Experimental comparison of serpentine and parallel-tubes flat plate collectors for different flow rates

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ABSTRACT

Most of solar domestic hot water systems (SDHW) use flat plate solar collectors of two configurations, with regard to the hydraulic configuration: parallel tubes and serpentine.. Serpentine type collectors simplify the hydraulic connection of many collectors, the manufacturing costs is lower and work better at low flow rates, improving thermal stratification in the water tanks. However, these collectors offer greater hydraulic losses in the solar circuit, which increases the economic cost associated with pumping.

On the other hand, parallel tube collectors have a greater market share, especially in moderate climates with high radiation, such as the Mediterranean, and particularly in domestic thermosyphon-type solar installations, despite offering greater difficulty in the automatization of their manufacturing. This type of collector offers lower hydraulic losses. Nevertheless, when the mass flow is low, hydraulic problems may appear due to the unbalanced flow between riser tubes.

In the paper, two flat plate solar collectors were simultaneously tested in a side-by-side facility at the UMA, following the steady-state procedure proposed by the EN12975-2:2006 standard. Both collectors have been manufactured at the same date by the same company and belong to the same product range.

To demonstrate the relationship between the thermal performance of each collector and the mass flow rate, both collectors were tested for a total of seven flows, which covered all the technically possible flows in a real SDHW installation. Next, the measurement uncertainties were propagated and the different performance curves for each test were obtained, which allowed to have the three characteristic curve coefficients that are usually used to compare different collectors in terms of their operation (collector efficiency and linear and quadratic heat losses coefficients). The paper analyses the suitability of each type of absorber plate configuration and the influence of the mass flow rate. The paper recommends a type of collector depending on their working conditions.

Keywords: *Flat plate collector, solar energy, variable flow, serpentine, parallel tube, SST, experimental testing*

1. Introduction

The goal of this article focuses on comparing the performance of two identical solar thermal collectors, differing only in the hydraulic configuration of their absorber plates: parallel-tubes and sepentine. To achieve this, the thermal efficiency is obtained from several efficiency test based on EN12975-2:2006 standard. In addition, it is checked how the flow rate affect each type of collector, as the test are done for various flow rates within the normal operating range for SDHW systems. A high number of experimental tests, on two identical solar collectors, were performed simultaneously, so that the results obtained reduce any external influence, being dependent only on the existing heat transfer mechanism inside each collector. To assure the validation of the results, all the facilities and sensors used meet the requirements of the current solar collector efficiency test standard. The research also applied an uncertainty propagation of the measurement in the final results, an essential point to obtain clear conclusions.

Many studies show the mistake resulting when comparing solar collectors based only on the three efficiency test curve coefficients, (Osorio, 2014; Rojas, 2008; and Zambolin, 2010). These three coefficients: zero loss collector efficiency, (η_0), collector heat loss linear coefficient, (k_1), and collector heat loss quadratic coefficient, (k_2), are obtained after applying a MLR, (Multi Linear Regression), statistical analysis to a data set recorded from the tests, which must satisfy the conditions included in the standard to assure steady state flow. In addition, these data samples can be selected freely by the test laboratory, making that the efficiency predicted by the test curve obtained adjusts as best as possible with the data recorded in the test, (obtaining a coefficient of determination R^2 near to 1). Since it is essential that the final result be affected by the uncertainty of measurement, it is necessary to propagate the uncertainty based on official methods, such as the one proposed in (GUM, 1995).

To calculate the efficiency of both types of collector, and see their dependence in terms of the flow rate, we can quote (Duffie, 2006), where the efficiency of a solar collector, η, is obtained according to Eq.1:

$$\eta = \frac{Q_u}{A_U \, G_T} \qquad (1)$$

The useful collector power gain of the collector, Q_U, is obtained according to Eq.2, where F_R and U_L are the collector heat removal factor and the collector overall heat loss coefficient, G_T is the irradiance on collector tilted plane, A_U is the useful area of the collector, t_{in} is the inlet temperature to the collector and t_a is the ambient air temperature.

$$Q_U = A_U \; F_R \; [G_T - U_L \; (t_{in} - t_a)] \qquad (2)$$

Applying these equations, and in the case that the only difference between the two collectors is the type of absorber, the value of the flow rate has a great importance in the heat transfer mechanism for each case. This mechanism depends mainly on the Nusselt number inside the tubes, which in turn depends on the internal flow regime in the absorber tubes, (laminar, transition or turbulent), This regime is estimated by the Reynolds number and depends directly on the flow rate. These theoretical conditions suppose that there are no hydraulic imbalances between risers.

