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Modeling of a solar collector of parabolic dish and a plate heat exchanger to improve the industrial drying process of calcium propionate

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Abstract : Spray drying is a method of producing a dry powder from liquids solutions and is widely used in the food industry. However, fossil fuels is used to generate the hot air, which produce ambiental pollution. In this work will be used renewable and clean energy to design a solar energy concentrator, and its overall thermal energy generation at different operating conditions was evaluated. The system consist of three parabolic dish solar collectors coupled to a plate heat exchanger. In this design, the mathematical models of the different equipment involved in the system (solar collectors, heat exchangers, pump, pipes and fittings, etc) were established.
Keywords: Solar radiation, thermal energy, solar parabolic dish collector.

1. Introduction

Drying is a process widely used in the food industry to preserve and maintain the quality of food [1]. In many process it is necessary to obtain solids from an aqueous solution, or any other solution with a different solvent, in order to obtain a granular material [2, 3], purified and dried product that can be packaged and sold [4]. Spray drying is a method of producing a dry powder from liquids solutions and is widely used in the food industry. Spray dryers need a continuous supply of hot air [5], which generally requires the burning of fossil fuels such as natural gas.

A compound commonly used in the food industry is calcium propionate, which is a food additive used as a preservative, with the purpose of keeping food in a suitable and attractive condition while is delivered to market or while being stored at home. Food that commonly contain calcium propionate are bread, cookies, pies, cakes and other flour products in general [6]. Due to its production process, the calcium propionate is produced in slurry form and must be separated from the water. Currently, the company C. I. Real S.A. it is dedicated to developing, manufacturing and marketing food additives, including calcium propionate; and within its facilities it has 4 spray dryers which are intended for obtaining calcium propionate powder. The company uses natural gas to generate 1 000 000 BTU / h, necessary to heat air from 30 °C to 285 °C and which will subsequently be used in each of the dryers to evaporate 1.200 L/h of water contained in the calcium propionate slurry.

The use of natural gas as fuel generates carbon dioxide emissions and significantly increases the production costs of calcium propionate, which limits the participation of the company in new markets that are highly competitive. For this reason, there is a need to find an alternative energy source [7-22] capable to replace the use of natural gas as a heat source, or that at least contributes to use a less amount of that fuel, because it is crucial to decrease production costs to allow the company expand its market boundaries. The use of solar collectors is an alternative to the problem experienced by the company C. I. Real S. A., which consists of high consumption of natural gas in the drying of calcium propionate. Solar collectors are devices that capture solar energy and transmit it to a heat transfer fluid for a later use [23].With respect to the use of solar energy, Spain is the country with most thermal power plants in the world, while the United States is located in the second place. Most solar thermal plants built around the world (71%) use parabolic troughs to concentrate solar radiation while only 3.2% use parabolic dish collectors [24].It is important to note that in the region there are very few studies about on concentrating solar power systems, as the one performed by Almanza and Cabarcas [25].

This research proposes the use of a system of parabolic dish solar collectors to solve the problem of heating the air used in the drying process of calcium propionate. The operating mechanism of each parabolic dish collector can be described as follows: As collector follows the sun, the light from the sun shines on the collector, which has a highly reflective surface and reflects light onto the receiver in a concentrated way. A pump circulates the heat transfer fluid through the receiver and the concentrated solar energy heats the fluid as it passes through it. The hot thermal fluid is then pumped to a plate heat exchanger that transfers the thermal energy produced to the air. In this way, the company can reduce the costs associated with natural gas consumption and the adverse environmental effects caused by the carbon dioxide emissions.

