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Experimental studies coupled by Computational Fluid Dynamic (CFD) findings for the of Heat Transfer enhancement in Flat Plate Solar Water Collectors

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Abstract : *The aim of this experimental study is to improve the thermal performance of passive flat plate solar collectors using a novel cost effective enhanced heat transfer technique. The project work focuses on the process of energy conversion from the collector to the working fluid. This is accomplished by employing an aluminium grid placed in the channels of a collector to induce a gradient of heat capacitance. This novel technique is tested experimentally using two unglazed collectors. One collector has the aluminium net inserted in its channels and it is tested against an identical conventional collector in order to have a direct comparison at the same time. The results are coupled by the findings of simplistic Computational Fluid Dynamics (CFD) designs, obtained in previous work. The experimental data obtained and the CFD findings show a good agreement.*
Keywords: *Flat Plate Solar Water Collectors, Heat Transfer Enhancement, Metallic Mesh, CFD*

I. Introduction

Solar energy can be utilised as a form of heat, such as solar water heating and as electricity, such as solar photovoltaic. Solar water heating systems are commonly referred to in the industry as Solar Domestic Hot Water (SDHW) systems and it is a technology that is not entirely new. The solar collector is the key element in solar energy systems. It absorbs the solar radiation and converts it into a useable form of energy that can be applied to meet a specific demand. During the last 60 years a lot of research has been conducted in order to increase the efficiency of the Solar Water Collectors by employing numerous experimental studies.

One such technique was to apply absorbing coatings on the top surface of the collector. Solar radiation is commonly divided into various regions or bands on the basis of wavelength [1]. Spectrally selective surfaces for solar absorbers have been studied since the 1950s, when the idea of using wavelength separation was introduced by Tabor [2]. Spectrally selective solar surfaces are used for the conversion of solar radiation into thermal energy. Since then numerous studies have been conducted using different types of absorber coatings such as Zr–ZrO_2 cermet solar coatings, electrochemically deposited thin films of black chromium or a thin layer of black nickel is applied by electrolysis onto the surfaces [3, 4]. Selective coatings enable the collector to operate better in weak sunlight. The temperature that drops over the night, or the heat loss after cloudy spells, are relatively small [5]. This is a result of the coating's high absorbptance and low emittance.

Other work was focused on the glazing (top cover) of the solar collectors since glass is a good material for flat plate solar collectors as it transmits almost 90% of the received shortwave solar radiation. Types of plastics could also be used as covers as few of them can endure ultraviolet radiation for a long time. Antireflective coatings on glass sheets were also a possibility for increasing the efficiency of solar energy systems by reducing the reflection of the incoming light. Studies have shown that the transmittance of glass can be increased by 4% if the glass is equipped with antireflection surfaces and also in return the efficiency of the solar collector can also be increased [6]. Recently, a silica low-reflection coating via a dip-coating process has been developed. The value of the film refractive index that leads to a minimum of reflection on the surface of the glass cover was achieved [7].

The above described techniques were solely focused on the surface and glazing of the solar collectors without altering its customary shape. Though flat plate solar collectors used in modern domestic hot water systems have not changed significantly in the past twenty years, a number of designs have been investigated in order to reduce heat losses and improve their efficiency. These designs were focused on altering the usual shape of the collectors.

A reverse flat plate solar collector had an inverted absorber plate with a stationary concentrating reflector beneath [8]. A bifacial collector incorporated two stationary concentrators with a flat plate absorber mounted above them [9]. These designs achieved higher efficiencies than other flat plate collectors under low irradiation conditions. A rectangular built in storage water system was investigated but this shaped systems have a low solar gain and poor heat transfer during winter in northern latitudes [10]. A triangular built-in-storage solar water heater, had also been studied under winter conditions that resulted to a higher solar gain and enhanced natural convection, leading to a higher water temperature [11]. An inverted absorber 'Integrated Collector Storage Solar Water Heater' ICSSWH mounted in the tertiary cavity of a compound parabolic concentrator with a secondary cylindrical reflector using several types of transparent baffles at different locations within the collector cavity was experimentally investigated [12]. The results of these research methods were encouraging. The drawback was that were bulky, expensive and hard to mount on the rooftops. Therefore as mention before the aim of this research was focused on how to increase the heat transfer within the pipes of the collector.

