HEFAT2010
7th International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics
19-21 July 2010
Antalya, Turkey

SIMULATION OF HEAT TRANSFER ENHANCEMENT IN A FLAT PLATE SOLAR PANEL WITH WIRE COIL INSERTS USING TRNSYS

Herrero Martín R.*, García Pinar A., Pérez García J., López Galiana E., García Soto F. J. and García Cascales J. R.
Department of Thermal Engineering and Fluids,
Technical University of Cartagena,
Cartagena, 30202,
Spain,
E-mail: ruth.herrero@upct.es

ABSTRACT

Enhanced surfaces can be used to increase heat exchange, reduce the size of equipments or save pumping power. Thermal solar collectors are potential candidates for enhanced heat transfer, but not many studies have focused on this aspect. However, enhancement techniques can be applied to thermal solar collectors towards more compact and efficient designs. This work presents the study of heat transfer enhancement in a tube on sheet solar panel with wire coil inserts, using TRNSYS as the simulating tool. For comparison purposes, a standard collector has been also simulated. Both collectors have been tested under the same ambient, radiant and testing conditions. The corresponding standardized efficiency curves are provided. The optimum operating range has also been established ($750<Re_o<2250$).

INTRODUCTION

In industrial applications, a set of enhancement techniques are widely used to improve the performance of heat exchangers. Enhancement techniques can be classified into two groupings [1]: "passive" and "active" techniques, depending on whether they need or not external power. In most cases, the commercially available enhancement techniques are currently passive.

Tubeside enhancement techniques can be classified according to the following criteria: (1) additional devices which are incorporated into a plain round tube (twisted tapes, wire coils) and (2) non-plain round tube techniques such as surface modification of a plain tube (corrugated and dimpled tubes) or manufacturing of special tube geometries (internally finned tubes). Wire coils are a type of inserted elements which present some advantages compared to other enhancement techniques, such as artificial roughness by mechanical deformation. They may be installed in an existing smooth tube heat exchanger. They keep the mechanical strength of the smooth tube. Their installation is easy and their cost is very low. Enhanced surfaces can be used to increase heat exchange, reduce the size of equipments or save pumping power. Thermal solar collectors are potential candidates for enhanced heat transfer, but not many studies have focused on this aspect.

NOMENCLATURE

a_1	[W/m²K]	Coefficient of thermal losses for $(T_m-T_a)=0$
a_2	[W/m²K]	Coefficient of thermal losses depending on temperature
A_A	[m²]	Absorber plate area
Bi	[-]	Biot number
c	[-]	Contact conductance parameter between fin and tube
Cp	[J/kgK]	Specific heat of the heat transfer fluid
C	[-]	Parameter of the Klein´s loss equation
D_i	[m]	Internal tube diameter
e	[-]	Parameter of the Klein´s loss equation
f	[-]	Internal to external thermal resistance ratio
F	[-]	Fin efficiency factor
F'	[-]	Local collector efficiency factor (for a dl element)
F_R	[-]	Global collector efficiency factor
g	[m]	Contact zone between fin and tube
g_{earth}	[m/s²]	Earth´s gravity
G	[W/m²]	Hemispheric solar irradiance
Gr	[-]	Grashof number
Gz	[-]	Graetz number
h_w	[W/m²K]	Heat transfer coefficient due to wind flow over the collector
h_f	[W/m²K]	Fluid heat transfer coefficient
k_{abs}	[W/mK]	Absorber thermal conductivity
k_{ins}	[W/mK]	Insulation thermal conductivity
k_f	[W/mK]	Fluid thermal conductivity
k_{tube}	[W/mK]	Tube thermal conductivity
k_{gd}	[W/m²K]	Heat transfer coefficient between contact zone and tube
k_{gf}	[W/m²K]	Heat transfer coefficient between fluid and contact zone
$\dot{m}$	[kg/s]	Mass flow rate
Nu	[-]	Nusselt number
N_G	[-]	Number of collector covers
N_{tubes}	[-]	Number of tubes
Pr	[-]	Prandtl number
$\dot{Q}$	[W]	Collector thermal power
Ra	[-]	Rayleigh number
Re	[-]	Reynolds number
S_∞	[-]	Shape factor
t_m	[ºC]	Average temperature of the heat transfer fluid
t_a	[ºC]	Ambient temperature
t_o	[ºC]	Output solar collector temperature
t_{in}	[ºC]	Input solar collector temperature
T^*_m	[m²K/W]	Reduced temperature difference
$\bar{T}_{abs}$	[K]	Absorber temperature