2. Conceptual design for the concentrating system of solar power

The figure 1shows a diagram of the conceptual design for the concentrating system of solar power. It is proposed an initial preheating cycle for the fluid Dowtherm Q, so that it quickly reached the desired operating temperature. For this, it was decided that the collectors and the pump should operate while valves 1, 2 and 5 remain open and the other closed. After reaching the desired temperature, the valve 5 was closed, the valves 3, 4 and 6 are opened, and the blower is turned on. In addition, the arrangement of equipment and distances between them are shown in figure 2. It is important to mention that the recommended distance between the solar collectors is equivalent to twice the aperture diameter, or the diameter of the parabolic dish collector.

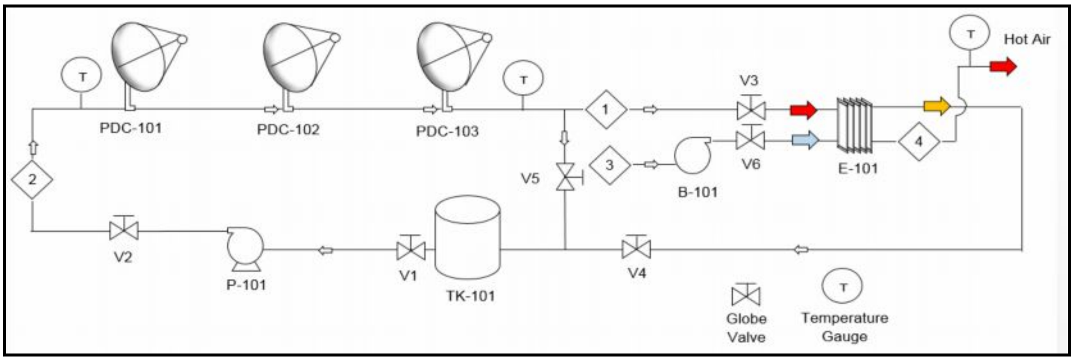

Figure 1. Process diagram of the CSP system proposed

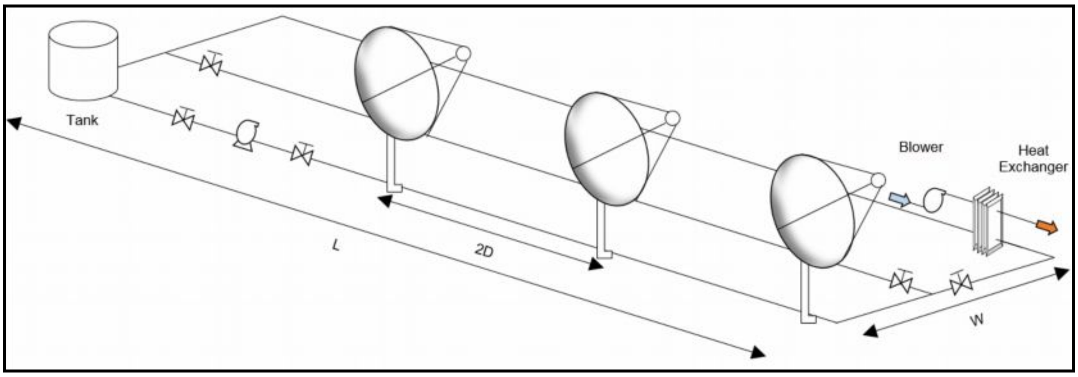

Figure 2. Isometric representation of the CSP system proposed

3. Determination of solar radiation levels in Cartagena

The value of solar irradiation (kWh/m^2) was recorded for each of the months of the year, as well as an annual average. This information was obtained from the maps of global solar radiation that are found in the Solar Radiation Atlas of Colombia [26]. After that, the values of hours of solar brilliance (h) were recorded for each of the months of the year and an annual average, from available maps in the Solar Radiation Atlasof

Colombia. Subsequently, data of solar irradiance (Is, W/m^2) were obtained by dividing the values of irradiation

between the hours of solar brilliance.