Previous research showed that the efficiency of solar collectors is related to the flow distribution through the parallel riser tubes. The sensitivity of the flow distribution to the collector design parameters focusing on junction pressure losses and continuous energy losses due to friction [13]. An experimental study was conducted in a water solar flat plate collector with laminar flow conditions to analyse the flow distribution through the collector and showed that it depends on the relation between energy loss in the risers and the energy losses in the manifolds [14]. The flow distribution through the collector's finned tubes clearly affects the operational efficiency of the collector system as it decreases due to the uniformity of flow being reduced [15, 16]. Therefore, the more uniform the flow is through the tubes, the higher is the efficiency of the collector [17, 18]. Heat transfer enhancement methods can be: passive requiring no direct application of external power or active which require external power. One passive method is widely used to enhance heat transfer between the working liquid and the metal part is the application of a metal porous medium placed in channels of heat exchangers [19, 20]. The presence of a metal porous medium (stainless steel, aluminium, etc.) inside a pipe causes a better thermal dispersion and also increases the interface between the fluid and absorber. The overall thermal conductivity in this case is considerably higher than that of the water. Disadvantage of using porous medium is the rise in the hydraulic resistance [21]. Hence the research method employed in this experimental setup would use partially filled pipes with aluminium mesh to avoid pressure drops and keep the cost of the design low.

II. Experimental Setup

The experimental work involved tests conducted on a rig equipped with two types of unglazed solar collector panels and artificial insolation. The first type of the collector used was similar to a conventional design and the second had aluminium mesh inserted in all its pipes as shown in Fig. 1. Each panel consisted of 4 sections which were in fact individual tubes with attached fins and these were 1.35 m long. All the pipes had an external radius of 10 mm and the fins were 110 mm wide. The two central sections of the panel had copper pipes with aluminium alloy fins. The sections which were placed on each side of the panel had pipes and fins fully made of copper. The absorption side of the panel was matt black painted.

Fig. 1. Aluminium mesh inserted in copper pipe.

Plastic water tanks, having a capacity of 15 litres were drilled on either side along its length and a copper pipe was inserted. The pipe was positioned to act as a heat exchanger between the hot water flowing inside the copper pipe and the cold water stored inside the water tank. A metal extension was built at the top part of the metal frame and a wooden platform was attached to it in order to accommodate the water tanks. Thermocouples attached permanently on the surface of the collectors would measure and record its temperature accurately and instantaneously. The top manifolds were linked to the inlet of the water tank and from its outlet was directed to the bottom manifold of the panel as shown below in Fig.2. The flow rate of the fluid in the panels was also needed to be determined since it would give more information concerning the behaviour of the fluid flow in each of the collectors. To reduce the overall heat losses, the water tanks and the copper pipes at its inlet were insulated as shown in Fig. 3.

Fig. 2. Water tanks connected to the collectors.

Fig. 3. Insulated water tanks and artificial insolation

The final experimental setup's schematic diagram is presented in Fig. 4. below:

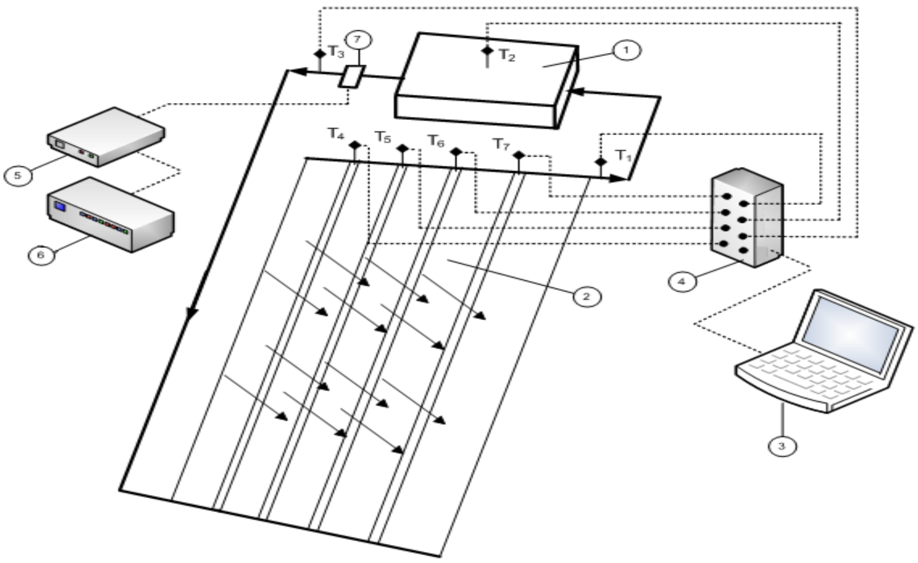

Fig. 4. Schematic of the experimental setup.

where: 1-Water Tank; 2-Flat plate collector; 3-Laptop; 4-Data logger; 5-Power supply; 6-Frequency meter; 7-Flow sensor /Pressure transducer. T1, T2, T3, are thermocouples that measure the temperatures at the inlet, in the water tank and at the outlet respectively. T4 to T7 measure the temperatures at the exit of each finned pipe. All thermocouples were connected to a data logger and the temperatures were monitored via a laptop. The pressure in both collectors was measured when the fluid was at rest and also in motion until it reached steady state. A pressure transducer was connected at the top part of the collector The pressure transducer was connected to a power supply set to 18 V and 0.4 mA. An 817 Ω resistor was connected in series with the transducer in order to measure the voltage drop across it. This would relate the recorded voltage drop to pressure, using an accurate multi-metre. The transducer was calibrated and an initial voltage of 3.30 V represented the pressure inside the system at rest before the experiment started. From the manufacturer's specifications and by using the resistor of 817 Ω it was calculated that an increase of 0.523 V represented 1 bar of pressure in the system. As the experiment progressed instant recording of pressure was visible till steady state was reached. At steady state in the conventional collector, a voltage of 3.8 V was recorded that represented a pressure of 0.95 of a bar (gauge pressure) above the atmospheric pressure, that corresponds to a temperature of 119^{o}C. On the other hand a voltage of 3.96V in the collector with the metal mesh was recorded which represented a pressure of 1.26 bar (gauge pressure) above the atmospheric pressure, that corresponds to a temperature of 125^{o}C. This justified the higher temperatures occurred in the collector having a metal mesh as higher pressure results to higher temperature in a closed system. The mass flow rate in both systems was determined by incorporating a low-flow positive displacement flow meter . This had been proven to be a reliable and highly accurate volumetric method of measuring flow, as it has a high accuracy over a wide range of viscosities and flow rates. Prior to the installation of the flow meter to the system it was calibrated by connecting it to the water mains and via a measuring tube, a power supply and a digital frequency meter.

The mass flow rate was measured for various frequencies. It was important to obtain an accurate mass flow rate since it would prove, into a certain extent, that could affect the output temperature of both collectors.

For a heat flux of 905 W/m^2, the flow rate on the convectional collector was measured 0.62ml/sec and for the one with the metal mesh 0.58 ml/sec. The difference between the two flow rates was not significant. It was calculated that the flow rate of 0.58 ml/sec would only increase the temperature of the conventional collector by 1.7^{o}C. Therefore this would justify that the flow rate is not the main reason for the higher temperature at the collector with the metallic mesh within its pipes.

A series of K-type thermocouples were attached onto the surface of each of the panels so more accurate and instantaneous temperature readings could be recorded. The thermocouples were placed as pairs in each of the finned pipes as shown in Fig. 5 below:

Fig. 5. Surface temperature measurements.