u	[m/s]	Circulating air velocity
U_L	[W/m^{2}K]	Collector overall heat loss coefficient
U_T	[W/m^{2}K]	Cover heat loss coefficient
U_{be}	[W/m^{2}K]	Back and lateral heat losses coefficient
v	[m/s]	Velocity of the heat transfer fluid
w	[m]	Fin width
x^*	[-]	Dimensionless length
X_p	[m]	Characteristic length to evaluate local Nusselt number
Special characters		
α_{abs}	[-]	Absorptance of absorber plate
β	[°]	Inclination angle with respect to the horizontal plane
β_r	[-]	Thermal dilatation coefficient
δ	[m]	Plate thickness
δ_{ins}	[m]	Insulation thickness
δ_{tube}	[m]	Tube wall thickness
ε_{abs}	[-]	Absorber emittance
ε_g	[-]	Transparent cover emittance
η_A	[-]	Collector efficiency in terms of T^*_m and absorber area
η_d	[-]	Fin-tube efficiency
η_{oa}	[-]	Optical collector efficiency
μ	[kg·s/m]	Dynamic viscosity
μ_d	[-]	Biot-type parameter
ρ	[kg/m^3]	Density
σ	[W/m^{2}K^4]	Stephen -Boltzmann constant
τ_g	[-]	Transparent cover transmittance
υ	[m^2/s]	Fluid kinematic viscosity

The vast majority of studies carried out deal with heat transfer enhancement in air solar collectors, mainly inserting artificial roughness within the exchange surfaces. The first work carried out was developed in 1988 by Prasad and Saini.[2]

Other remarkable works were developed by Cortes and Piacentini [3] who developed a steady-state mathematical model for a bare collector. Saini and Saini [4] carried out an experimental investigation for fully developed turbulent flow in a rectangular duct with large aspect ratio and having expanded metal mesh as artificial roughness. Heat transfer enhancement was achieved but joined with very high friction losses.

More recently, Kumar et al [5] carried out an experimental investigation to study the heat transfer and friction characteristics in solar air heater by using discrete W-shaped roughness on one broad wall of solar air heater.

Within enhanced solar water heaters works, Kumar and Prasad [6] presented an interesting study inserting twisted tapes inside the fluid flow tubes in a serpentine collector. They observed that heat losses were reduced (due to the lower value of the plate temperature) consequently an increase on the thermal performance by about 30% over the plane solar water heaters under the same operating conditions was achieved. The effect of twisted-tape geometry, flow Reynolds number (from 3200 to 20000) and intensity of solar radiation on the thermal performance of the solar water heater was presented. They found that the twisted-tape collectors perform remarkably better in the lower range of flow Reynolds number (Re≈12,000), beyond which, the increase in thermal performance was monotonous. An important result achieved was to fix the minimum value of the thermal enhancement factor in 1.3 for the twisted tape insert solar water heaters. They also concluded that such collectors might perform even better at higher values of intensity of solar radiation.

Recently, Jaisankar et al [7] published their experimental investigation of heat transfer, friction factor and thermal performance of twisted tape solar water heater with various twist ratios. They concluded that, heat transfer and pressure drop were higher in twisted tape collector compared to the plain one. Among the various twist ratios, the minimum twist ratio 3 was found to enhance the heat transfer and pressure drop due to swirl generation.

Also Hobbi and Siddiqui [8] conducted an experimental study to investigate the impact of heat enhancement devices on the thermal performance of a flat-plate solar collector. Different passive heat enhancement devices that included twisted strip, coil-spring wire and conical ridges were studied. The comparison showed no appreciable difference in the heat flux to the collector fluid. A detailed investigation of the observed trend showed that the heat transfer mode in the solar collector is of mixed convection type with free convection as the predominant mode. It was concluded that due to the significant damping of shear-produced turbulence by buoyancy forces, the applied passive methods based on the enhancement of shear-produced turbulence are ineffective in augmenting heat transfer to the collector fluid in flat-plate solar collectors. However, it should be mentioned that they were not testing a collector, just a portion of it and all the tests were carried out indoors.