4. Thermodynamic model of parabolic dish collector

The thermodynamic model of the parabolic dish collector consist of an energy balance which enables to determine the useful heat that can be obtained from solar radiation [W/m^2] (DNI, Direct Normal Irradiance) incident on the collector. In this model, the energy losses that occur in the collector due to the optical properties of the reflector as well as the heat transferred from the receiver to the surroundings are determined. These losses affect the amount of useful heat that can be obtained.The incident heat is proportional to the aperture area of the dish concentrator and the direct normal irradiance, which can varies with geographical position, the orientation of dish concentrator and meteorological conditions. In this analysis, it is assumed that the DNI is constant and always parallel to the optical axis of the dish concentrator. The incident energyis defined as the product of irradiance and the aperture area of parabolic dish, and corresponds to the maximum amount of energy available for the heat transfer process.

$$Q_S = I_S \times A_a \qquad (1)$$

The amount of radiation that falls onto the receiver is a function of the optical efficiency which is defined as the ratio of the energy falling on the receiver to the energy incident on the concentrator's aperture [27]. The optical efficiency of the dish collector is determined by the optical properties of the materials used in the dish concentrator and receiver, as well as the geometry of the collector. Therefore, the following expression can be used to determine the optical efficiency:

$$\eta_o = \lambda\rho\tau\alpha\gamma \qquad (2)$$

After calculating the optical efficiency, the energy reflected to the receiver was calculated by using the expression:

$$Q_R = Q_S\eta_o \qquad (3)$$

The total heat loss rate in the receiver of a parabolic dish collector usually includes the three modes of heat transfer: conductive heat loss from the receiver (Q_{LK}), convective heat loss at the opening of the receiver (Q_{LC}), and radiation heat loss at the opening of the receiver (Q_{LR}). Thus, the total heat loss rate is expressed by the Eq. 4. Because the outer surface of the receiver is usually covered with a film of insulating material, it has been considered that the conductive heat loss is negligible compared to convective and radiation heat loss (Eq. 5).

$$Q_L = Q_{LK} + Q_{LC} + Q_{LR} \qquad (4)$$

$$Q_{LK} = 0 \qquad (5)$$

Determining heat loss by convection on the receiver of the solar collector is one of the most difficult tasks, because a correlation must be estimated in order to find a convective heat transfer coefficient. Many authors have proposed correlations for estimating heat transfer coefficient based on conditions in which only natural convection occurs, despising the possible effects of forced convection when the wind intervenes. We considered this assumption reasonable based on research performed by A. Tafur and A. Pizza [28] in which it was noted that the average wind speed in Cartagena was between 1.4 and 1.8 m/s for the period between 2006 and 2010.In this research, the set of equations used to determine the convective heat transfer coefficient is the model proposed by Shuang-Ying Wu et al [27]:

$$Nu_L = 0{,}106 Gr_L^{\frac{1}{3}} \left(\frac{T_W}{T_a}\right)^{0.18} \left(\frac{4.256 A_C}{A_W}\right)^{S} f(\theta) \qquad (6)$$

$$Gr_L = \frac{g\beta(T_W - T_a)L^3}{\nu^2} \qquad (7)$$

$$s = 0{,}56 - 1.01\left(\frac{A_C}{A_W}\right)^{\frac{1}{2}} \qquad (8)$$

$$f(\theta) = 1{,}1677 - 1{,}0762 sen(\theta^{0,8324}) \qquad (9)$$

In the equations above, the characteristic length of the receiver is the diameter of the cavity, D. After calculating the Nusselt Number, we estimated the convective heat transfer coefficient and the convective heat loss:

$$h = \frac{Nu_L k}{L} \qquad (10)$$

$$Q_{LC} = hA_W(T_W - T_a) \qquad (11)$$

The following equations were used to estimate the radiation loss from the receiver:

$$Q_{LR} = A_C \varepsilon_{eff} \sigma(T_W^4 - T_a^4) \qquad (12)$$

$$\varepsilon_{eff} = \frac{1}{\left[1 + \left(\frac{1}{\varepsilon_C} - 1\right)\frac{A_C}{A_W}\right]} \qquad (13)$$

