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Multiparameter Analysis in Total Cost Evaluation of a Low Temperature Solar Thermal Assisted Heat Pump for Industrial Applications

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The low-temperature solar thermal facility presents important challenges that must be solved to guarantee the heat load and target temperature by means of a parametric analysis that defines the design, operation, and performance of a solar-assisted heat pump. Through the analysis, objectives such as minimum total cost, zero emissions, and a competitive kWh cost can be reached. In this work, a multiparameter study was carried out for a system made up of low temperature solar thermal source and a heat pump to define the operation and performance, using a case study of 2nd generation (2G) bioethanol production. Together, these devices define the system design and must guarantee the operation of the industrial process. One of the system designs seeks to achieve the lowest cost of energy (LCOE = 0.101 USD/kWh) and deliver a heat flow of 637 kW with a temperature of 105 °C for 16 h; the heat pump must work with a heat source of 40 °C and a coefficient of performance (COP) of 3.5 with the refrigerant R1234ze(z). Another of the system designs seeks to achieve the best performance of the heat pump, which was obtained with a COP= 5.1. Its operating conditions are reached when the heat source is 60 °C, and the compression work is 125 kW using the refrigerant R1336mzz(E). The system levelized cost of energy is 0.124 USD/kWh. The importance of this work lies in the fact that a parameter evaluation allows to know the relevant objectives of the system, previous the design to make a custom suit.

1. Introduction

Worldwide is being carried out the transition toward the production and use of solar process heat for industrial applications. In general, the solar process heat deals with maximizing the use of solar thermal energy and minimizing greenhouse gas generation, guaranteeing the heat load supply at the target temperature. In solar thermal integration, there are some challenges related to the cost, environment, operability, and continuous delivery of solar thermal energy to widespread its use in the industrial sector. Some of these challenges have been solved, for example, variability of solar resources, target temperature, costs, supply time, and others pending to be resolved, like big areas for solar installation.

The total cost of any solar thermal installation is always important, but it is not the only variable that defines the design. It also impacts the construction, operation, and sustainability of a solar thermal installation. Fuentes-Silva et al. (2020) established a way to deliver the heat load using a solar thermal storage system to have solar thermal energy available when the process requires it. The thermal storage system is used to guarantee the supply of energy in continuous and batch processes, but also to increase the period of solar thermal energy supplied, looking at the end for solar fraction equal to one. The network of solar collectors is designed with the minimum irradiance that occurs during the winter period to guarantee supply during all year (Martínez-Rodríguez and Fuentes-Silva, 2021). However, there are several consecutive days during the year with irradiance levels of those presented in the winter period. Silviano-Mendoza et al. (2021), reported for the state of Guanajuato (Mexico) that there are 20 consecutive days below 700 W/m^2, to solve the challenge posed by low irradiance levels, Martínez-Rodríguez et al. (2023) developed a study to determine the total minimum cost of a heat pump

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coupled with solar thermal energy, with which it is possible to guarantee the supply of the thermal load, at the target temperature, throughout the year, considering those low levels of irradiance.
Currently, heat pumps have expanded their range of use. Zhou et al. (2019) performed a theoretical evaluation of a dual system that uses solar energy as a heat source through a system of photovoltaic thermal units RB-PVT, a roll-bond heat exchanger, and a heat pump system that also provides power to the compressor. Razi and Dincer (2020) carried out the thermodynamic analysis of a multi-system powered by solar thermal energy to produce electric power, space-cooling, space-heating, hot water, and hydrogen, employing a heat pump with a thermal storage system. The efficiency of the multisystem is 52 %. Díaz-de-León et al. (2022) designed a low-temperature solar system to assist a heat pump with the objective of reducing the storage volume.
Continuous or batch industrial processes require the energy supply to be guaranteed at the target temperature during the days of operation of the process in a year. The limitations of a solar thermal installation due to low levels of irradiance during the year are resolved with the use of the assisted heat pump with low-temperature solar thermal energy. The characterization of the proposed system, low temperature solar thermal energy-assisted heat pump, allows the designing of new systems, evaluating existing systems, and determining the ad-hoc system for each need of an industrial process.
In this work, the parameters that define a heat pump assisted with low-temperature solar thermal energy are identified. The parameters that define the studied device can be used for the design, construction, and operation of a new device or to evaluate the operation of an existing device thermally. The parameters are determined based on the design objective of the device, which may be the total cost, the greenhouse gases (GHG) that are not emitted, and the operating time of the device.