There are a very few works concerning the modelling of enhanced solar collectors. Recently, Varun [9] used a genetic algorithm to optimize thermal performance of a flat plate solar air heater. He considered using different enhancement techniques to increase. This work aims to contribute to this field of work presenting a novel study of heat transfer enhancement in a tube on sheet solar panel with wire coil inserts, using TRNSYS as the simulating tool. For comparison purposes, a standard collector has been also simulated.

COLLECTOR DESCRIPTION

The most relevant data concerning the collector modelled are given in Table 1.

Table 1. Initial data values

Initial data	Value
Material properties	
k_{abs}	209.3 W/mK (aluminium)
k_{ins}	0.05 W/mK
k_{tube}	372.1 W/mK (cupper)
ε_{abs}	0.05 (Miro-Therm)
ε_g	0.88 (glass)
τ_g	0.93 (glass)
α_{abs}	0.95 (Miro-Therm)
Geometrical data	
D_i	0.007 m
w	0.0995 m
g	0.0035 m
β	20°
δ_{abs}	0.0005 m
δ_{tube}	0.5*10^{-3} m
δ_{ins}	0.025 m
N_G	1
X_p	1 m (tube medium point)
N_{tubes}	9
A_A	2 m^2
Measurements	
t_a	20° C
t_{abs}	70° C
u	3 m/s
$\dot{m}$	Total: 80-160 l/h = 0.022-0.044 kg/s Per tube: 2.44*10^{-3}-4.88*10^{-3} kg/s
G	700 W/m^2
T_{in}	20° C
Physical data	

σ	$5.67 \cdot 10^{-8}$ W/m^{2}K^4
k_f	0.6 W/mK (water at 25ºC)
k_{gd}	∞
ρ	996.93 kg/m^3 (water at 25ºC)
μ	8.125 10^{-4} kg/sm
c_f	4182.97 J/kgK
g_{earth}	9.81 m/s^2
β_f	0.000215 K^{-1} (water at 25ºC)

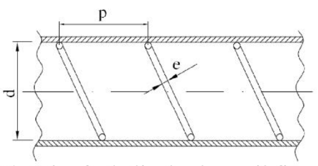

Figure 1 Sketch of a helical-wire-coil fitted in a smooth tube.

Dimensionless pitch p/D=1.65 and dimensionless wire-diameter e/D=0.072 have been chosen for the geometrical definition of the wire coil (Fig.1).

COLLECTOR MODEL

The most representative equations used within this collector model are the following:

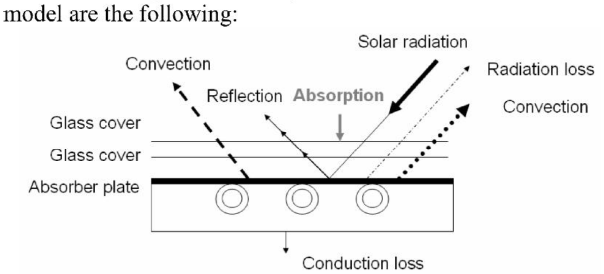

Figure 2 Energy balance in a sheet-and-tube solar collector

Collector overall heat loss coefficient

U_L the collector overall heat loss coefficient [W/m^{2}K] is the sum of top, bottom, and edge loss coefficients.