Under steady state conditions, the useful energy that a parabolic dish solar collector can produce is equal to the energy transmitted to the heat transfer fluid, which was calculated by the energy that fall on the receiver minus the total energy loss from the receiver:

$$Q_U = Q_R - Q_L \qquad (14)$$

By setting the temperature of the heat transfer fluid at the entrance and exit of the system, we established the overall temperature increase. If the amount of useful energy given by a single parabolic dish collector is not enough to achieve the desired temperature increase, the use of more units is necessary.To determine the temperature increase of the heat transfer fluid (Dowtherm Q) that pass through a single collector, we used the following expressions:

$$T_{OC} = T_{IC} + \frac{Q_U}{w_C (C_p)_C} \qquad (15)$$

$$\Delta T_{COL} = T_{OC} - T_{IC} \qquad (16)$$

The number of collectors needed was estimated by dividing the overall temperature increase between the temperature increase given by a single parabolic dish collector [25]:

$$N_{COL} = \frac{\Delta T_{TOTAL}}{\Delta T_{COL}} \qquad (17)$$

5. Design of the plate heat exchanger

To design the plate heat exchanger was calculated the heat exchange area and the number of plates required for the desired temperature increase. Overall, in the design of this heat exchanger was assumed that the following conditions are considered [5]:

- Heat loss to the surrounding is negligible.
- Air pockets are not formed within the region of the heat exchanger through which the hot fluid circulates.
- The overall heat transfer coefficient is constant throughout the exchanger.
- The temperature within each channel only varies along the flow direction.
- In parallel flow, the overall current is divided equally among all channels.

It was designed an algorithm in Matlab to compute the value of the heat transfer area and the number of plates of the plate heat exchanger. To run the algorithm, it necessary the value of the heat flow, some of the flow's temperatures, the flow of fluid Dowtherm Q (hot fluid) and specifications of the plates. Properties of the fluids, such as heat capacity, dynamic viscosity and thermal conductivity, were calculated by the algorithm from correlations that are a function of temperature. Such correlations were obtained by adjusting the reported experimental values of these variables to functions, which were almost entirely of polynomial type.

The design of the heat exchanger is based on the energy transferred between the fluids, which is equal to the total useful energy obtained through the system of parabolic dish collectors. If ***n*** is the total number of collectors, then:

$$Q' = nQu \qquad (18)$$

$$\dot{Q} = w_h (C_p)_h (T_e - T_s) = w_C (C_p)_C (t_s - t_e) \qquad (19)$$

It was used the last expression to determine the unknown temperatures of the fluids and flows. Once we found the values of input and output temperatures for each fluid, we calculate the logarithmic mean temperature difference and the number of transfer units - NTU, which is defined as the ratio of the increase in temperature experienced by the fluid being processed and the logarithmic mean temperature difference:

$$\Delta T_{ml} = \frac{(T_e - t_s) - (T_s - t_e)}{ln\left(\frac{T_e - t_s}{T_s - t_e}\right)} \qquad (20)$$

$$NTU = \frac{T_e - T_s}{\Delta T_{ml}} \qquad (21)$$

After computing the value of the NTU, we estimate the correction factor for the logarithmic mean temperature difference: F, through the graphical 13.16 reported in the book of Unit operations in Food Engineering[5].To determine the convective heat transfer coefficients, it is necessary to calculate the Reynolds number to identify the flow regime of each fluid. Depending on the flow arrangement (in series or parallel), the computing formula for Reynolds number varies. A plate heat exchanger with a configured flow in series, have aReynolds number expressed by:

$$Re = \frac{\rho v D_e}{\eta} = \frac{G D_e}{\eta} \qquad (22)$$

On the other hand, a plate heat exchanger with a configured flow in parallel, have a Reynolds number expressed

by the Eq. 23. In this equation n is the number of channels through which circulates each type of fluid.