2. Design parameters of the thermal system

To develop this study, the authors will work with the proposed system constituted by a heat pump assisted by a low-temperature solar thermal device as described in Martínez-Rodríguez et al. (2023). Hereafter this system will be called a solar thermal assisted heat pump (STAHP). It is relevant to determine, calculate and apply the parameters of a low-temperature heat source and a heat pump that constitute the main equipment of the thermal system of an industrial process. The heat source assures the heat flow required in the heat pump evaporator, and the heat pump, in turn, guarantees the heat load at the temperature level demanded by the industrial process 24 h/day, 365 days/y. The process can be continuous or batch.
Current technology and materials allow breaking the paradigms of the applications of low-temperature heat sources when coupled with a heat pump, making the design of the proposed system flexible and wide-spreading the scope of applications. This proposal allows us to evaluate different designs and select the best ones through a flexible, efficient, and viable system, as a potentially sustainable solution for the heat demand of an industrial process, completely substituting conventional technology and fossil fuels. In this paper, the following parameters are studied for the heat source: solar thermal heat flow, target temperature, and supply time; for the heat pump: the heat load to be supplied and temperature to be reached, to determine how they impact the final design of the STAHP.

2.1 Low-temperature solar thermal device: the heat source

The design of the solar thermal device depends on the solar resource, the ambient temperature, wind speed, and the operating conditions of the process, parameters such as the heat load, the target temperature, and the supply time. The availability of the solar resource and the demand for heat must match to carry out the integration of solar thermal energy into the process. The design of the solar thermal device is based on the model proposed by Martínez-Rodríguez et al. (2022), where the main objective is to increase the use of low-temperature solar thermal energy, henceforth this device will be called solar thermal energy. Table 1 shows the data of the environmental conditions used in the design of the solar thermal device, the conditions of a winter period of an average day were considered.

Table 1: Environmental conditions used to design the low-temperature solar thermal device

Season of year	Irradiance (W/m^2)	Ambient temperature (°C)	Wind velocity (m/s)
Winter	487.90	19.80	2.25

2.2 Heat pump

The heat load supplied by the heat pump is defined by the heat duty and target temperature of the process. By means of these two process operation parameters the refrigerant and the evaporator temperature are selected. Eq(1) shows the condenser heat capacity, $\dot{Q}_{cond}$.

$$\dot{Q}_{cond} = \dot{m}(h_4 - h_3) \quad (1)$$

Where $\dot{m}$, is the mass flow rate of the working fluid, kg/s; h_4, is the enthalpy of subcooled refrigerant outside the condenser, kJ/kg; h_3, is the enthalpy of overheated refrigerant at the entrance of the condenser, kJ/kg.
The heat absorbed by the evaporator, $\dot{Q}_{eva}$, is assumed like a variable because the objective to analyze the effects of different evaporation temperatures (related with solar variability) to simulate the working typical conditions, and to study the system performance. $\dot{Q}_{eva}$ is calculated multiplying the mass flow rate of the working fluid, kg/s; with the enthalpy difference. See Eq(2).

