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Thermo-hydraulic Design of Solar Collector Networks for Industrial Applications

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The design and selection of banks of solar energy collectors for thermal applications requires that two simultaneous design objectives be met: the working fluid must provide the heat load to the process and this must be supplied within the specifications of pressure drop. In this work, a solar collector will be termed heat exchanger and the total collector surface area is referred to as the network of solar collectors (NSC). A NSC is used in large scale applications and they can exhibit arrangements in series, parallel or any combination of these. Contrary to domestic applications where water flow through the exchanger on natural convection, in the case of large scale applications the flow of water is forced through the use of a pumping system. This paper looks at the selection of the most appropriate arrangement for a given application. The approach presented in this paper is based on the assumption of constant wall temperature; however, it can easily be extended to account for the case of variable wall temperature. The approach is graphically displayed with the length of the exchanger plotted against the number of arrays in parallel. Two curves are produced: the curve that shows the thermal length (constant heat load) and the hydraulic length (constant pressure drop). The thermal length is the dimension for which the exchanger meets the required heat duty and the hydraulic length is the dimension for which the exchanger absorbs the specified pressure drop. The point where the two curves meet determines the network structure that fulfils the required heat duty at the specified pressure drop. A systematic approach to solve this design problem is looked at in this work.

1. Introduction

A solar collector is a type of heat exchanger where a fluid absorbs energy from a solid surface exposed to solar radiation. Depending of the temperature that can be achieved these types of equipment can broadly be classified into: low temperature, medium temperature and high temperature. In this work we concentrate on low temperature solar collectors where the maximum temperature achieved by the working fluid is below 100 °C. The thermal characterisation of these types of devices is based on the determination of the collector efficiency and the heat losses to ambient. Growing interest in reducing the capital cost of these systems has focused on increasing the thermal efficiency at the expense of optimising the use of materials of construction (Eisenman et al., 2004).

Liu et al. (2012) have looked at the application of solar heating systems for industrial applications where a considerable amount of hot water is required. In their study they considered the use of a set of solar collectors arranged in series and the system was experimentally tested to check for its sufficiency in supplying the required heat load. Zago et al. (2011) have investigated the energy efficiency of independent and centralised heating systems considering the integration of solar plants; auxiliary energy consumption such as pumping power is determined and it is shown how this additional energy consumption tends to reduce the overall energy efficiency in buildings. Lin et al. (2012) looked at the installation of large scale solar water collector fields for applications in dormitories, swimming pools, restaurants, and manufacturing plants. An installation of three cascade arrangement was investigated; each cascade consisted of the following: two with a six-collector array and one with a four-collector array. The work concentrated on evaluating the performance of the existing collector field; no consideration is given to the actual design of the bank of collectors and the pumping power. Kiraly et al. (2012) present the integration of renewables

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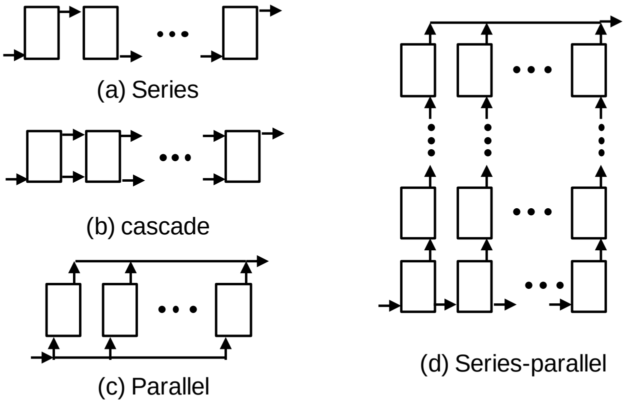
within an existing large scale meat company. The renewable energy sources considered are: solar, biomass, certain types of waste, and geothermal energy sources. Quijrea et al. (2012) look at the thermal integration of solar energy in a fish canning process using a combined pinch analysis and exergy analysis approach.