$$Re = \frac{\left(\frac{G}{n}\right) D_e}{\eta} \qquad (23)$$

For a turbulent flow regime, the Nusselt number was computed using the expression:

$$N_U = 0.2536 (Re)^{0.65} (Pr)^{0.4} \qquad (24)$$

Once determined the Nusselt number, the convective heat transfer coefficient was calculated by the expression:

$$N_U = \frac{hD_e}{k(25)}$$

For a laminar flow regime (*Re*<400), the convective heat transfer coefficient can be find directly using the equation:

$$\text{h} = 0.742 C_p G (Re)^{-0.62} (Pr)^{-\frac{2}{3}} \left(\frac{\eta}{\eta_W}\right)^{0}.14 \quad (26)$$

The equivalent diameter that appears in the formulas for determining the Reynolds number and Nusselt number is defined as four times the hydraulic radius or the ratio between the area through which fluid flows between the plates and the wetted perimeter.

$$D_e = 4r_H = 4\frac{ab}{2a} = 2b \qquad (27)$$

where***a*** is the width of the plates and ***b*** the space gap between them.To determine the overall heat transfer coefficient, is necessary to know the resistance to heat transfer due to sediment deposits generated by the hot and cold flows. The values for these "fouling factors" were found in Table 13.1 of the book Unit Operations in Food Engineering. The expression used to calculate the overall heat transfer coefficient is:

$$1/U = 1/h_1C + 1/h_1H + e/k_1p + R_1C + R_1H \quad (28\ )$$

To calculate the total heat transfer area, we used the expression for the heat transfer rate, which includes the correction factor for the logarithmic mean temperature difference:

$$\dot{Q} = UA_z\Delta T_{ml}F \qquad (29)$$

$$A_T = \frac{\dot{Q}}{U\Delta T_{ml}F} = \frac{w_h (C_p)_h (T_e - T_s)}{U\Delta T_{ml}F} \qquad (30)$$

The total number of plates of a flat plate heat exchanger is easy to obtain once are known the total heat transfer area and the area of each plate (A_P). Thus, the expression to calculate the number of plates is:

$$N = \frac{A_T}{A_p} \qquad (31)$$

To determine the heat transfer from the pipe to the atmosphere, the thermal contact resistance between the surface of the steel pipe and the thermal insulation was not taken into account (see Figure 3). Thus, the expression for determining the rate of heat transfer to the atmosphere is.

$$Q_{LC} = \frac{T_{\infty 1} - T_{\infty 2}}{\frac{1}{(2\pi r_i L)h_1} + \frac{Ln\left(\frac{r_e}{r_i}\right)}{2\pi L\, k_{st}} + \frac{Ln\left(\frac{r_{ins}}{r_i}\right)}{2\pi L\, k_{ins}} + \frac{1}{(2\pi r_{ins} L)h_2}} \qquad (32)$$