$$\dot{Q}_{eva} = \dot{m}(h_2 - h_1) \quad (2)$$

Where h_1, is the refrigerant enthalpy at the exit of the expansion stage, kJ/kg; and h_2, is the refrigerant enthalpy at the entrance of the compressor, kJ/kg.
To determine the evaporator temperature was carried out as described by Martínez-Rodríguez et al. (2023). The condenser temperature is adjusted to 110 ºC to consider the critical temperature limitation of the working fluids R1234ze(Z) (150.12 °C and 2,900 kPa) and R1336mzz(E) (137.7 °C, 3,149 kPa). Subcooling was made at 5 K, and the degree of overheating was from 10 to 20 K. Thermodynamic properties of refrigerant R1234ze(Z) were published by Huong et al. (2022), and for refrigerant R1336mzz(E) were published by the manufacturer Opteon™ (2023). It is also supposed to be an isenthalpic process in the expansion valve, the heat transfer around it is despicable, and the same is about pressure drop.
The power consumption by the compressor, $\dot{W}_{comp}$, it is expressed in Eq(3), there is the relationship between the product of the mass flow rate of the working fluid, $\dot{m}$, by the isentropic enthalpy difference in the compressor, Δh_{ise}, dividided by the product of the isentropic and electromechanical efficiencies, η_{ise} y η_{em}.

$$\dot{W}_{comp} = \frac{\dot{m}\Delta h_{ise}}{\eta_{ise}\eta_{em}} \quad (3)$$

Isentropic and electromechanical efficiencies are assumed with values 0.85 and 0.95 (Soares, 2005; Campbell, 2014). Finally, performance coefficient, COP, is calculated by means the heat capacity and the electrical energy consumed power consumption by the compressor, $\dot{W}_{comp}$, using Eq(4).

$$COP = \frac{\dot{Q}_{cond}}{\dot{W}_{comp}} \quad (4)$$

Figure 1 shows the conceptual proposal of this article. Showing the main components and defining the device by means of parameters. The heat source can be adjusted to the energy requirements of the heat pump depending on the objective to be achieved when designing or studying a STAHP to supply the heat load to an industrial process. Some of the parameters that define the proposed system are also shown.

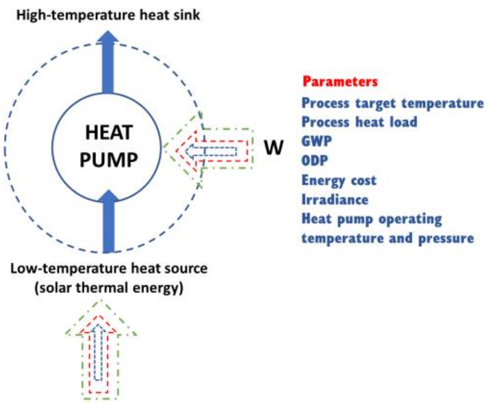

Figure 1: Conceptual diagram of proposed devices

2.3 Case study: 2G anhydrous bioethanol production by solar process heat

A viable option to produce biofuel anhydrous bioethanol can be from residues of agave like bagasse and pencas (Oseguera-Villaseñor, 2016). Bioethanol, with a purity of 99.5 %, is mixed with gasoline to be useful. Agave residues are ground to undergo acid hydrolysis; a fraction of this one enters to enzymatic hydrolysis to end with its fermentation. The product of fermentation is a dilute solution with less than 10 % ethanol. Finally, this mix ethanol-water is purified to obtain ethanol with the required purity. Figure 2 shows a simplified scheme of the main operation stages to produce 2G anhydrous bioethanol.

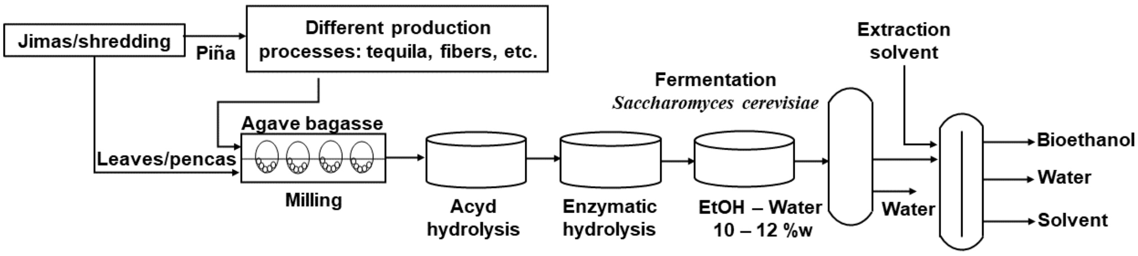

Figure 2: Simplified process to obtain 2G anhydrous bioethanol from agave residues