If the flow inside the risers is reduced, the heat transfer to the fluid decreases, so that the average temperature of the absorber, t_m, increases. This make the heat losses to the environment to rise. The result is a reduction of both the collector efficiency factor, F', and the heat removal factor, F_R, which are obtained according to Eq.3 and Eq.4, which, as seen above, reduce the thermal efficiency.

$$F' = \frac{\frac{1}{U_L}}{W\left[\frac{1}{U_L(D + (W - D)F)} + \frac{1}{C_b} + \frac{1}{\Pi \; D_i \, h_{fi}}\right]} \qquad (3)$$

$$F_R = \frac{\dot{m}\, C_P}{A_U \, U_L}\left[1 - exp\left(-\frac{A_U \, U_L \, F'}{\dot{m}\, C_P}\right)\right] \qquad (4)$$

In the previous equations, W, D, F, C_b and D_i are values that depend on the design and construction of the absorber plate, while h_{fi} depends on the properties of the fluid: flow and temperature, since these are the ones that define the type of existing flow regime. To compare the variation of thermal efficiency between collectors with different flow rates, it is needed to apply an efficiency test method, such as the one presented in EN12975-2:2006 standard, which generates test curves according to Eq.5, Eq.6 and Eq.7, where t_{in} is the inlet fluid temperature to the collector, t_{out} is the collector outlet fluid temperature, t_m is the average temperature and T* is the average reduced temperature difference:

$$\eta = \eta_0 - k_1 T^* - k_2\, G_T\, {T^*}^2 \qquad (5)$$

$$T^* = \frac{t_m - t_a}{G_T} \qquad (6)$$

$$t_m = \frac{t_{out} + t_{in}}{2} \qquad (7)$$

Predicting what really happens in the internal flow of the absorber tubes is a complex process, whereof there are several references, especially in the case of parallel-tubes collectors, where it can be found hydraulic imbalance problems. We can mention (Bava, 2016; Chiou, 1982; Façao, 2015; Jones, 1994; Weitbretcht, 2002 and Wang, 1990). In all these works, it is concluded that a maldistribution of flow between risers, (which would be the same as a poor distribution of flow between collectors connected in parallel), generates quantifiable efficiency losses, which are produced by the increase in t_m produced in the points with the lowest flow rates. Other references quantify the thermal efficiency improvement obtained when conditions of turbulent regime are induced inside risers, by inserting wire coils, reporting a little improvement in efficiency but not providing data on the final uncertainty of the results, (Herrero, 2011).

2. Material and methods

The research is based on the use of a side-by-side test facility, which allows to test two collectors at the same time, as shown in Fig. 1. The entire set of sensors complies with the terms of EN12975-2:2006 standard, with two exceptions: the class of the pyranometers and the precision of the temperature probes are different from the required, as is shown in Table 1. This has not been inconvenient to obtain the results fixed as the target of the article, like it shows the value of the coefficients of determination obtained when comparing the calculated and the predicted efficiency, for more than 900 valid points for parallel-tubes collector, (R^2 = 0.9917) and for serpentine collector, (R^2 = 0.995), which are represented in Fig. 2.

The test method is based on the steady state test method, SST, of the EN12975-2:2006 standard, which uses test data sets recorded during a minimum period of operation, to ensure that there are sufficient registration points for each inlet collector temperature. These zones with different inlet temperatures are known as stages, and as a general rule, it is established that for each test there must be at least 4 stages. The standard does not establish anything respect the value for the flow-rate selected for the test, leaving freedom for the manufacturer to select the flow rate that consider optimal to test the equipment. The conditions that must be kept to consider steady state are shown in Table 2. Once there are sufficient valid registration points obtained, a data selection is made, choosing periods of operation that show the least variation of their main variables: flow and temperatures. Next, the three efficiency curve coefficients are obtained, for each collector and for each flow rate tested, after an MLR analysis performed in Matlab, (Fig. 3), which is based on Eq. 5. Finally, with these coefficients and with the measurement accuracies of the different sensors, the measurement uncertainty is propagated in another Excel spreadsheet, obtaining the final efficiency curve coefficients together with their associated measurement uncertainty.