$$U_L = U_T + U_{BE} \tag{1}$$

U_{BE} [W/m^{2}K] includes the edge loss heat coefficient, term that can be neglected, and the bottom heat loss coefficient (U_B) defined by:

$$U_B = \frac{k_{ins}}{\delta_{ins}} \tag{2}$$

U_T, the top heat loss coefficient, [W/m^{2}K] was calculated employing the empirical equation developed by Klein [10] following the basic procedure of Hottel and Woertz [11]:

$$U_T = \left[\frac{N_G}{\frac{C}{\overline{T}_{abs}} \left[\frac{\left(\overline{T}_{abs} - T_a\right)}{N_G + f_w} \right]^e + \frac{1}{h_w}} \right]^{-1} + \frac{\sigma \cdot \left(\overline{T}_{abs}^2 + T_a^2\right) \cdot \left(\overline{T}_{abs} + T_a\right)}{\left(\frac{1}{\varepsilon_{abs} + 0.00591 \cdot N_G \cdot h_w}\right) + \left(\frac{2 \cdot N_G + f_w - 1 + 0.133 \cdot \varepsilon_{abs}}{\varepsilon_g}\right) - N_G} \tag{3}$$

where:

$$f_w = \left(1 + 0.089 \cdot h_w - 0.1166 \cdot h_w \cdot \varepsilon_{abs}\right) \cdot \left(1 + 0.07866 N_G\right) \tag{4}$$

$$e = 0.43 \cdot \left(1 - \frac{100}{\overline{T}_{abs}}\right) \tag{5}$$

$$C = 520 \cdot \left(1 - 0.000051 \cdot \beta^2\right) \tag{6}$$

The convective heat transfer coefficient h_w [W/m^{2}K] for air flowing over the outside surface of the glass cover depends primarily on the wind velocity u [m/s]:

$$h_w = 5.7 + 3.8 \cdot u \tag{7}$$

Fin and tube efficiency factor

The typical geometry of an absorber of fin-and tube type can be seen in Figure 3a). The absorber is regarded as a fin, where w is the width, and this fin is welded to the tube by an area of width g. The heat transfer through the fin is described by a model that combines those by Duffie and Beckman [12] and by Lund [13]. This model was previously used by Eisenman [14], however weighting factors were applied in order to take into account the constructive differences. (Fig. 3) Furthermore, in the collector modelled, the fin is welded to the tube in a discontinuous way.

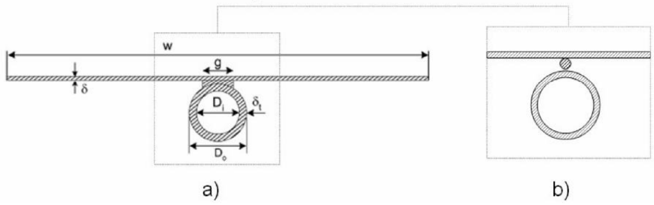

Figure 3 Absorber of the fin-and tube type reported by Eissenman [14] vs detail of the one used in the present work

Neglecting the temperature gradient in flow direction (inside the tubes) and through the plate, the power collected by the absorber at both sides of the contact area between fin-and-tube, per length unit [W/m] can be expressed as:

$$q_{fin} = (w - g) \cdot F \cdot \left[S - U_L \cdot (T_b - T_a)\right] \tag{8}$$

For the fin efficiency the following formula was used:

$$F = \frac{\tanh[M(w - D)/2]}{M(w - D)/2} \tag{9}$$

where

$$M = \left[U_L / (k_{abs} \delta_{abs})\right]^{1/2} \tag{10}$$

The power collected within the contact fin-and-tube surface per length unit [W/m] is given by:

$$q_{contact} = g \cdot \left[S - U_L \cdot (T_b - T_a)\right] \tag{11}$$

The useful power per length unit [W/m] will be the addition of the previous two terms:

$$q = q_{fin} + q_{contact} \tag{12}$$

This useful power [W/m] is transferred to the fluid, as follows:

$$q = w \cdot F' \cdot \left[S - U_L \cdot (T_f(y) - T_a)\right] \tag{13}$$

Where F′ is the collector efficiency factor. This was defined as the ratio of the actual thermal collector power to the power of an ideal collector whose absorber temperature is equal to the fluid temperature. (cf. Duffie and Beckman, 1992)

F′ is constant for any collector geometry and mass flow. Applying the Kirchoff rules to the appropriate thermal network (Fig. 4) the following expression is derived [14]:

$$F' = \frac{1/U_L}{W \cdot \left[\frac{1}{U_L\left[D + (w-D)F\right]} + \frac{1}{C_b} + \frac{1}{\pi D_i h_{fi}} \right]} \tag{14}$$

Within this expression F′, represents the relation between the following thermal resistances: ambient-and-collector and ambient-and-working fluid resistances.