All thermocouples were positioned half way along the length of each of the panels. One thermocouple was placed on the fin to measure the fin temperature and the second at the joint between the pipe and the fin to measure the so called base temperature Tb important factor for further analysis of the system i.e. the heat transfer coefficient *h*. Measurements showed that the surface temperature of the collector utilising porous medium was lower to the conventional collector proving that there is a better heat transfers. Two cases were examined for two different heat fluxes 610 W/m2 and 1070 W/m2 which covered the upper and lower limit of heat fluxes used in the experiments. The data recorded at the point where the temperatures reached a steady state. From the recorded data the local Rayleigh number for each case was calculated from:

$$Ra = \frac{g\beta \Delta T D^3}{\nu\alpha} \quad (1)$$

where g is the gravitational force, β is the thermal expansion coefficient, ΔT is the wall and ambient temperature, D is the diameter of the pipe, ν the kinematic viscosity and α is the thermal diffusivity. Table 1 displays the obtained *Ra* number for both cases under investigation and for two different heat fluxes.

Table 1. Rayleigh number for different Heat Flux for the examined cases.

Heat Flux (W/ m²)	***(Ra)* Porous Case**	***(Ra)* Conventional Case**
1070	$3.4\cdot10^{7}$	$3.3\cdot10^{7}$
610	$1.2\cdot10^{7}$	$1.1\cdot10^{7}$

The results showed that the convective forces are dominant in both geometries as they are well in the range of the free convection limits. A higher *Ra* number was obtained for the case with porous medium.

The difference between the two findings was not great since for the characteristic length the diameter *D* of the pipe was used, which meant it was measured at a slice (locally) of the pipe but still it was higher. The heat transfer coefficient was also obtained from the expression:

$$h = \frac{Q}{\left(T_b - T_{fref}\right)} \quad (2)$$

where Q is the heat flux in W/m2, T_b is the temperature at the joint of the fin and pipe and T_{fref} is the fluid reference temperature in Celsius. Table 1 displays the values obtained for heat transfer coefficient *h* in both cases under investigation and for two different heat fluxes. Results showed that there was a higher heat transfer coefficient in the collector with porous medium, supporting further the investigation so far.

Table 1. Heat transfer coefficient *(h)* of the examined cases.

Heat Flux (W/ m²)	***h* Porous Case (W/m² C)**	***h* Conventional Case (W/m² C)**
1070	375	334
610	196	179

From the findings a relation could be made between the experimental results and the results obtained previously by the CFD findings [22]. Hence the average *Ra* and *Nu* number for the porous cases could be obtained over a characteristic pipe length and then compared to the findings of Table 2 below:

Table 2. Heat transfer coefficients (*h*) of the CFD examined cases.

Heat Flux (W/ m²)	***h* Porous Case (W/m² C)**	***h* Conventional Case (W/m² C)**
1070	207.67	194.83
610	115.60	99.00

and the expressions below :

$$Nu_{(1070)} = 0.013(Ra)^{0.285} \quad (4) \qquad Nu_{(610)} = 0.007(Ra)^{0.285} \quad (5)$$

The average *Ra* number could be obtained from the expression :

$$Ra = \frac{C_p \rho^2 g\beta(\Delta T)L^3}{\kappa\mu} \quad (6)$$

where *Cp* is the specific heat of the water, is the density of the water, *g* is the gravitational force at (55o inclination), β is the thermal expansion coefficient, ΔT is the temperature difference between the outer pipe surface and the water temperature at the pipe wall, L is the characteristic length of the pipe, κ is the thermal conductivity and μ the dynamic viscosity.

The average *Ra* number was obtained for the heat fluxes of 1070 W/m² and 610 W/m² as exploited on the CFD simulation cases [22]. The data used to obtain the Rayleigh numbers is listed on Table 3. The average temperatures of 152°C and 121°C generated by 1070 W/m² and 610 W/m² respectively

Table 3. Data used to obtain the Rayleigh numbers.