Pressure drop in the design of water networks for energy distribution was considered by Carravetta et al. (2013) who found that large energy savings can be achieved by exploiting the pressure drop by a network pressure control strategy in a series-parallel hydraulic circuit. Garg (1973) presents some aspects of the design of solar water heaters suitable for the large and intermittent demands for hot water in hospitals and hostels. He experimentally analyzed various arrangements for connecting the field of solar collectors such as cascade, series, series-parallel and true parallel. From his results a system with a large number of solar collectors finds the maximum efficiency and economy when the arrangement is true parallel. Karagiorgas et al. (2001) performed an economic evaluation of industrial applications of solar thermal installations in Greece. They analyzed the case of applications in the food industry such as: dairy products, cold cut and process meat factories, pastry and cake confectioneries, olive oil refineries, tinned goods, slaughterhouses; agro-industries such as: solar drying, horticulture–nursery greenhouses, slaughterhouses, meat processing, livestock landings; textiles such as: tanneries, leather treatment, cloth, refineries, textile treatment workshops; chemical industry such as: cosmetics, detergents, pharmaceuticals, wax, distilleries, breweries and beverage industry. In their work they do not make reference to the actual arrangement of the bank of solar collectors and the installed surface area goes from 50 m^2 to as large as 2,700 m^2.

The purpose of this work is to link the thermal design with the hydraulic design for the determination of the arrangement of the network of solar collectors that will meet both the required heat load and the specified pressure drop.

2. Solar collector-network arrangement

Figure 1 illustrates the four types of arrangements in which a bank of solar collectors can be arranged. These are: series, cascade, pure parallel and series-parallel. The cascade arrangement works in a similar manner as the parallel arrangement. So, for the purposes of this work, it will not be considered separately. The choice between any of the arrangements responds to the water flow rate to be handled and to the amount of heat to be delivered. Low flow rates are easily handled in a series arrangement. As the heat load or exit temperature requirements increase, the number of collector in series increases. Parallel branches are added as the flow rate increases.


(a) Series

(b) cascade

(c) Parallel

(d) Series-parallel

Figure 1: Typical arrangements of banks of solar collectors

3. Thermo-hydraulic model

The heat load of the exchanger can be calculated from the energy balance as:

$$Q = \dot{m} \; C_p \; \Delta T \qquad (1)$$

Where Q is the heat load, $\dot{m}$ is the water flow rate and ΔT is the temperature rise of the water given by (T_{out}-T_{in}), where T_{out} is the water outlet temperature and T_{in} is the inlet temperature. From the design equation and assuming that the collector absorbing surface is at a constant temperature (T_w), the required surface area is:

$$A = {}^{Q}\!\big/_{h\ \Delta \mathrm{T}_{\mathrm{LM}}} \tag{2}$$

Where A is the heat transfer surface area, h is the film heat transfer coefficient and ΔT_{LM} is the logarithmic mean temperature difference available for heat transfer. If the collector surface area (A) is expressed in terms of the geometry of the collector then we have that:

$$A = \pi\ d_i\ L_t\ \mathrm{N_t}\ U_p \tag{3}$$

Where d_i is the tube internal diameter, N_t is the number of tubes per collector and U_p is the number of collectors in parallel. The thermal length (L_t) is the exchanger length required to meet the required het load. Combining Eqs (1), (2) and (3) and rearranging we have that the thermal length is:

$$L_t = \frac{\dot{m}\ \mathrm{C_p}\ \left(T_{out} - T_{in}\right)}{\pi\ d_i\ \mathrm{N_t}\ U_p\ h\ \Delta T_{LM}} \tag{4}$$

The film heat transfer coefficient can be calculated from the expression:

$$h = \frac{4200\ \left(1.35 + 0.02\ \mathrm{T}\right)\ v^{0.8}}{d_i^{0.2}} \tag{5}$$

