EVALUATING PERFORMANCE FROM SPIRAL POLYETHYLENE TUBES AS SOLAR COLLECTORS FOR HEATING SWIMMING POOLS

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Abstract. *The solar energy is very common in the daily of citizens from different regions in world. Environmental questions and the consequent Development of renewable energy techniques were a decisive factor for expanding this market. Currently, the solar energy is present in many different devices: as direct conversion through photovoltaic panels as in solar domestic for hot water systems(SDHWS). Another common use is in the heating system for swimming pools, that could be utilized for therapeutic or comfort reasons. The main aspect that increments this use is the economy for operation of these systems. On the other hand, these systems need a high initial investment. Reducing this cost without reduction in collector efficiency using new materials and / or alternative projects is important target for new researches. Thus, this paper aims to analyze the efficiency of one of these alternative models for heating swimming pools. The conceptual device evaluated is a low cost model. It could be made from polyethylene tubes forming spiral heat exchangers. Analysis of the system is based on a dynamic model using differential equations system including solar collector and swimming pool. Experimental radiation and other environmental conditions in the region of Bauru-SP are used for analyse the dynamic behavior of the system. The simulations are based on analysis of three main parameters: number of collectors, the pump drive time and wall thickness of the collector of polyethylene. Based on these numerical tests one can conclude that this new model of solar collector for swimming pool has a better cost benefit ratio when superficial area is equal to 80% of pool area, pump operation is alternating with four minutes turned on and 28 turned off and the polyethylene wall thickness is 1.5 mm*

Keywords: solar collector; swimming pool solar heater; evaporation losses

1. INTRODUCTION

A major characteristic of the modern society is the increasing demand for energy use. Investments on scientific, political, technological and financial resources have demonstrated that some fields of energy application converted from renewable resources can result in economic advantages for users. In these resources that have become economically and technologically advantageous one can cite some applications of solar energy. The warm water pool, used for therapeutic or comfort reasons, is one such solar energy applications that can be best utilized. This kind of system presents as economically viable and with better efficiency when compared to other systems that use conventional energy. Due to these characteristics, Ruiz and Martinez (2009) observed that other traditional systems for auxiliary heating swimming pool water have been abandoned.

Heating the swimming pool is necessary due to the short time its use along the year, since it is impractical in colder periods. The swimming season opened extends basically over the summer, but in this period, some days still are cold and even on hot days the early morning hours are usually cold. The use of conventional water heaters are disadvantageous due to increased energy costs. Therefore, Hahne and Kübler (1994) proposed a project to use solar energy for the swimming season, extending its use throughout the day. To obtain a satisfactory efficiency in solar heating for swimming pool, Almanza and Lara (1994) note that meteorological parameters are a major factor for a successful device. These results added to those observed in various parts of the world like by Jirka and Watanabe (1980a) and Jirka and Watanabe (1980b) corroborate to emphasize their conclusions.

The solar heater system has the major advantage of using clean, renewable energy, which does not occur in most other systems with power, oil, gas and electricity. Studies carried out by Smith, Jones and Lof (1994) show that there are over 5.9 million swimming pools in the U.S. that consumes billions of dollars. Solar energy can enter in this market as a viable alternative for eliminating this consumption energy.

However, the use of renewable energy requires a high initial investment although the cost of subsequent maintenance and operation are significantly lower. To reduce this cost and maintain the efficiency of the collector, it has increasing importance the selection of materials and methods that reduce the cost of the system. Thus, this study aims to evaluate the efficiency of a new altenative for solar collector model for swimming pools than the traditional model. This new model is made of polyethylene tubes forming a spiral geometry as shown in Fig. 1. For this analysis it is proposed a numerical model developed a GNU Octave code which were implemented two ordinary differential equations to evaluate the dynamic behavior of the system. These equations were obtained from the energy balance in each one of the