The heat transfer from the absorber to the fluid is described by a series connection of several heat transfer coefficient.

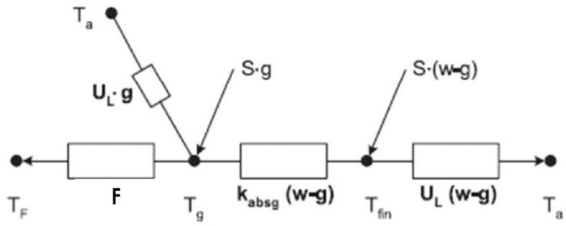

Figure 4. Thermal network of a single fin (adapted from Eisenman et al [14])

Heat removal factor

The useful gain of power may be calculated by:

$$\dot{Q} = \dot{m} \cdot c_f \cdot (T_e - T_{in}) = A_A \cdot F_R \cdot \left[S - U_L \cdot (T_{in} - T_a)\right] \tag{15}$$

In which the heat-removal factor F_R is:

$$F_R = \frac{\dot{m} \cdot c_f}{A_A \cdot U_L} \cdot \left(1 - \exp\left(-\frac{F' \cdot U_L \cdot A_A}{\dot{m} \cdot c_f} \right) \right) \tag{16}$$

The heat-removal factor F_R is defined by the working fluid capacity of removing heat from the fin and the collector. It represents the relationship between the useful actual power gain and the power gained in case the absorber temperature was the inlet temperature of the fluid.

Temperatures

The temperature distribution of fluid along the flow direction of a circular pipe can be obtained from energy balance as [19]:

$$\frac{T_e - (T_a + S/U_L)}{T_{in} - (T_a + S/U_L)} = \exp\left(-\frac{U_L \cdot F' \cdot A_A}{\dot{m} \cdot c_f} \right) \tag{17}$$

From the previous equation the fluid temperature difference can be obtained as follows:

$$T_e - T_{in} = \left(\exp\left(-\frac{U_L \cdot F' \cdot A_A}{\dot{m} \cdot c_f} \right) - 1 \right) \cdot \left(T_{in} - (T_a + S/U_L)\right) \tag{18}$$

Using the heat-removal factor definition, the temperature at the outlet can be expressed as

$$T_e = \frac{A_A \cdot F_R (G \cdot (\tau_g \cdot \alpha_{abs}) - U_L \cdot (T_{in} - T_a))}{\dot{m} \cdot c_f} + T_{in} \tag{19}$$

Finally, the average absorber temperature formula is given by:

$$\overline{T}_{abs} = T_{in} + \frac{\dot{Q}}{A_A \cdot F_R \cdot U_L} \cdot (1 - F_R) \tag{20}$$

Correlations

The corresponding correlations were implemented in the FORTRAN code developed.

Flat plate solar collector:

Re<2300

The fanning friction factor has been estimated with the analytical solution:

$$f = 16/\text{Re} \tag{21}$$

An estimation of the Gr/Re^2 ratio for the flow conditions studied at present work has shown that the heat transfer mode inside collector's tubes is forced convection. Thus, the local Nusselt number in laminar flow for the standard (plain tube) collector has been calculated by means of the Churchill and Ozoe (1973) correlation, in order to account for a simultaneously developing laminar flow with uniform heat flux:

$$Nu_x = 4.36 \cdot \left[1 + (Gz/29.6)^2\right]^{1/6} \cdot \left[1 + \left(\frac{Gz/19.04}{\left[1 + (\Pr/0.0207)^{2/3}\right]^{1/2} \cdot \left[1 + (Gz/29.6)^2\right]^{1/3}} \right)^{3/2} \right]^{1/3} \tag{22}$$

2300<Re<25000

The fanning friction has been obtained with the Petukhov's [15] friction factor correlation:

$$f = (1.58 \cdot \ln(\text{Re}) - 3.28)^{-2} \tag{23}$$

The Nusselt number has been estimated with the Gnielinski [16] equation:

$$Nu = \frac{\left(\frac{f_F}{2}\right) \cdot (\text{Re} - 1000) \cdot \Pr}{1 + 12.7 \cdot \left(\frac{f_F}{2}\right)^{0.5} \cdot \left(\Pr^{2/3} - 1\right)} \tag{24}$$

Flat plate solar collector with wire coil inserts:

For the enhanced collector with wire coil inserts, the experimental data of García et al. [17] have been used to estimate de Nusselt number and friction factor inside collector's tubes.