The plant operates for 350 days/y in a batch process for 16 h/day. To obtain 206 kg/day of bioethanol with a specified purity, there are processed 7,206 kg/day of agave residues. A fire tube boiler produces a heat load of 4,250 kW with a temperature of 206 °C to feed the process. The purification section requires the largest heat load, to reduce the energy consumption Pinch Analysis is applied. With a ΔT_{min} of 5 °C the heat load is 637 kW for a temperature of 105 °C (Martínez-Rodríguez et al., 2022).

3. Results

Some parameters considered in this study are heat load, the target temperature of the process, and irradiance. Eight designs of solar-assisted heat pumps were calculated to supply the heat load to deliver the required energy by a 2G anhydrous bioethanol process at the target temperature. The heat load is 637 kWh, target temperature of 105 °C. Figure 3 shows the relationship between COP and the required electric power by the compressor while the evaporator temperature varies with different refrigerants from 40 to 60 °C. As can be seen in Figure 3a, from 55 °C the behavior of both refrigerants is exponentially increasing, however, the COP of refrigerant R1234ze(Z) is better at low operating temperatures in the evaporator, because the heat removed in the condenser is greater. Figure 3b, shows the relationship of the compression work with the temperature in the evaporator for refrigerants R1234ze(Z) and R1336mzz(E), where decreasing exponential trend regarding temperature is shown, however, for low temperatures in the evaporator it is required higher compression work due to isotherms in the refrigerant R1336mzz(E) require more energy to go from one temperature level to another. The results obtained in Figures 3a and 3b show that refrigerant R1234ze(Z) has thermodynamic properties that allow it to have a higher COP and a lower compression work.

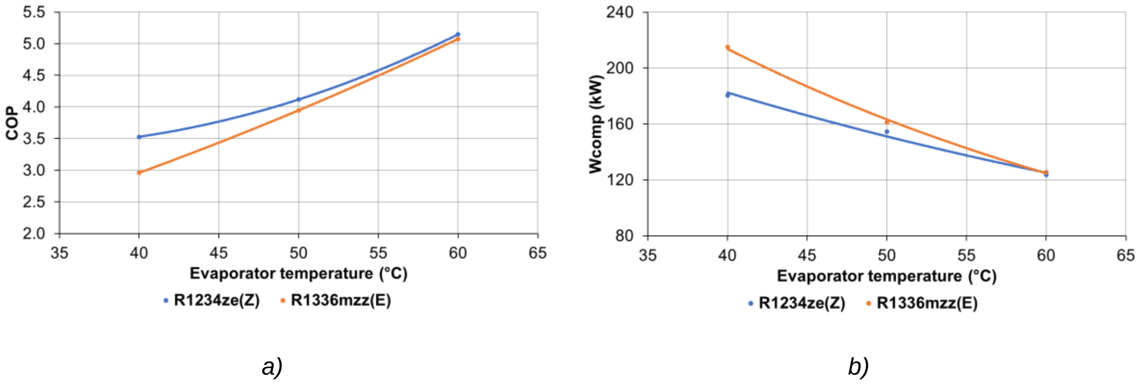

Figure 3: Relationship of COP and the required electric power by the compressor ($\dot{W}_{comp}$) regarding different temperatures in evaporator