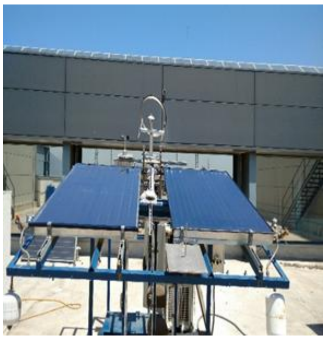


Figure 1. Side-by-side testing facility at Universidad de Málaga.

Table 1. Sensor accuracy comparison between standard EN12975 requirements and those installed.

Measurement	**EN12975-2:2006**	**UMA Test Facilities**
Fluid Temperatures	± 0,1 K	± 0,2 K
Irradiance	Class I	Class II
Ambient temperature	± 0,5 K	± 0,2 K
Flow rate	± 1 %	± 0,25 %

Table 2. Standard EN12975-2:2006 maximum deviations allowed for steady-state tests.

Measurement	**EN12975-2:2006**
Fluid Temperatures	± 0,1 K
Irradiance	± 50 W/m^2
Ambient temperature	± 1,5 K
Flow rate	± 1 %

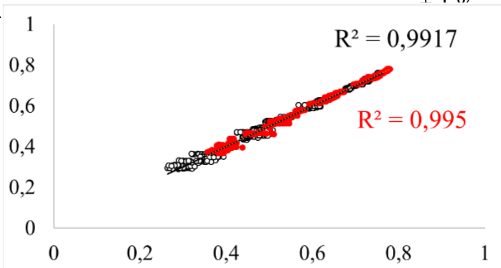


Figure 2. Coefficients of determination R^2 between predicted and measured efficiencies for both collectors tested.

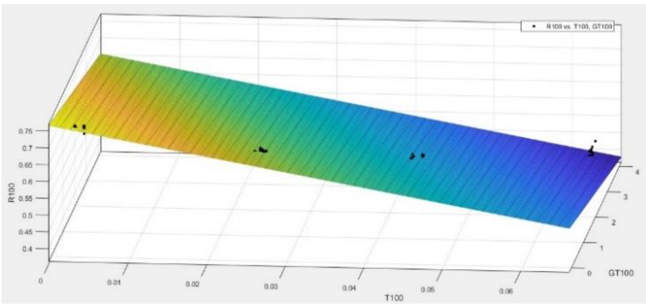


Figure 3. MLR statistical analysis results in Matlab for one of the test done.

Table 3 shows all the values obtained, for each of the 7 flows tested in both collector. For each flow, 2 valid tests were carried out on 2 different days, performing 1 simultaneous test per day. The values of the coefficient of determination and the total valid samples studied, n_{total}, show that more reliable data were obtained for the serpentine collector tests, as in the last stages, the parallel-tubes collector showed short-term stationary state zones, affecting the final results of the goodness of fit.

Both collectors have the same geometric and thermal physical properties, only differing in the hydraulic configuration of their absorber plate, presented in Fig. 4. The technical data of each collector are shown in Table 4. The flow limits selected in the tests were 40 l/h, (20 l /(hm^2)) and 160 l/h, (80 l/(hm^2)). These were chosen for two limitations: a) lower flow rates causes the vaporization of the fluid in the last stages of the tests, invalidating the data records and discarding the test. b) higher flows will generate higher hydraulic head losses, which are not real work conditions in typical solar thermal installations.

Table 3. Number of valid points, R-squared coefficient of determination, efficiency curve parameters and their associated uncertainties for each collector and flow rate tested.