200<Re<1200 [17]

$$f = 14.8/\text{Re}^{0.95} \quad (\text{Re}<400) \tag{25}$$

$$f \approx 5 \cdot 10^{-2} \quad (400<\text{Re}<1200) \tag{26}$$

$$Nu = 0.100 \cdot \text{Re}^{0.99} \tag{27}$$

1200<Re<25000 [17]

$$f = 9.35 \cdot (p/e)^{-1.16} \mathrm{Re}^{-0.217} \quad (28)$$

$$Nu = 0.122 \cdot (p/d)^{-0.354} \mathrm{Re}^{0.724} \mathrm{Pr}^{0.370} \quad (29)$$

Calculation Procedure:

The procedure for calculating the theoretical value of the useful power per area unit q´´[W/m^2], will now be described.

The initial available data are summarized in Figure 5 and the calculation procedure schema is showed in Figure 6.

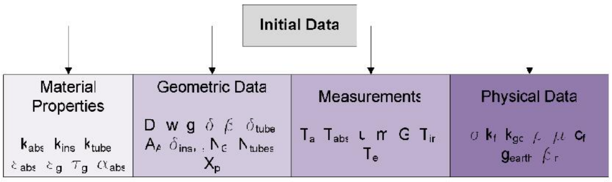

Figure 5. Initial data.

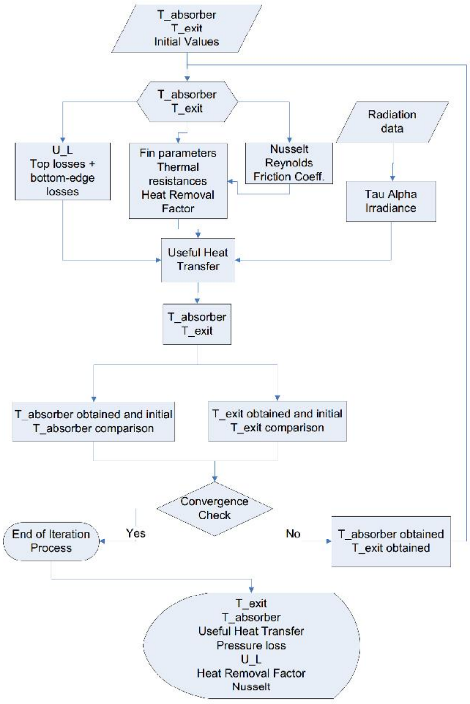

Figure 6. Calculation procedure.

According to the useful power per area unit definition:

$$q'' = F_R \cdot (G \cdot (\tau_g \cdot \alpha_{abs}) - U_L \cdot (T_{in} - T_a)) \quad (30)$$

It is necessary to first estimate U_L and F_R. To compute U_L, the previous parameters to determine are C, h_w, e, f_w and U_{BE}. Nevertheless, to compute F_R is necessary to follow an iterative process, due to the fact that F_R depends on F´.

TRNSYS MODEL

The major aim of this work is to study the heat transfer enhancement in a tube on sheet solar panel with wire coil inserts. Due to this fact, the main component to model in detail is the collector.

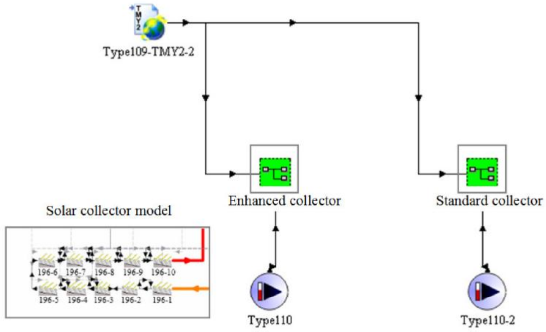

Figure 7 Simplified TRNSYS Model

The approach chosen was to integrate both collectors (the standard and the enhanced one) in the same model and to divide the collectors´ tubes of 2 m length in 10 parts each of 0.2 m, to study accurately the variables evolution along the tubes. The simplified model used in TRNSYS is shown in Figure 7.