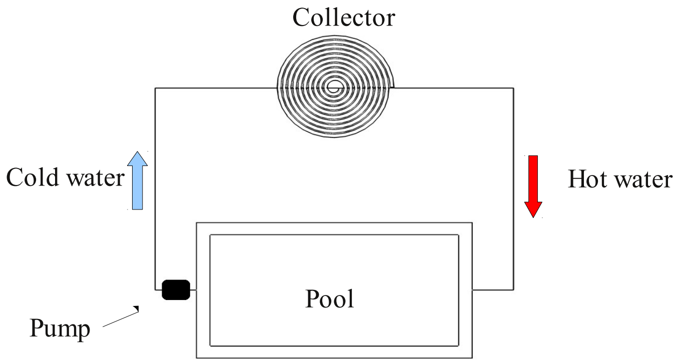

Figure 1. Solar collector and swimming poll scheme

components in question: the solar collector and the pool. The mass flow in the collector and in the pool are the same as a function of time. It is necessary to obtain the solution of system of ordinary differential equations in a coupled way. In this work it was considered a 18 m^2 surface area pool and volume of 25.2 m^3, which is located in the city of Bauru in southeast Brazil. For this study used the pool has only a simple filtering system and has no conventional heating auxiliary systems.

2. FORMULATION

Considering the solar radiation and the forms of heat loss to the environment, one can obtain the equations relating to the solar collector as shown in Eq. (1), and to the pool, Eq. (4). The expressions required to calculate the rate of heat loss associated with each phenomenon, can be seen in Eqs. (2), (3) e (5).

$$\dot{m}_{pe} \cdot c_{pe} \cdot \frac{dT_{suc}}{dt} = Q_{sol} - Q_{col} - Q_{lossc} \tag{1}$$

where

$$Q_{col} = \dot{m}_w \cdot c_w \cdot (T_{aoc} - T_p) \tag{2}$$

and

$$Q_{lossc} = U_P \cdot A_{suc} \cdot (T_{suc} - T_{amb}) \tag{3}$$

$$V_p \cdot \rho_w \cdot c_w \cdot \frac{dT_p}{dt} = Q_{col} + Q_{sol} - Q_{lossp} \tag{4}$$

where

$$Q_{lossp} = Q_{eva} + Q_{rad} + Q_{conv} \tag{5}$$

According to Eq. (2), it can be observed that the rate of useful heat transfer (Q_{col}) is dependent from average temperature output of the collector (T_{aoc}). This is achieved by determining the film coefficient inside the tube collector (U_i) and the logarithmic mean temperature difference, Eq. (6), from the surface temperature of the solar collector and the inlet temperature of pool water. Regarding Eq. (3), heat losses related to the solar collector will be detailed by calculating the heat loss coefficient (U_p) in Section 2.1.

$$T_{aoc} = \exp\left(- \frac{\pi \cdot D_i \cdot L \cdot U_i}{\dot{m}_w \cdot c_w} \right) \cdot (T_{suc} - T_p) + T_{suc} \tag{6}$$

As noted in Eq. (5), heat losses are the sum of the terms by evaporation, radiation and convection, since the rates of heat transfer by conduction to the soil have to be considered negligible (Section 2.2.). These losses are compensated by direct solar radiation incident (Q_{sol}) on the pool surface and useful heat (Q_{col}) from the solar heating system.