				Parallel-tubes				
Flow (l/h)	**n_{total}**	**R^2**	**η_o**	**$u(\eta_o)$**	**k_1**	**$u(k_1)$**	**k_2**	**$u(k_2)$**
40	121	0,9899	0,7375	0,0013	-6,0880	0,1205	0,0002	0,0023
60	138	0,995	0,7398	0,0015	-6,3511	0,1306	0,0130	0,0023
80	108	0,9922	0,7229	0,0020	-3,2334	0,1636	-0,0594	0,0025
100	119	0,9918	0,7409	0,0020	-5,3705	0,1534	-0,0110	0,0024
120	144	0,9898	0,7358	0,0018	-5,0658	0,1402	-0,0312	0,0023
140	151	0,9938	0,7355	0,0020	-5,3897	0,1563	-0,0235	0,0025
160	139	0,9899	0,7371	0,0022	-5,4187	0,1758	-0,0174	0,0028
				Serpentine				
Flow (l/h)	**n_{total}**	**R^2**	**η_o**	**$u(\eta_o)$**	**k_1**	**$u(k_1)$**	**k_2**	**$u(k_2)$**
40	144	0,9904	0,7488	0,0015	-4,2018	0,1255	-0,0266	0,0023
60	118	0,9958	0,7541	0,0016	-3,9353	0,1345	-0,0303	0,0023
80	110	0,9923	0,7618	0,0021	-4,7397	0,1551	-0,0215	0,0023
100	117	0,996	0,7601	0,0022	-4,0712	0,1604	-0,0270	0,0023
120	132	0,9923	0,7587	0,0019	-4,4765	0,1435	-0,0223	0,0023
140	152	0,9976	0,7626	0,0020	-4,4190	0,1573	-0,0246	0,0024
160	139	0,9973	0,7618	0,0022	-4,6263	0,1765	-0,0179	0,0028

Figure 4. Flat plate solar collectors tested. (Left, parallel-tubes, right, serpentine.)

Table 4. Data sheet of tested collectors.

Dimensions	Parallel-tubes	Serpentine
Length (mm)	2.130	
Width (mm)	970	
Thickness (mm)	83	
Gross area (m^2)	2,07	
Plate		
Length (mm)	2.088	
Width (mm)	925	
Useful area (m^2)	1,93	
Risers tubes D_{ext} x thickness (mm)	8 x 1	
Manifold tubes D_{ext} x thickness (mm)	18 x 1	
Type	8 Parallel tubes	1 Serpentine
Material	Selective, aluminium-CERMET sputtering	
Welding	Laser	
Absorptance (%)	95	
Emittance (%)	5	
Insulation		
Type	Backside	
Length x Width (mm)	2.100 x 960	
Thickness (mm)	40	
Conductivity (W/(mK))	0,034 (at 20 ºC)	
Glass		
Type	Tempered, low Fe_2O_3	
Thickness (mm)	3,2	
Solar transmittance (%)	>90	
Others		
Fluid capacity (l)	1,02	1,29
Weight (kg)	37	32

3. Results and Discussion

To analyze the influence of the flow rate on the operation of each type of collector, it must be reminded that the only difference between both collectors is the hydraulic configuration of each absorber. It may appear hydraulic imbalances between risers in the case of the parallel-tubes collector and, on the other hand, the hydraulic configuration determines what type of internal flow regime will exists for each flow and temperature couple.

Fig. 5 and Fig. 6 show the theoretical conditions that could be found inside the absorber tubes of each collector for the temperature-flow conditions recorded during each test.. The transition Reynolds numbers is 2.300, for the change of laminar regime to transition regime, and 10.000 for fully developed turbulent regime. In the typical SDHW, the flows for the parallel-tubes collector would make the internal flow regime always remain in the laminar zone, beginning to be transition regime with 120 l/h and 65ºC. In the case of serpentine collector there would be conditions of transition regime from 40 l/h, and from 80 l/h it is possible that it starts to enter in turbulent regime zone, if the average temperature of the fluid exceeds 60ºC.

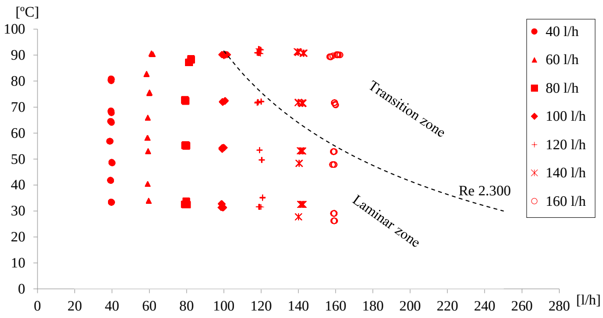


Figure 5. Flow regimes inside collector's absorber risers tube during each test. Parallel-tubes collector.