	1070	**610 W/m²**
C_p *(J/Kg°C)*	4316	4246
ρ *(Kg/m³)*	915	943
g (m/s²)	5.62	5.62
β*(1/°C)*	$1.03\cdot10^{-3}$	$0.88\cdot10^{-3}$
ΔT *(°C)*	3.0	3.2
L(m)	0.25	0.25
κ(W/m² °C)	$0.682\cdot10^{-3}$	$0.683\cdot10^{-3}$
μ*(Kg/m·sec)*	$0.18\cdot10^{-3}$	$0.23\cdot10^{-3}$

The average *Ra* numbers obtained for the heat flux of 1070 W/m2 and 610 W/m2 was 7.95•10^9 and 6.14 •10^9 respectively, showing a good agreement to the simulations findings which were 8•10^9 and 6.2 •10^9 for the heat flux of 1070 W/m2 and 610 W/m2 respectively. The average Nu numbers could be determined in relation to *Ra* numbers using the expressions (4) and (5). The resulting *Nu* numbers calculated was 8.61 and 4.32 for the heat flux of 1070 W/m^2 and 610 W/m^2 respectively showing a good agreement to the simulations results which were 8.4 and 4.3.

The experiment started having a heat flux of 490 W/m2, changing approximately every forty seven minutes to 610 W/m2 then rose to 750 W/m2 and finally to 905 W/m2. The halogen bulbs were then switched off in order to observe the response of both collectors under no heat flux conditions. The experiment terminated when the temperatures on both collectors decreased considerably. The variation of the average temperatures of the water at the exit of the finned copper pipes in each of the two collectors under investigation is illustrated on the graph of Fig. 6 below:

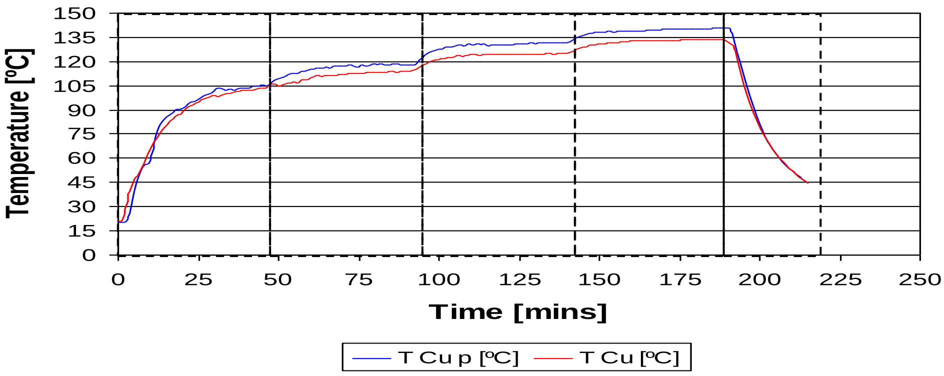

Fig. 6. Temperature vs. time at the exit of the copper finned pipes.

A temperature of 7°C higher was attained for the porous medium copper finned pipes (T $_{Cup}$) compared to the conventional (T $_{Cu}$). For the first twenty minutes the water temperature builds up slower in the pipes with porous medium since much of the heat is drawn by the aluminium mesh at the start of the experiment. The heat is then past to liquid from the aluminium net and the water temperature increases above the conventional finned copper pipes.

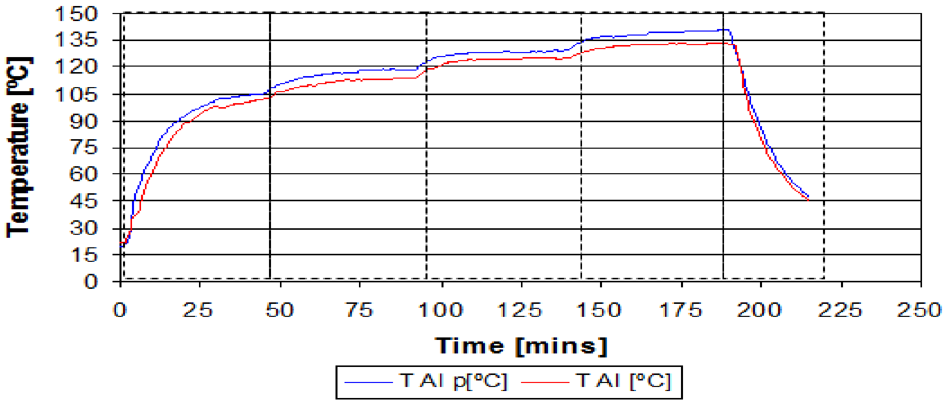

Fig. 7. Temperature vs. time at the exit of the aluminium finned pipes.