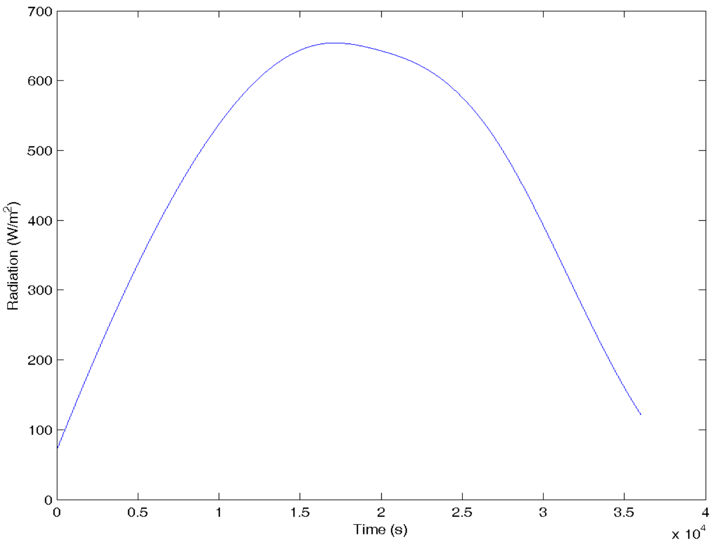

Figure 2. Solar radiation intensity during the day

The curve illustrated in Fig. 2 is obtained from experimental data of solar radiation for each hour of the day period. These data were obtained on 01/09/2009, in the period from 7h30min to 17h30min. Approximation at any time throughout the day based on the experimental points could be obtained by a spline. In the next sections is presented the expressions used for determine the heat losses.

2.1. Methodology applied to the solar collector

The determination of the heat transfer coefficients can be obtained through the heat losses coefficient (U_P) and heat transfer coefficient inside the tubes of solar collector (U_i). The methods used are based on different procedures described in the literature and can be observed throughout this section.

2.1.1. Heat transfer coefficient inside the tubes of solar collector (U_i)

The heat transfer coefficient inside the tube of collector is a factor that depends crucially on the mass flow of the flow. In pumped systems, the determination of the flow is simple and constant during the day. However, for systems using thermosyphon the calculation is more complex and the internal flow on collector depends from the mass rate and solar intensity. Anyway, the Reynolds number is critical for determining the flow regime inside the tube. Therefore, it is considered laminar if $\mathrm{Re} \leq 2200$ and the transitional or turbulent when the Reynolds value that exceeds this limit.

In case of internal flow in circular ducts is used the expression that directly relates the Reynolds number with the mass flow, according to Eq. (7):

$$\mathrm{Re} = \frac{4 \cdot \dot{m}}{\pi \cdot D_h \cdot \mu} \tag{7}$$

Expression for laminar flow

As mentioned before the laminar flow is characterized with $\mathrm{Re} \leq 2200$. Thus, Duffie and Beckman (1991) proposed an expression that meets this condition and expresses the values of average Nusselt number according to Eq (8):

$$\overline{\mathrm{Nu}} = \mathrm{Nu}_\infty + \frac{a \cdot (\mathrm{Re} \cdot \mathrm{Pr} \cdot D_h / L)^m}{1 + b \cdot (\mathrm{Re} \cdot \mathrm{Pr} \cdot D_h / L)^n} \tag{8}$$

where

$$\mathrm{Nu}_{\infty}=3.7 \ \mathrm{e} \left| \begin{array}{ccccc} \mathrm{Pr} & a & b & m & n \\ 0.7 & 0.0791 & 0.0331 & 1.15 & 0.82 \\ 5 & 0.0534 & 0.0335 & 1.15 & 0.82 \\ 10 & 0.0461 & 0.0316 & 1.15 & 0.84 \end{array} \right| \tag{9}$$

Expression for the transition and turbulent flow

For flows that have a behavior of the transition or turbulent flow is considered situations where Re > 2200. Thus, Duffie and Beckman (1991) proposes the use of the expression of Pethukov according to Eq. (10):

$$\bar{\mathrm{Nu}}=\frac{(f/8)\mathrm{Re}\cdot\mathrm{Pr}}{1.07+12.7\cdot\sqrt{f/8}\cdot(\mathrm{Pr}^{2/3}-1)}\cdot\left(\frac{\mu}{\mu_w}\right)^{0.11} \tag{10}$$

The calculation of friction factor (f) inside the tubes can be removed directly from the Moody chart or can be calculated for smooth pipes according to Eq. (11):

$$f=\left(0.79\ln(\mathrm{Re})-1.64\right)^{-2} \tag{11}$$

2.1.2. Determination of the heat losses coefficient (U_P)

According Scalon (2008) and Duffie & Beckman (1991), the heat losses coefficient in the solar collectors can be determined through certain procedures. These procedures depend on the knowledge several environmental factors and components of the solar flat-plate collector. Figure 3 shows the solar collector plan and the thermal resistance from the top, since the losses from the bottom of the solar collector were discarded. Thus, the heat losses coefficient can be calculated by Eq. (12).