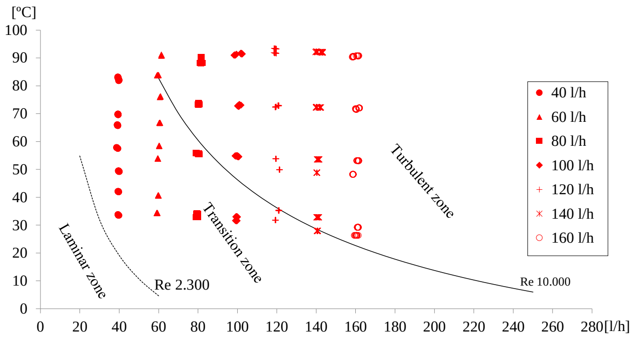


Figure 6. Flow regimes inside collector's absorber tube during each test. Serpentine collector.

To show clearly the analysis of each collector performance, we have presented different figures that include the efficiency curves for an irradiance of 1.000 W/m^2. In the case of lower irradiances, the results remain valid. After the tests carried out, we can verify that the serpentine collector has a higher efficiency if compared with the parallel-tubes collector. This statement is evident even without the need to perform the entire process of data processing and statistical analysis, as can be seen in Fig.7, where all the valid points recorded during all the tests are represented for each collector.

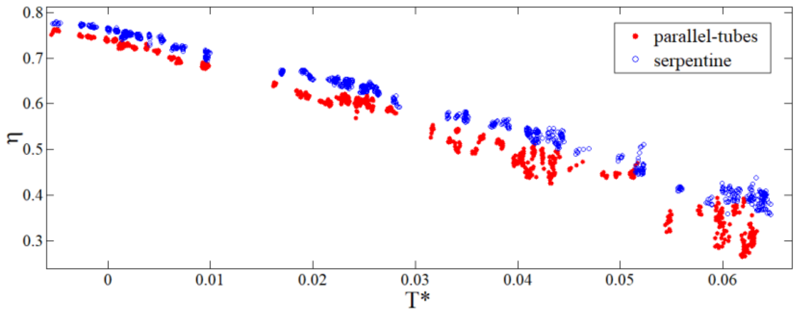
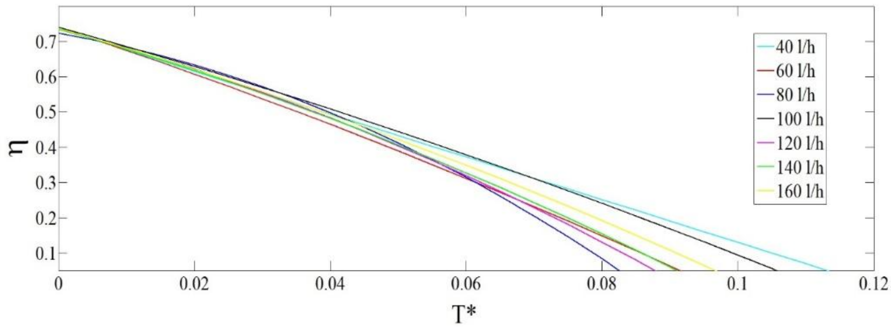


Figure 7. Total valid data set recorded during all the test for both type of collectors for an irradiance of 1.000 W/m^2.

Fig.8 and Fig.9 show the curves obtained for each collector and flow rates tested. As it can be seen, in the range of low T*, the efficiency is high, close to its maximum value, η_0, and the curves of the different flows are grouped for both collectors, being no great differences in terms of efficiency according to the flow rate selected. The differences between collectors appear when increasing the average temperature of the working fluid, that is, when T* increases above 0.05. In this range, the efficiency differences for different flow rates are much more marked in the parallel-tube collector. This is because under these conditions, the fluid begins to have substantial changes in its physical properties, as it approaches the zone of phase change. The viscosity and density begin to vary more sharply and, in addition, the collector produces little useful energy, because h_{fi} becomes a controlling factor.

Figure 8. Efficiency curves obtained from each test at different flow rates and an irradiance of 1.000 W/m^2. Parallel-



tubes collector

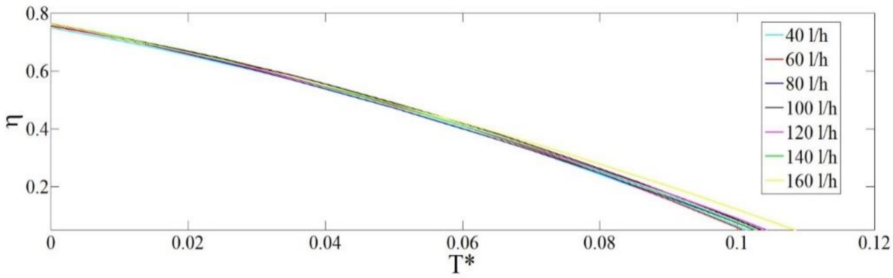


Figure 9. Efficiency curves obtained from each test at different flow rates and an irradiance of 1.000 W/m^2. Serpentine collector.

