar_Heat_in_Industrial_Processes.pdf; 2008 (Consulted on October, 8th, 2010).
[12] Weiss W, Rommel MIEA-International Energy Agency. Process heat collectors. State of the art within task 33/IV: solar heat for industrial processes, Madrid, www.ciemat.es/recursos/doc/Centro_Documentacion/323497443_6820091193.pdf; 2008 (Consulted on October, 8th, 2010).
[13] EVE-Ente Vasco de la Energía. Atlas de radiación solar del País Vasco, http://www.eve.es/web/Documentacion/publicaciones/DetallePublicacion.aspx?NodeID=2867&;Categoria=; 2002 (Consulted on October, 8th, 2010).
[14] Censolar. Distribución horaria de la irradiación solar global incidente sobre superficie horizontal en las cinco zonas climáticas definidas en el Código Técnico de la Edificación de España, http://www.asif.org/files/Distribucion_horaria_irradiancia_solar_vzonas_CTEEsp.pdf; 2007 (Consulted on October, 8th, 2010).
[15] Mahjouri F. Vacuum tube liquid-vapor (heat pipe) collectors, Thermo Technologies, thermomax.com/Downloads/Vacuum%20Tube%20Paper.pdf. Maryland. (Consulted on October, 8th, 2010).
[16] Li X, Wang Z, Wang B. Fundamental study of the thermal performance of a solar collector with evacuated tube for solar heating system. Denmark: EuroSun 2008, http://www.les.ufpb.br/portal/index2.php?option=com_docman&task=doc_view&gid=157&Itemid=30; 2008 (Consulted on October, 8th, 2010).
[17] Duffie J, Beckman W. Solar engineering for thermal processes. 3rd ed. Wiley; 2006.
[18] Oonk R, Cole-Appel B. Calculation of performance of N collectors in series from test data on a single collector. Sol Energy 1979;23:535.