$$U_P=\frac{1}{A_{suc}\cdot R_{eq}}=\frac{1}{R_{eq}''}=\frac{1}{(R''^{-1}_{conv,n-c}+R''^{-1}_{rad,s-c})^{-1}}=\frac{1}{(h_{conv,n-c}+h_{rad,s-c})^{-1}} \tag{12}$$

Through the thermal circuit shown in Fig. 3 may observe two thermal resistances considered in the upper solar collector:

1. Radiation thermal resistance
2. Convection thermal resistance

The procedure for this resistances calculations is described in the subsequent sections.

Radiation resistance between the sky and the solar collector

In this case, due to the fact that heat transfer occurs by radiation, it is used the definition of the heat transfer coefficient between two arbitrary surfaces, as shown in Eq. (13), where it is considered $T_s=T_{amb}$ e $\varepsilon_{col}=0.1$.

$$h_{rad,s-c}=\varepsilon_{col}\cdot\sigma\cdot\left(T^2_{suc}+T^2_s\right)\cdot\left(T_{suc}+T_s\right) \tag{13}$$

Thus, the thermal resistance can be expressed by Eq. (14).

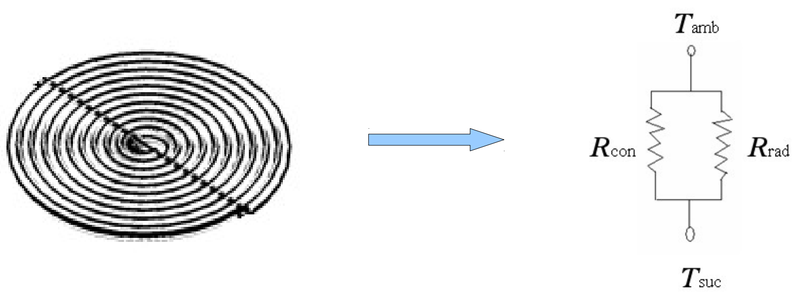

Figure 3. Heat resistance losses through the top of solar collector

$$R''_{rad,s-c}=\frac{1}{h_{rad,s-c}} \tag{14}$$

Convection resistance between the neighborhoods and the solar collector

In the case of calculating the thermal resistance of convection, as the collector usually gets exposed to a wide open area, wind speed represents a crucial role in this analysis, thus proposing the use of calculating the heat transfer coefficient inside the tubes represented by Eq. (15), according to Duffie and Beckman (1991).

$$\bar{Nu}_F=1.72\cdot Re^{1/2}\cdot Pr^{1/3} \quad (\text{valid for } 10^4<Re<4.5\times10^4) \tag{15}$$

If the wind intensity is negligible, convection should be approached as external natural convection and thus suggest the use of Eq. (16) e Eq. (17).

$$\bar{Nu}_N=0.59\cdot Ra^{1/4} \quad (\text{valid for } 10^4<Ra<10^9) \tag{16}$$

$$Nu_N=0.13\cdot Ra^{1/3} \quad (\text{valid for } 10^9<Ra<10^{12}) \tag{17}$$

where

$$L_{suc}=\frac{A_{suc}}{P_{suc}} \tag{18}$$

The adequate convection model can be obtained by estimating the Richardson number ($Ri = Gr / Re^2$). Thus, if $Ri > 10$, predominate the effects of natural convection, and if $Ri < 0.1$, predominate the effects of forced convection. If there is a combination of effects, $0.1 < Ri < 10$ is used, usually, the expression of Equation (19) to perform the combination:

$$\frac{\bar{h}_{conv,n-c}\cdot L_{suc}}{k}=\bar{Nu}_c=(\bar{Nu}_N^n\pm\bar{Nu}_F^n)^{(1/n)} \text{ , where in this case } n=7/2 \tag{19}$$

Thus, the convection thermal resistance, in any case, can be expressed by Eq. (20):

$$R''_{conv,n-c}=\frac{1}{\bar{h}_{conv,n-c}} \tag{20}$$

2.2. Methodology applied to pool

Due to the mixing effect caused by the filtering system, the pool is modeled as a lumped system and thus the temperature variation of the pool can be determined from Eq. (4). According to this analysis the total heat loss of the pool is the sum of the losses by evaporation, convection and radiation. However, these losses are compensated by a gain due to direct solar irradiation. The methodology used to evaluate each of these losses is presented in subsequent subsections.