In the case of flow rates imbalances and average temperatures close to 100 °C, hot points begin to appear in the lower flow areas and the fluid is vaporized, generating air within the risers. This causes that the curves obtained when representing Eq.5 for each flow and collector are separated in their final zone, where there is a lower efficiency. Fig. 10 compares the curves obtained by each collector for the 7 flow rates tested.

Another outstanding point is the limitation of the conclusions obtained once the uncertainty of measurement is propagated, applying what is proposed by (GUM, 1995), in a similar way to works like (Mathioulakis et al., 2004; Sabatelli et al. , 2002; Domínguez Muñoz, 2008; Budig et al., 2009). In this way, instead of thermal performance curves, we compare thresholds with greater probability of finding the true value of the thermal efficiency of the collector, which allows us to know if the differences obtained are significant, or, on the contrary, remain inside the uncertainty zone, being unable to make definitive statements about which flow-collector combination is recommended for certain working conditions, represented by T*.

This situation is verified in Fig.11, where, for simplicity, the efficiency thresholds obtained for the limit flow rates tested, 40 l/h and 160 l/h, have been represented. As it is observed, with average low temperatures, where the collectors offer efficiencies close to the maximum, the difference between collectors is clear up to T* = 0.05, being a better option to select the serpentine collector, both with 40 l/h and with 160 l/h. In addition, it is observed that it is not possible to know the efficiency difference for each collector between the two limit flows tested: it cannot be established which flow is more favorable, since both are within the same probability threshold. If we analyze what happens from T* = 0.06, we can only conclude that the serpentine collector would have higher performance than the parallel-tubes one if both select 160 l/h, however, in the case that 40 l/h were selected, there is no clear conclusion, and both collectors would offer equivalent results. As T* increases, in both collectors the thresholds for each flow are extended, so it cannot be concluded that there is any favorable condition over another, neither of collector nor of flow. As a general rule, this situation has little importance, since it corresponds to conditions where the solar collectors should not work for long periods of time, since they reflect conditions in which the temperature of return of the fluid to collectors is very high, so that the collector hardly gives heat to the fluid, so the efficiency is minimal. If these conditions are maintained, conditions of overheating are usually reached.

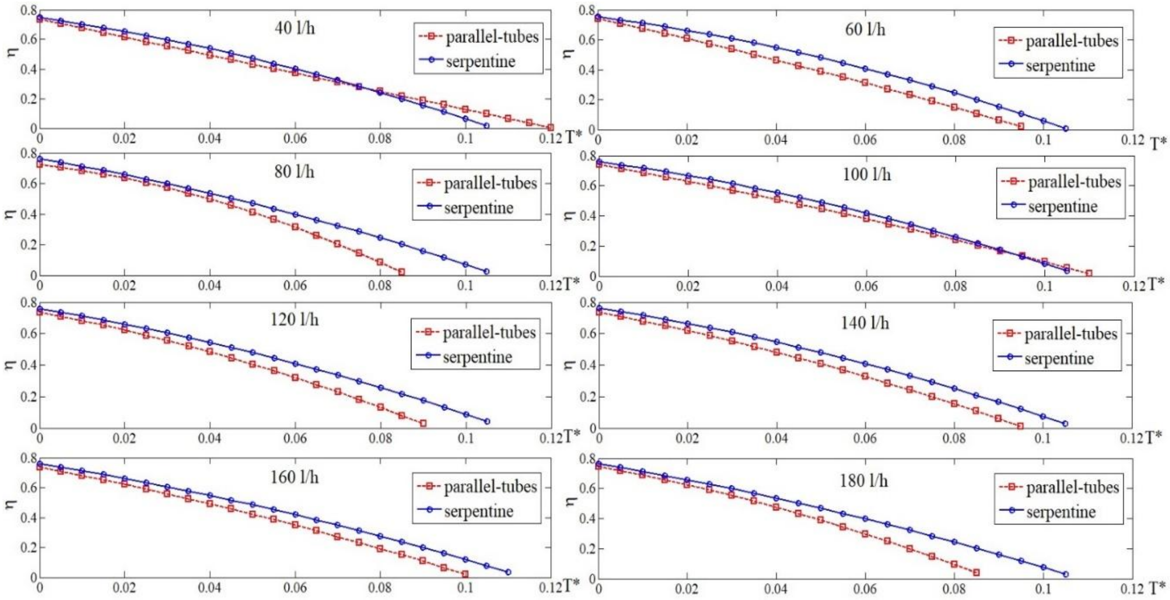


Figure 10. Comparison between efficiency curves of both type of collectors under each flow rate tested for an irradiance of 1.000 W/m^2.