At discharge when there is no heat flux, and for about twenty minutes, it can be seen that the copper pipes with porous medium can hold up more heat compared to the conventional until both system reach almost the same temperature the last fifteen minutes of the experiment.

A similar pattern was obtained of the water temperature at the outlet of the finned aluminium pipes in each of the two collectors. A temperature of 7.5 °C higher was obtained for the porous medium aluminium finned pipes compared to the conventional. This is illustrated in Fig. 7 above.

Another temperature reading was taken of the water at the inlet of the water tanks of the porous medium (T input wt p) and the conventional (T input wt) .

The results once again showed a better performance of the collector that utilised the aluminium porous material. This can be seen in Fig. 8 below. All piping and compression fittings from the collector's outlet to the water tank's inlet was insulated in order to minimise heat loses.

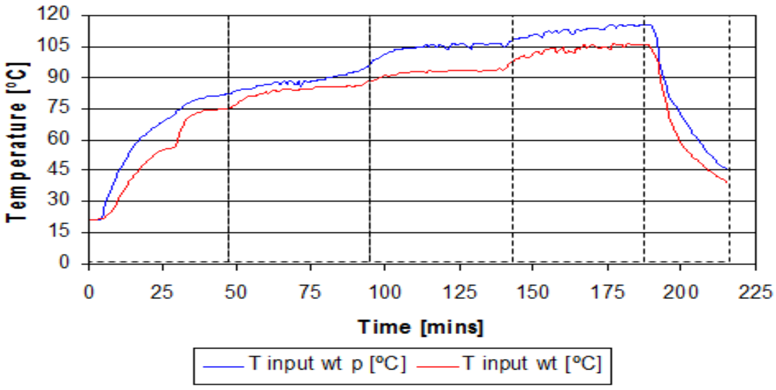

Fig. 8. Temperature vs. time at the inlet of the water tanks.

III. Results and Discussion

The experiment proved that there was a better heat transfer using porous medium, as the temperature difference between the two collectors at the water tank inlet for two different heat fluxes i.e. 750 W/m2 and 905 W/m^2 on average was 10.5^{o}C higher for the collector utilised porous medium.

There were also periods during the experiments that the temperature difference between the two graphs reached to 16^{o}C as it is obvious from the graphs presented above. Hence it was concluded that it was beneficial having porous inserted in the channels of the collector but also having in all the piping of the system.

Probably it could be advantageous to utilise porous medium from the collector all the way to the domestic appliances since it could provide a better heat transfer.

The pipes though need to be very well insulated. Both tanks were filled up with 8 litres of water in order to cool down the water flowing inside the collectors.

The temperatures recorded two hours into the experiment as it was when temperature started to increase inside the water tanks and it was then noticeable the temperature difference. During experiments it was observed that at about 180 minutes of operation the temperature inside the water tank with the porous medium was about 36^{o}C and for the conventional was 31^{o}C resulting to a temperature difference of 5^{o}C.

From this point onwards and when there was no heat flux, could also be seen that the temperature inside the water tank with porous medium remains higher for a significant amount of time, maintaining a relatively constant temperature difference of about 5.5^{o}C higher compared to the conventional one.