2.2.1. Evaporative losses

According to the literature, there are several semi-empirical correlations that describe the behavior of the evaporative losses applied in pools located outdoors. These correlations include the difference in vapor pressure between the surface of the pool water and air environment, as well as an empirical function of the coefficient of heat transfer by evaporation as a function of wind speed. Thus,

$$Q_{eva}=A_p\cdot h_{eva}\cdot(P_{v,sat}-P_{v,amb}) \tag{21}$$

and according the correlation presented by ASHRAE (2003),

$$h_{eva}=a+b\cdot w^n \tag{22}$$

where the coefficients *a*, *b* and *n* are, respectively, 0.089 W/m^2.Pa, 0.0782 W.s/m^3.Pa and 1.

2.2.2. Radiative losses

Because the exchange of infrared radiation between the sky and the pool surface, the radiation heat losses can be calculated according to Eq. (23).

$$Q_{rad} = A_p \cdot \epsilon_w \cdot \sigma \cdot (T_p^4 - T_s^4) \tag{23}$$

The term referring to the temperature of the sky, shown in Eq. (23), will be approximated by the value obtained from the local temperature. The emissivity referring to the water pool is considered 0.95 (Ruiz and Martinez, 2009).

2.2.3. Convective losses

For the calculation of convection heat losses is used the following equation:

$$Q_{conv} = A_p \cdot h_{conv} \cdot (T_p - T_{amb}) \tag{24}$$

The convection heat transfer coefficient is a factor dependent on wind speed, as shown in Eq. (25). This empirical correlation is obtained from a solar heating system for pools and established by the Australian standard (Australian Standard, 1989). Thus, the wind speed for the proposed problem is considered *w = 2.5 m/s*.

$$h_{conv} = 3.1 + 4.1 \cdot w \tag{25}$$

2.2.4. Solar radiation incident

The direct solar radiation incident and absorbed through the surface of the pool is determined according to Eq. (26):

$$Q_{sol} = A_p \cdot \alpha_p \cdot G \tag{26}$$

and the absorptance is a factor relating to pool in question. In this case one can consider the value of $\alpha_p = 0.85$ (ISO/TC, 1995).

3. RESULTS AND DISCUSSION

Based on the proposed formulation, simulations were performed for conditions previously determined by evaluating the behavior of the system collector and pool. The parameters used for temperature and solar radiation were based on data obtained from the Southeast of Brazil during 10 hours throughout the day, as shown in Fig. 3. The simulations were done using the software GNU Octave with the parameters previously described. The results were based on three main points: response time for heating the pool through the collecting area, the electric power consumption with the activation of the pump and the collector efficiency as function of wall thickness. The mass flow recirculated by the pump was assumed to be constant throughout the day and equal to 3.5 l/min.

The first hypothesis establishes a relationship between the response time for heating the pool water and the collecting area of the solar heater. There are two different simulations whose results are shown in Figs. 4 e 5.