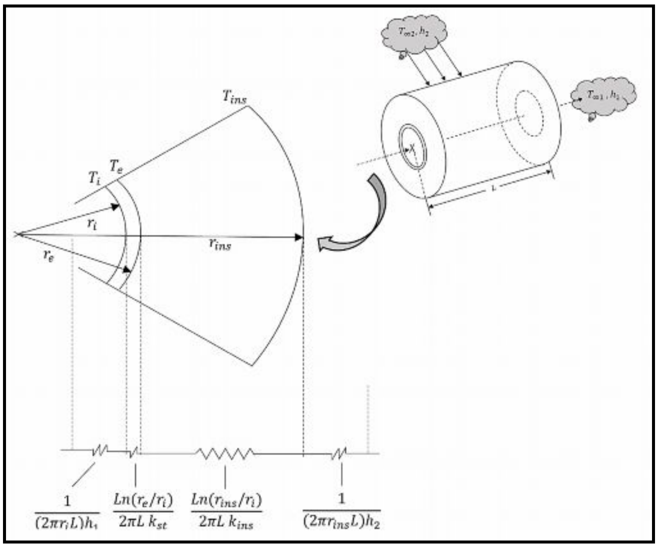

Figure 3.Representation of the piping line

The average temperature of the fluid Dowtherm Q ($T_{\infty 1}$) was taken as if it were constant and equal to the average between the inlet and outlet temperature of this fluid, while the temperature of the film air ($T_{\infty 2}$) was taken as an average between the temperature of the outer surface of the insulation and the ambient air temperature. Equation 33 was used to determine the convective heat transfer coefficient from the liquid Dowtherm Q to the inner wall of the pipe (h_1). Where D_i represents the internal diameter of the steel pipe and k_{DT} is the thermal conductivity of the liquid Dowtherm Q. This equation can be used for laminar flow conditions fully developed through a pipe with constant circular section.

$$Nu_D = \frac{h_1 D_i}{k_{DT}} = 4{,}36 \qquad (33)$$

To determine the convective heat transfer coefficient from the outer surface of the thermal insulation to the atmosphere, we used a correlation to estimate the average Nusselt number over the entire cylindrical surface of the pipe. This correlation is shown in eq. 34 and 35, where α_{air} represents the air thermal diffusivity, ν the air kinematic viscosity, and β the air coefficient of thermal expansion, K^{-1}.

$$\overline{Nu}_D = \left\{0{,}6 + \frac{3{,}87\, Ra_D^{\frac{1}{6}}}{\left[1 + \left(\frac{0{,}559}{Pr}\right)^{\frac{9}{16}}\right]^{\frac{8}{27}}}\right\}^2 \qquad (34)$$

$$Ra_D = \frac{g\beta(T_{ins} - T_{\infty 2})D_{ins}^3}{\nu\alpha_{air}} \qquad (35)$$

6. Conclusions

Mathematical model of the concentrating system of solar power was designed based on the thermodynamic fundamentals. System consist of three parabolic dish solar collectors and a plate heat exchanger. In addition, a pump was designed to circulate the heat transfer fluid through the receiver, and the concentrated solar energy heats the fluid as it passes through it. The hot thermal fluid is then pumped to a plate heat exchanger that transfers the thermal energy produced to the air. In this way, the company can reduce the costs associated with natural gas consumption and the adverse environmental effects caused by the carbon dioxide emissions.This solar technology is possible in Cartagena city due to high solar radiation to generate thermal energy, which will be determined in the next research.

Acknowledgements

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Notations

a : Width of plates, m
A_a : Dish concentrator aperture area, m^2
A_C : Entrance aperture area of receiver, m^2
A_P : Area of a plate, m^2
A_T : Total heat transfer area of plate heat exchanger, m^2
A_W : Cavity internal area of receiver, m^2
b : Distance between the plates, m
I_s : Solar Irradiance, W/m^2
Nu_L : Nusselt number based on length L
NTU : Number of Transfer Units
Q_S : Solar energy incident on the dish collector, W
Q_R Solar energy falling on the receiver, W
Q_L : Heat losses from the receiver to surroundings, W
Q_{LK} : Conductive heat loss from the receiver, W
Q_{LC} : Convective heat loss through the receiver aperture, W
Q_{LR} : Radiative Heat loss through the receiver aperture, W
Qu : Useful energy collected, W

Greek letters:

η_o : Optical efficiency of parabolic dish collector
λ : Fraction of un-shaded area
ρ : Reflectivity of the parabolic dish reflector.
τ : Transmittance
α : Absorptance.
γ : Intercept factor of receiver.
θ : tilt angle of cavity
β : Volumetric expansion of the ambient air, K^{-1}
ε_{ef} : Effective emittance of cavity
ν : Kinematic viscosity of air, m^2/s
η_C : Dinamic viscosity of air, Pa*s

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