This can be clearly observed in Fig. 9 below:

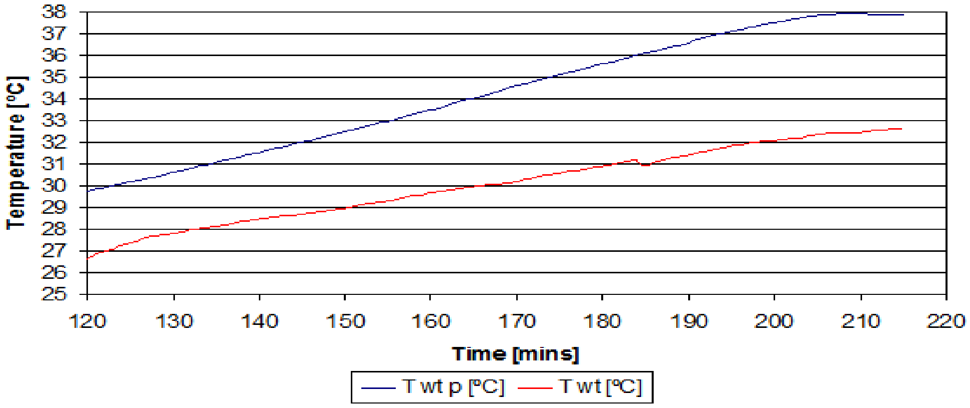

Fig. 9. Temperature versus time inside the water tanks.

IV. Conclusions

Research has been conducted with the aim of enhancing the heat transfer in a passive flat plate solar water collector using a cost effective technique that could be easily applied in a typical (conventional) flat plate collector without changing or redesigning its shape. The experimental results showed that the metallic mesh inserted in the collector, provided a higher water temperature compared to the conventional collector and it is the presence of the aluminium mesh inside the channels that distributes heat more evenly. It was apparent that using partially filled porous mesh in the pipes the heat transfer area increased. The roughness of the aluminium net created a thermal diffusion at the boundary layer that could make the collector more efficient. The experimental results are supporting the theoretical findings obtained during CFD simulations. Computational

Fluid Dynamics (CFD) modeling and experimental work demonstrated an increase in the water temperature in the solar collector with the aluminum mesh insertion by $7^{o}C$ in comparison to the conventional one. Although a simplified CFD model was used, it was directly compared to the experimental setup, since the findings in both cases were cross-examined over the same characteristic length (0.25m). Both numerical and experimental tests were in good agreement that the aluminium metal insertion considerably increased the output water temperature in the collector. Such technique would allow the reduction of the solar collector area and its associated manufacturing costs. The presence of the aluminium mesh inside the channels increases the heat transfer and also changes the flow pattern in such a way which increases heat transfer from fluid present in the near-wall zone to the internal layers of the water. Parameters such as Rayleigh (*Ra*) and Nusselt (*Nu*) numbers obtained and the results showed that existence of a laminar flow that had a dominant convective heat transfer mechanism in the case of the collector with the metal insertion. The average *Ra* numbers calculated using experimental data, for the heat flux of 1070 W/m^2 and 610 W/m^2 was $7.95 \bullet 10^9$ and $6.14 \bullet 10^9$ respectively, showing a good agreement to the simulations findings which were $8 \bullet 10^9$ and $6.2 \bullet 10^9$ and for the heat flux of 1070 W/m^2 and 610 W/m^2 respectively. The resulting Nu numbers obtained experimentally was 8.6 and 4.3 for the heat flux of 1070 W/m^2 and 610 W/m^2 respectively showing a good agreement to the CFD results which were 8.3 and 4.2 and to the ones calculated which were 8.4 and 4.3 respectively. The presence of the aluminium mesh insertion enhanced the convective heat transfer in the conventional collectors on average by 9.3% (CFD findings) and by 10 % (experimental investigations). Such technique provided a higher temperature at the outlet of the solar collector that increased by 9.2%. The average *Ra* and *Nu* numbers obtained for the heat flux of 1070 W/m2 and 610 W/m2 were higher in the collector that utilised the aluminium mesh, compared to the conventional and in both cases they showed a good agreement to the simulations findings, determined from the expressions (4) and (5). These correlations can be applied on any flat plate collector with an aluminium grid in its pipes, in order to predict its performance.

References

Journal Papers:

[1]. L F. Caslake, D J. Connolly, V Menon, CM. Duncanson, R Rojas and J Tavakoli, Disinfection of Contaminated Water by Using Solar Irradiation, Appl. Environ Microbiol. Volume 70, 2004, 1145–1150.