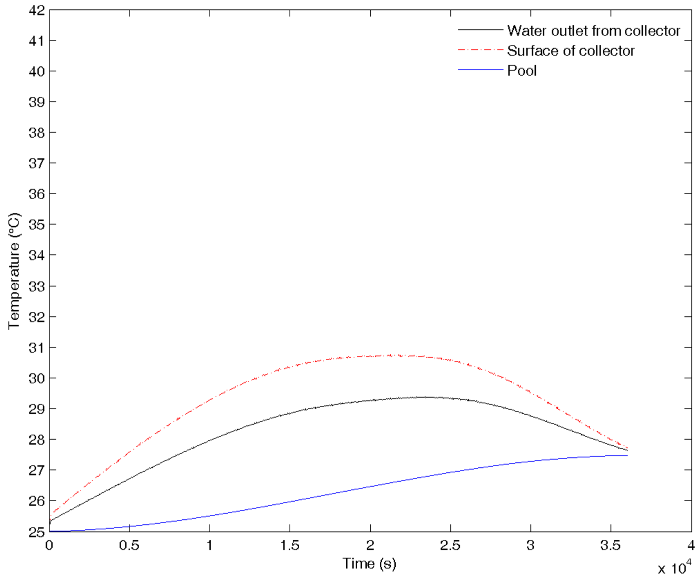

Figure 4. Temperature profile versus time profile for a pickup of 100 m in length and wall thickness of 2.5 mm.

The Fig 4 shows three curves for a collector of 100 meters, 2.5 mm wall thickness and the pump turned on throughout the day. One can observe that the average temperature of the water outlet of the collector has similar behavior with the curve of the surface temperature of the collector. Considering the heating time for the pool water of 5.5 hours, an increase of 25°C to 26.46°C is observed on the model. Although a small increase in temperature throughout the day has been achieved, one can conclude that it is insufficient to permit use of the pool on days unsuitable.

Because of the small solar energy capitation in the previous case, it was increased the area of solar panels to make them usable. Figure 5 show the results for the same parameters of Fig. 4, but with a collecting area at 80% of the surface pool area (8 collectors). Thus, one can observe the increase of temperature of pool water 25°C to 26.84°C at the same instant of time of 5.5 hours described for the previous case. This fact confirms that the response time of pool heating is directly related to the number of collectors. In both cases, either to one collector, either for eight collectors, you can heat the pool water, but the response time for heating will be relatively faster for second device.

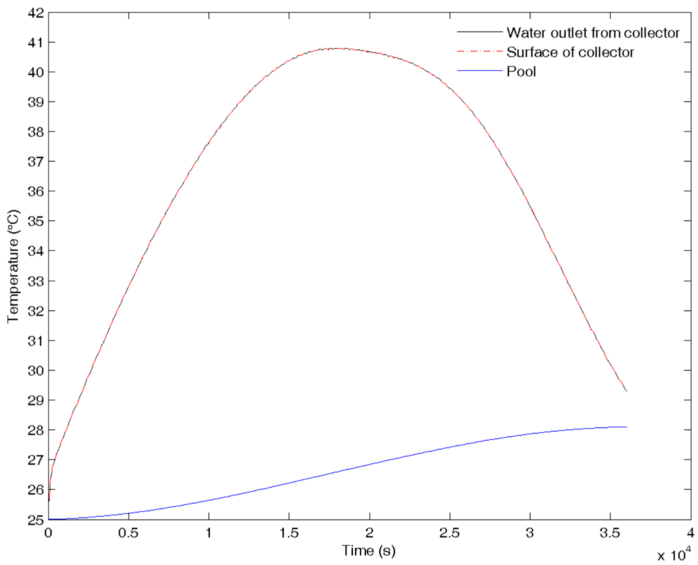

Figure 5. Temperature profile versus time profile for a pickup for eight collectors of 100 m in length and wall thickness of 2.5 mm.

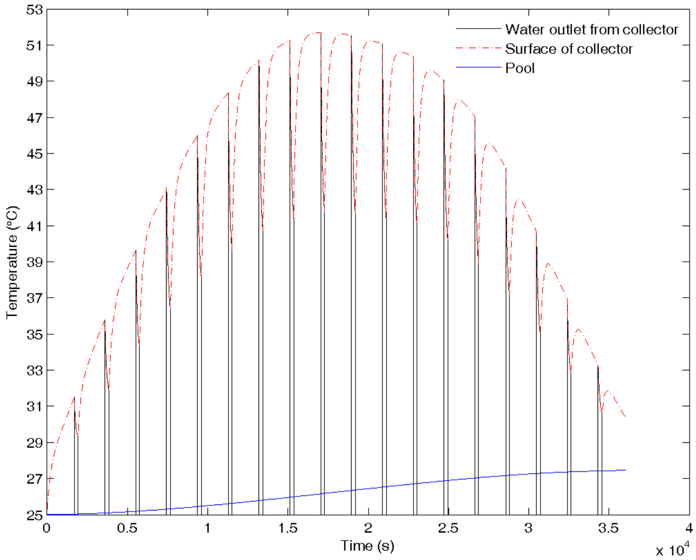

Figure 6. Temperature profile versus time for eight collectors of 100 m in length, with wall thickness of 2.5 mm and pump with 4 minutes turned on and 28 turned off