It is important to note that in the equation above d_i must be used in (mm); T is the water temperature, and v is the velocity, which can be calculated form:

$$v = \frac{\dot{m}}{\rho\ A_c} \tag{6}$$

Where the term ρ is the density of water and A_c is the free flow area that can be expressed as:

$$A_c = \frac{\pi\ d_i^2}{4}\ N_t\ U_p \tag{7}$$

The hydraulic length (L_h), is the exchanger length required to meet the specified pressure drop. The pressure drop across the core of the exchanger can be expressed by:

$$\Delta P = \frac{2\ f\ \rho\ L_h\ \mathrm{v}^2}{d_i} \tag{8}$$

Where f is the friction factor which for Reynolds numbers less than 2100 is given by:

$$f = \frac{16}{\mathrm{Re}} \tag{9}$$

And for Reynolds numbers greater than 2100 is given by:

$$f = \frac{0.054}{\mathrm{Re}^{0.2}} \tag{10}$$

Where the Reynolds number is expressed as:

$$\mathrm{Re} = \frac{d_i\ \dot{m}}{\mu\ A_c} \tag{11}$$

Combining Eqs (6), (7) and (8) and rearranging we get:

$$L_h = \frac{\Delta P \; \rho \; d_i^5 \; N_t^2 \; U_p^2 \; \pi^2}{32 \; f \; \dot{m}^2} \tag{12}$$

Eqs (4) and (12) can be solved for the given water flow rate and specified pressure drop for a number of collector arrays. The intersection between the curves that represent each of these dimensions on a length versus number of parallel branches is the ideal network structure.

4. Application of the methodology

The methodology is demonstrated on a set of case studies and Table 1 shows the geometrical and operating data. For the analysis, flat plate collectors are considered. They consist of an aluminum sheet blackened at the exposed side and attached to a set of galvanized pipe of 0.019 m diameter with 0.l0 m spacing between tubes. The dimensions of the collector are 1.2 × 0.75 m thus fitting 7 tubes for a total heat transfer surface area of 0.5 m^2 per collector. The cross sectional area for water flow is 1.9847×10^{-3} m^2. For the study, a constant wall temperature of 80 °C is considered. Figure 2 shows a flow diagram of the calculation procedure. Figures 3(a), (b), (c) and (d) show the results of the design.

Table 1: Geometrical and operating data for the case studies

	Case 1	Case 2	Case 3	Case 4
Inlet temp. (°C)	25	25	25	25
Outlet temp. (°C)	55	55	55	55
Water flow rate (kg/s)	5	10	40	10
Pressure drop (kPa)	15	15	15	5

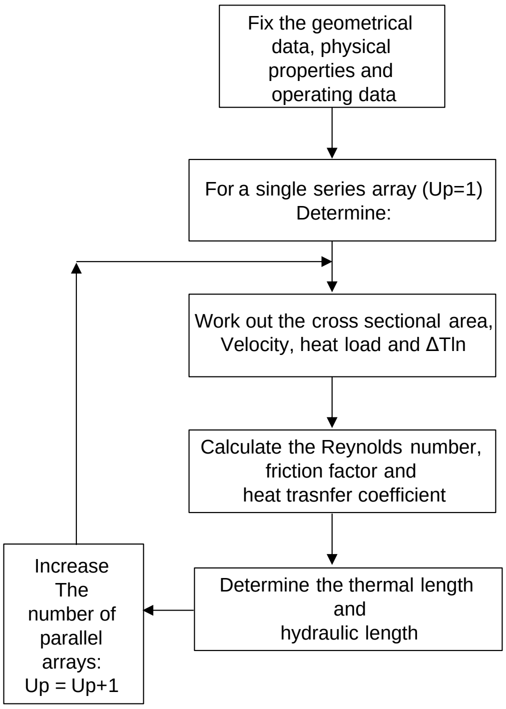
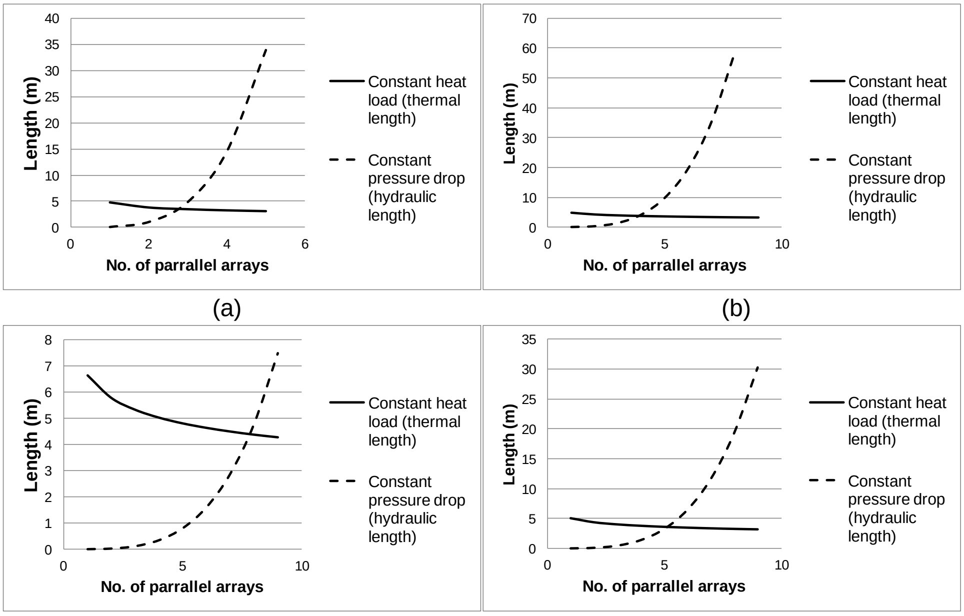