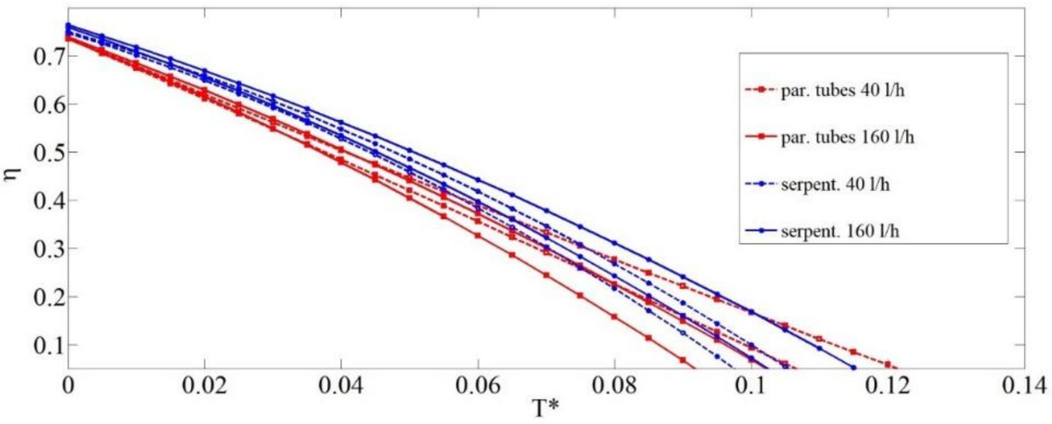


Figure 11. Comparison between efficiency curves after uncertainty propagation for both type of collectors under limit flow rates tested. Irradiance of 1.000 W/m^2

To estimate an advisable flow rate for each type of collector, the influence of the flow must be considered not only on the collectors, since the flow also takes an important role in other aspects when sizing SDHW systems, (Plaza Gomariz et al., 2019). From an economic point of view, low flow achieves the lowest initial investment and operating costs, since the pipes would be of smaller diameter, and the energy consumption by pumping would be lower, (lower hydraulic losses). In addition, the slight reduction of thermal efficiency in collectors is usually compensated by the improvement of the thermal stratification in tanks, which guarantees that the inlet temperature to collectors remains controlled, avoiding the situations of high T* already mentioned.

4. Conclusions

In view of the results obtained in the research, the comparison of two solar thermal collectors based solely on the hydraulic configuration of its absorber plate, must be carried out only after having propagated the measurement uncertainties of the test. The results obtained for the tests at different flow rates, according to the SST proposed in the EN12975:2-2006 standard show that the differences are only appreciable in the area of better performance of both collectors. The serpentine type would have better efficiency than the parallel-tubes type until conditions of $T^* \leq 0.05$. From $T^* > 0.06$, it can not be verified that one type of collector works better than another, nor that there is a flow that generates higher efficiency than another.

It can be concluded that in the serpentine collector there is less probability that the convection heat transfer mechanism inside the tubes controls the heat transfer. In addition, the serpentine type collectors guarantee no hydraulic imbalance, while in the case of parallel-tubes collector, the imbalances cause hot points, changing the properties of the fluid and even vaporizing the fluid. It would be advisable to project the lowest possible flow that allows to reduce the initial costs and pumping energy, besides guaranteeing that the collector works with low inlet temperatures, thanks to the good stratification achieved in tanks, but having as a limitation to protect the system from overheating conditions . This flow can always be lower in the case of the serpentine type collector, which is more suitable for working in low-flow conditions. The parallel-tubes collector needs a higher flow rate to avoid poor flow distributions, which depends on its design and construction. This does not greatly affect the costs, since their hydraulic losses are lower than in the serpentine type, and therefore, the pump costs would be equivalent in both cases if the minimum recommended flow is selected.

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