The second situation shows that the efficiency of the collector associated with the pump operation time. Through this simulations one can see that it is possible to reduce the electric power consumption by varying the time of pump operation, without significantly harming the heated pool at the end of the day. Figure 2 shows the temperature profiles versus time for eight catchers in which the pump remained on for 10 hours and at the end of the day, the pool temperature reached 28.01°C. In an attempt to reduce the energy consumption due to the period of pump operation, carried out simulations with operation defined by a timer, as shown by Figs. 6 and 7.

Figure 6 shows the behavior of the system with the partial operation of the pump and water circulation on the same conditions of Fig. 5. For this case the pump is in operation during four minutes and on the next 28 minutes it is turned off. Observing the Fig. 6 one can see that for this situation the pool temperature reached to 27.45°C at the end of the day.

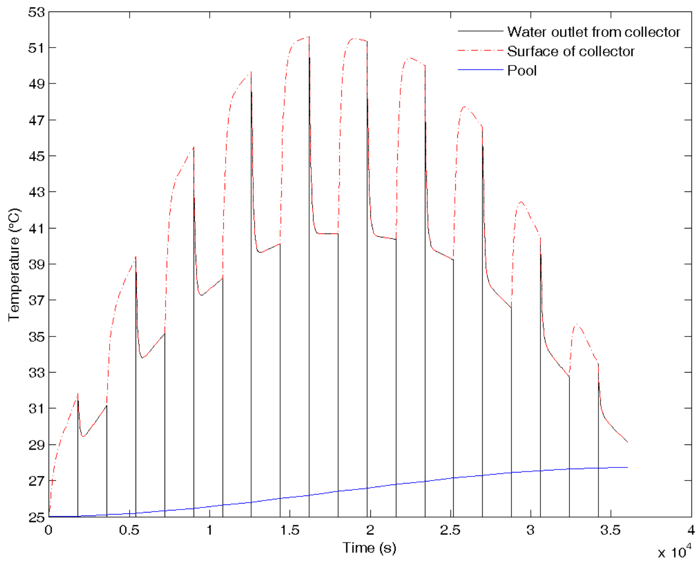

Figure 7. Temperature profile versus time for eight collectors of 100 m in length, with wall thickness of 2.5 mm and pump with 30 minutes turned and a same time turned off

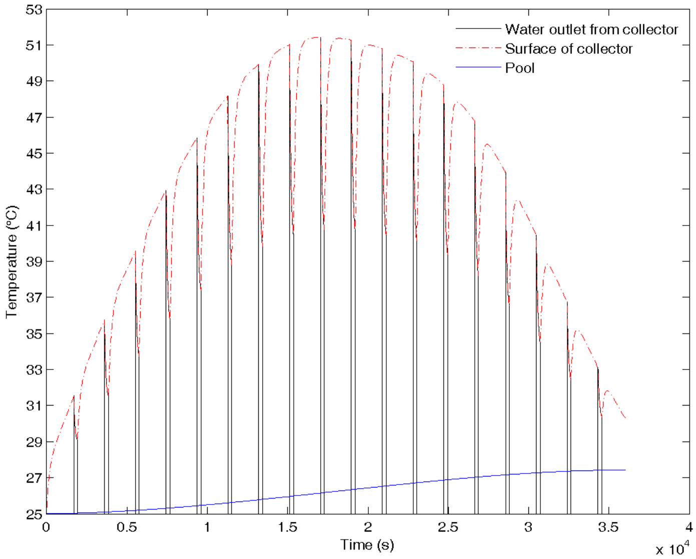

Figure 8. Temperature profile versus time for eight collectors of 100 m in length, with wall thickness of 2 mm and pump with 4 minutes turned on and 28 turned off

Considering the possibility for increasing the water temperature with a longer time pump operation, it was done a simulation with 30 minutes with the pump turned on and other 30 minutes turned off. The dynamic behavior of the system for this case is shown in the Fig. 7. The presented result shows there wasn't significant increase in temperature at the end of the process if compared with the simulation showed on Fig. 6. At the end of the day, the pool temperature has reached to temperature of 27.71°C.