Figure 2: Flow diagram of the calculation procedure

The results shown in Figure 3 indicate that the number of solar collectors in series is determined by the temperature approach or the heat load; this is, the larger the heat load the larger the number of collectors in series required; on the other hand, the number of arrays in parallel are related to the use of pressure drop. For instance, for the same pressure drop of 15 kPa, as the water flow rate is increased thus increasing the heat load, both, the number of collector in series and the number of arrays in parallel also increase. For the same water flow rate as the allowable pressure drop is reduced, the number of collectors in series and parallel also increase. The model permits the analysed of a number of arrangements, like pure parallel and to explore the effect of the collector dimensions upon the total number of units required to meet the heat duty and the hydraulic specifications. From the results of Figure 3, the design objectives of case 1 are achieved by installing three parallel arrays of four collectors each; this is a total of 12 collectors. Case 2 is solved with the following combination: four parallel arrays each with four collectors in series giving a total of 16 collectors. Case 3 is solved as follows: eight parallel arrays with four collectors each given a total of 32 solar collectors. Case 4 is solved with six parallel arrays with three collectors each giving a total of 18 collectors. If the temperature approach is reduced, say if the water outlet temperature is allowed to reach 65 °C, the results are as follows: case 1: four parallel arrays with 5 collectors each, giving a total of 20 units. Case 2: five parallel arrays with five collectors each, giving a total of 25 collectors. Case 3: nine parallel arrays with 6 collectors in series each giving a total of 54 collectors. Case 4: six parallel arrays each with five collectors giving a total of 30 collectors.



(a) (b)

(c) (d)

Figure 3: Design results of a network of solar collectors for different flow rates and pressure drops: (a) 5 kg/s, 15 kPa, (b) 10 kg/s, 15 kPa, (c) 40 kg/s; 15 kPa, and (d) 10 kg/s; 5 kPa

5. Conclusions

A simple and easy to implement model for the design of networks of solar collectors for large scale application of solar energy has been introduced. At this stage, the model has been applied solely to the case of flat solar collectors; however, it can easily be extended to other type of technology such as evacuated tubes. An important consideration in the approach presented in this work is that the temperature of the absorber surface is assumed constant. This simplification can easily be removed by incorporating a model for the determination of the solar energy absorbed as a function of time. The analysis of any network arrangement can readily be carried out using this model. For instance, a pure parallel unit can be assessed by changing the geometry of the unit, this is, if the unit is originally said to be made up of "*n"* number of tubes, the number of tubes can be increased by "m" times or "*m x n*" and this exchanger is modelled as a combination of *m* parallel units. From the results, it can also be concluded that the number of unit in series is directly related to the heat load or the temperature approach, whereas the number of arrays in parallel is related to the use of pressure drop or the mass flow rate. The larger the flow rate, the larger the number of parallel arrays. This behaviour is similar to the case of reduced pressure drop for the same water flow rate.

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