[2]. Amnon Einav, Solar Energy Research and Development Achievements in Israel and Their Practical Significance Sol. Energy Eng. Volume 126, 2004, 92,.

[3]. S Suzer, F. Kadirgan, H.M. Sohmen, A.J. Werherilt, I.E. Ture, Spectroscopic characterization of Al_2O_3-Ni selective absorbers for solar collectors, Solar Energy Materials and Solar Cells, Vol. 52, 1998, 55-60.

[4]. F. Kadirgan, M. Sohmen, Development of Black Cobalt Selective Absorber on Copper for Solar Collectors, Turk J Chem Vol.23, 1999, 345-351.

[5]. S. Furbo and L. Jivan Shah, Thermal advantages for solar heating systems with a glass cover with antireflection surfaces, Vol. 74, 2003, 513-523.

[6]. K. Maatouk, M. Shigenao, K. Atsuki, B. Masud, Flat-Plate Solar Collector Performance with Coated and Uncoated Glass Cover, Heat Transfer Engineering, Vol. 27, 2006, 46-53.

[7]. V. Kienzlen, J.M. Gordon and J.F Kreider, The reverse flat plate collector: A stationary, non-evacuated, low-technology, medium-temperature solar collector, J. Solar Energy Engineering, Vol. 110, 1988, 23-30.

[8]. A. Goetzberger, J. Dengler, M. Rommel, and V. Wittwer, The bifacial absorber collector: a new highly efficient flat plate collector, International Solar Energy Society, USA, 1991, 1212-1217.

[9]. S.C. Kaushik, R. Kumar, H.P. Garg, J. Prakash, Transient analysis of a triangular built-in-storage solar water heater under winter conditions. Heat Recovery System CHP Vol. 14, 1994, 337–341.

[10]. M. Smyth, P. McGarrigle, P.C. Eames, B. Norton, Experimental comparison of alternative convection suppression arrangements for concentrating integral collector storage solar water heaters, Solar Energy, Volume 78, 2005, 223-233.

[11]. R. A. Bajura and E. H. Jones. Flow distributions in manifolds. J. Fluids Eng. Trans. ASME Volume 98 1976, 654-655.

[12]. V. Weitbrecht, D. Lehmann and A. Richter, Flow distribution in solar collectors with laminar flow conditions Solar Energy Volume 73, 2002, 433–441.

[13]. Chiou, J.P. The effect of non-uniform fluid flow distribution on the thermal performance of solar collector. Solar Energy Vol. 29 1982, 487–502.

[14]. Jianhua Fan, Louise Jivan Shah and Simon Furbo. Flow distribution in a solar collector panel with horizontally inclined absorber strips. Solar Energy, 2005, ISES. Volume 52, 1994.

[15]. Solar Thermal Technology. Sandia National Labs:http://solstice.crest.org/renewable

[16]. Baytas A.C and Pop I. Free convection in a square porous cavity using a thermal non-equilibrium model. International Journal of Thermal Sciences, Vol. 41, 2000.

[17]. Poulikakos D and Kazmierczak M. Forced convection in a duct partially filled with porous material. ASME Journal of Heat Transfer, Vol. 109, 1996.

[18]. G. Iordanou, G. Tsirigotis. Computational Fluid Dynamics (CFD) Investigations of the Effect of Placing a Metallic Mesh in the Channels of a Passive Solar Collector Model, Electronics and Electrical Engineering. – Kaunas: Technologija,. Vol. 5, 2012, 69–74.

Books:

[19]. O.P Agnihotri., B.K Gupta, (Solar Selective Surfaces, Wiley Series, New Delhi, 1981).

[20]. Nield, A.D. and Bejan, A. (Convection in Porous Media, Springer Verlag, N.Y., 1999).

Proceedings Papers:

[21]. I. Tanishita, Recent development of solar water heaters in Japan, Proceedings of the UN Conference on New Sources of. Energy, Rome Volume 5, 1964, 102.