Another condition that could influence the collector effectiveness is the variation of tube wall thickness. All the simulations already done used tubes with 2.5 mm for wall thickness. For comparison proposes, it was done one test with the same parameters of most efficient case (Fig. 6) but with a 2 mm in the tube wall thickness. The results can be seen in the Fig. 8.

The results presented in Fig. 8 show that for this case, the water temperature at the end of the day is 27.44°C. Comparing with the behavior shown in Figure 3, wig. 6 only difference is the wall thickness, one can observe that the temperature variation is negligible. Thus, this behavior indicates that it is possible to reduce the wall thickness of pipe for these conditions without affecting the efficiency of the collector. To confirm this observation one new simulation with wall thickness of 1.5 mm was done and it is shown in Fig. 9.

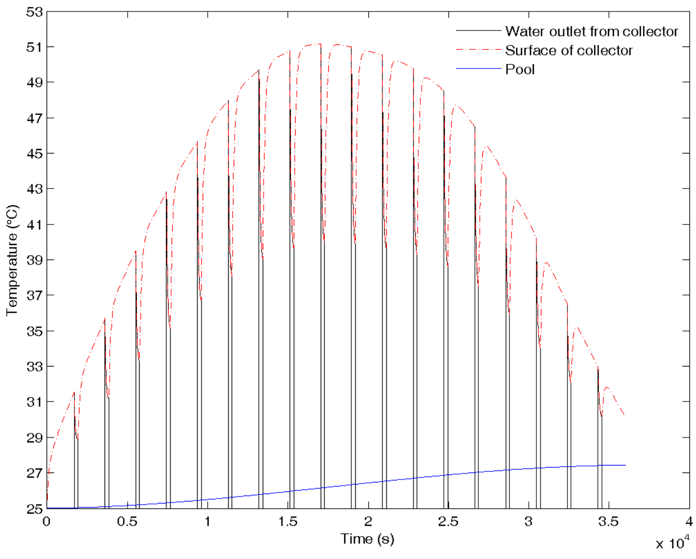

Figure 9. Variação da temperatura em função do tempo para oito coletores com espessura de parede de 1,5 mm e acionamento de bomba com 4 minutos ligada e 28 desligada

Figure 9 shows the result for this case and it can be observed that the temperature profiles are very closed with the other conditions of wall thickness. The pool temperature at the end of the day reached to 27.43°C. This observation confirms that reduction in collector wall thickness cause small changes on pool temperature at the end of the day but, by other way, can significantly reduce the device cost.

4. CONCLUSION

Based on the previous results one can observe that the new model for solar collector used in pools has a better cost benefit ratio in determined conditions. One of these conditions that deserves mention is the increase of surface area collector what changes the time necessary to heat the pool water and the cost of the heater. Considering the values tested in this work, an area equal to 80% of the pool surface is the most adequate.

Another important assessment is evaluate the regime of pump operation. Thus, the best one is needed to achieve energy saving and a satisfactory efficiency for heating pool water. It had been shown that the best situation is the pump operation with four minutes turned on and another 28 minutes turned off. Comparing it to the case of pump integrally linked, one can obtain a saving of 88% in power consumption at the end of the month. Other tests considering the use of thermostats yet to need be realized.

Finally, one can highlight the influence of wall thickness tube in collector cost and product efficiency. According to the simulations one can conclude that the reduction in tube thickness has negligible effect on the pool temperature at the end of the day. Due to this fact, the collector cost could be reduced by about 40% when using a wall thickness of 1.5 mm.

5. ACKNOWLEDGEMENTS

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