The solar collector model shows the aforementioned division. Furthermore, the meteorological condition simulator is shown (Type 109) and two pumps are also included to fix mass flows and inlet temperatures.

Validation of TRNSYS Model

In order to validate the TRNSYS model implemented, the efficiency data obtained by TRNSYS and standardized experimental data given by a Spanish National Certification Agency (CENER) for the implemented collector were used. In Fig. 8 the efficiency comparison is given. All the presented results have been obtained according to the procedures fixed by the Standard UNE 12975-2.

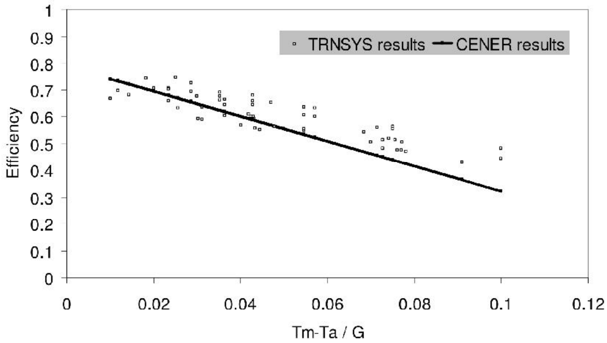

Figure 8 Validation of TRNSYS Model

It can be observed that for the lowest $(T_m\text{-}T_a)/G$ values, the advisable working area, the best results are obtained. However, for higher values in abscissas, around 0.075 the data given by TRNSYS are higher than the experimental ones, predicting higher efficiency values. The predicted errors reach 25% for the aforementioned higher values.

PROCESSING OF RESULTS

R1 Criterion

According to the results obtained in the simulations, classical performance evaluation criteria were employed to assess the real enhancement provided by the wire coil inserts.

This criterion allows computing heat transfer enhancement taking into account the increase in pumping power due to the wire coils inserted.

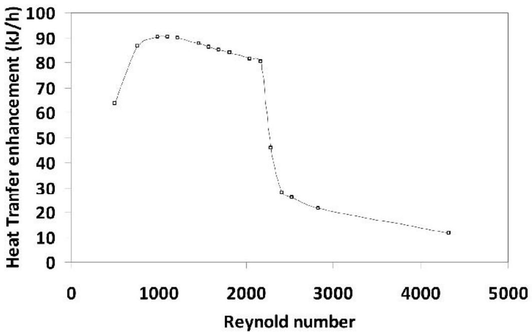

Figure 9. Heat transfer enhancement as a function of equivalent Reynolds number for a smooth tube.

The optimum area is located from Reo≈750 to Reo≈2250, where the heat transfer enhancement presents maximum values. (Fig. 9)

Efficiency Results

Once the TRNSYS model was validated for the highest efficiency values, the efficiency curves of the enhanced flat plate solar panel is plotted in comparison with the flat solar panel. (Fig. 10)

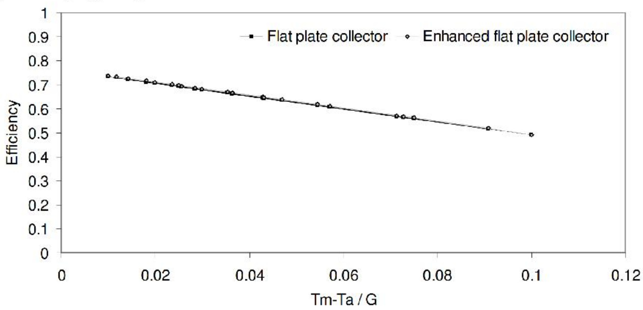

Figure 10. Efficiency comparison between the enhanced solar panel and the normal solar panel

The efficiency curves are very close, however the enhanced flat plate solar collector values are 1.5% higher that the obtained for the normal collector.

CONCLUSION

A new code was successfully implemented in TRNSYS to simulate accurately enhanced solar collectors taking into account the heat transfer enhancement and friction losses.