Two working fluids used, R1234ze(Z) with ODP= 0 and GWP= 1.4, and R1336mzz(E) with ODP= 0 and GWP= 7. The heat pump condenser must deliver a heat load of 637 kW, the required by the process, at temperature of 105 °C. In Table 2 are the results from the evaluation of the different STAHP designs considered for the study, in which the flow, temperature and refrigerant were varied. For the same flow of R1234ze(Z) refrigerant, the other refrigerant (R1336mzz(E)) cannot supply the heat load at the target temperature of the 2G bioethanol

process. Para ambos refrigerants, R1234ze(Z) y R1336mzz(E), as its mass flow increases, the cost of energy decreases, while the COP of the heat pump increases. While as the temperature in the evaporator increases, the cost of energy increases, and the COP of the heat pump also increases.
The COP is the design parameter that measures the performance of the heat pump. For proper efficiency and functionality, a heat pump must achieve a COP between 2 and 6 (a value less than 2 does not pass industry inspection) (Espiñeira, 2022). In the results obtained for all the proposed designs, the COP of the heat pump is located between 3.0 and 5.1, so all the designs analyzed are considered technically feasible. The design with the highest energy cost, 0.146 USD/kWh (last column), is presented for an operating temperature in the evaporator of 60 °C, where the cost of energy is determined by the cost of the solar heat source, the low-temperature heat source must provide a temperature of 70 °C (source). Compared to the design with the lowest energy cost, 0.101 USD/kWh (second column), which is reached for an evaporator temperature of 40 °C using R1234ze(Z). In both scenarios, emissions to the environment from fossil fuels are zero.

Table 2: Description of several designs for the low-temperature solar thermal assisted heat pump attending the studied parameters

Parameter	Refrigerant R1234ze(Z)				Refrigerant R1336mzz(E)			
Heat load (kWh)	10,192							
Target temperature (°C)	105							
Working fluid	Water							
Evaporator temperature (°C)	40	60	40	60	40	60	40	60
ΔT_{lift} (°C)	70	50	70	50	70	50	70	50
Mass flow rate (kg/s)	4.2	4.2	4.0	4.0	7.2	7.2	6.6	6.6
Absorption area of solar collector network (10^3 m^2)	3.49	5.35	3.96	6.15	4.15	6.54	5.95	7.74
Thermal storage volume (10^2 m^3)	1.57	1.19	1.42	1.13	1.25	1.07	1.08	1.00
Storage temperature (°C)	62	81	66	84	70	88	78	90
Operation time (h)	5.75	4.5	4.75	3.75	4.25	3.5	2.75	2.75
COP	3.5	5.1	3.4	4.9	3.0	5.1	3.0	4.3
LCOE (10^{-1} USD/kWh)	1.01	1.09	1.08	1.21	1.17	1.24	1.40	1.46
GHG emissions (t/y)	0	0	0	0	0	0	0	0

Electricity prices for business: 0.164 USD/kWh (world average), 0.138 USD/kWh (EEUU) in 2022, 0.068 USD/kWh solar PV (IRENA, 2022). Natural gas prices: 0.126 USD/kWh (world average) in 2022.

4. Conclusions

In all proposed device designs (SATHP), the full heat load was supplied at the target temperature for a 2G bioethanol process. The irradiance level with which the solar thermal energy source was designed guarantees the supply of energy to the process throughout the year. Emissions into the atmosphere are zero, and the environmental impact from the use of refrigerants is minimized with an ODP of zero for both refrigerants and a GWP≤7 for one of them. The design with the lowest energy cost (second column, Table 2), LCOE= 0.101 USD/kWh, was obtained with the following design parameters: heat load of 637 kWh, temperature of 105 °C, COP= 3.5, irradiance of 487.9 W/m^2.
In six of the evaluated designs, the energy cost of the proposed system is competitive (0.101 – 0.124 USD per kWh) compared to the cost obtained from the way the process currently operates (with natural gas) of 0.125 USD/kWh.

Nomenclature

COP – Coefficient of performance
GHG – Greenhouse gases
GWP – Global warming potential
LCOE – Levelized cost of energy (USD/kWh)
ODP – Ozone depletion potential
ΔT_{lift} (°C) - temperature difference between source and sink
R1234ze(Z) – Refrigerant R1234ze (Z)
R1336mzz (E) – Refrigerant R1336mzz (E)

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