The improvement achieved in the enhanced solar collector in comparison with the standard one was computed using two different approaches: directly comparing the efficiency curve obtained and employing the well-known performance evaluation criterion (R1) commonly used in industrial heat exchangers. In the first case, the improvement obtained was between 1-2%. Regarding the second method used, the optimum operating range for the flat plate solar panel with wire coils was obtained. These values correspond to Reynolds numbers between 750-2250.

REFERENCES

[1] Webb, R. L., and Kim, N-H, Principles of Enhanced Heat Transfer, 2005, 2nd ed., Taylor & Francis, New York.

[2] Prasad, B.N., and Saini J.S., Effect of artificial roughness on heat transfer and friction factor in a solar air heater, *Solar Energy*, Vol. 41, Issue 6, 1988, pp 555-560.

[3] Cortés, A., Piacentini, R., Improvement of the efficiency of a bare solar collector by means of turbulence promoters, *Applied Energy*, Vol. 36, Issue 4, 1990, pp 253-261

[4] Saini, R. P., Saini, J. S., Heat transfer and friction factor correlations for artificially roughened ducts with expanded metal mesh as roughness element, International Journal of Heat and Mass Transfer, Vol. 40, Issue 4, 1997, pp 973-986.

[5] Kumar, A., Bhagoria, J.L., R.M. Sarviya, Heat transfer and friction correlations for artificially roughened solar air heater duct with discrete W-shaped ribs *Energy Conversion and Management*, Vol. 50, Issue 8, 2009, pp 2106-2117.

[6] Kumar, A., Prasad, B. N., Investigation of twisted tape inserted solar water heaters-heat transfer, friction factor and thermal performance results, *Renewable Energy*, Vol. 19, 2000, pp 379-398.

[7] Jaisankar, S., Radhakrishnan, T.K., Sheeba K.N., Experimental studies on heat transfer and friction factor characteristics of forced circulation solar water heater system fitted with helical twisted tapes, Solar Energy, Vol. 83, Issue 11, 2009, pp 1943-1952.

[8] Hobbi, A., Siddiqui K., Experimental study on the effect of heat transfer enhancement devices in flat-plate solar collectors, *International Journal of Heat and Mass Transfer*, Vol. 52, Issues 19-20, 2009, pp 4650-4658.

[9] Varun, S., Thermal performance optimization of a flat plate solar air heater using genetic algorithm, *Applied Energy*, In Press, Corrected Proof, Available online 20 November 2009.

[10] Klein S.A., Calculation of flat plate loss coefficients, *Solar Energy* 17, 1975, pp. 17:79.

[11] Hottel HC, Woetz BB. *Transaction of the American Society of Mechanical Engineering* 1942, Vol. 64, pp 91.

[12] Duffie, J.A., Beckman, W.A., Solar Engineering of thermal processes, 2nd ed. Wiley Interscience, 1991, New York, pp 268-272.

[13] Lund, K.O., General Thermal analysis of serpentine-flow flat-plate solar collector absorbers. *Solar Energy* 42, 1989, 133-142.

[14] Eisenmann W., Vajen K., Ackermann H.. On the correlations between collector efficiency factor and material content of parallel flow flat-plate solar collectors. *Solar Energy* 76, 2004, pp. 381-387.

[15] Petukhov, B.S. Heat transfer and friction in turbulent pipe flow with variable physical properties. In: *Advances in Heat Transfer* 6 Academic Press, New York ,1970, pp. 503–564.

[16] Gnielinski,V., New Equations for Heat and Mass Transfer in Turbulent Pipe and Channel Flow, *International Chemical Engineering*, 16, 1976, pp.359-368.

[17] García A., Vicente P. G., Viedma A., Experimental study of heat transfer enhancement with wire coil inserts in laminar-transition-turbulent regimes at different Prandtl numbers, *International Journal of Heat and Mass Transfer*, 48, 2005, Issues 21-22, pp. 4640-4651

ACKNOWLEDGEMENTS

The authors want to thank the *Seneca Foundation* from Murcia (Spain) for the funding provided to develop the project: *Design, construction and characterization of an enhanced heat transfer collector.*