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OPTIMIZATION OF EVACUATED TUBE COLLECTOR PARAMETERS FOR SOLAR INDUSTRIAL PROCESS HEAT

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ABSTRACT: *Evacuated tube solar collectors (ETC) are increasingly in use worldwide because of their high thermal efficiency and high working temperature compared to the flat plate solar collectors. The efficiency of ETC is substantially enhanced due to the presence of vacuum between the absorber and the cover of evacuated tube solar collector (ETC). This is mainly attributed to the reduction in heat losses by convection ad conduction. The high energy absorption increases the values of solar fraction and instantaneous efficiency. The objective of this paper is to investigate thermal performance of evacuated tube solar water collector in hot and harsh climate like Kuwait climate. An experimental rig facility was first set up to monitor the thermal performance of ETC through one year period. The experimental data along with correlations obtained by linear regression are presented. A detailed developed nonlinear model for evacuated tube solar collectors is presented in the current work with more comprehensive optical and thermal analysis. The variation of the temperature along both the circumferential (fin) and the longitudinal (tube) directions is considered in the present model. The model analyzes separately the optics and the heat transfer in the evacuated tubes allowing the analysis to be extended to different configurations. The predicted numerical values are found to agree well with the experimental values obtained from experimental test facility. The optimum design parameters; collector tube size, mass flow rate and collector tilt angle based on year around ETC thermal performance is determined under the meteorological conditions of Kuwait. The maximum energy generation from the collector corresponds to tilt equal to 25° (i.e. latitude - 5°) and for collector facing south (azimuth angle=0°). The results indicate that the optimum tube length is 1.5 m as at this length a significant enhancement can be achieved in thermal efficiency for other tube diameters studied. The optimal mass flow rate is 30 kg/h.m^2 as thermal collector efficiency reaches its highest maximum value of 0.53 at this optimum mass flow rate. Finally, integration of evacuated tube collector in industrial process as soft drink industry can provide 90% of the heat required with life cycle savings of approximately $18,000.*

KEYWORDS: Evacuated Tube Collector, Collector Efficiency, Optimum Parameters, Solar Fraction, Solar Heat for Industrial Processes

INTRODUCTION

Solar collectors are the major component in solar thermal systems, with flat plate and evacuated solar tube collectors the most common ones. Flat plate collectors operate efficiently at low temperatures which limits their applications to domestic water heating and space heating. The high temperatures can be achieved by adapting a vacuum between the glass cover and the absorber plate to reduce or eliminate convection losses. Solar high temperatures applications include power generation, air conditioning systems as well as solar industrial heat processes. There are several configurations of evacuate tube collector. The simplest one uses a flat plate with a flow arrangement attached to an evacuated glass cylinder. Introducing a heat pipe as the

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absorbing element in the collector tube greatly enhances the performance of evacuated tube collector. A substantial thermal heat is transferred through the heat pipe with a small temperature difference between heat input and heat output. The heat pipe consists of a closed container filled with a capillary device and charged with small amount of working fluid suitable for the operating conditions. The incoming fluid is absorbed and then vaporized in the pipe. Heat pipes require different working fluids as fluid properties such as vapor pressure and density are temperature dependent. There are two categories of evacuated-tube solar collectors; first is the single-walled glass evacuated-tube and the other is the Dewar tube. The two basic types have a distinction differences. As an example, heat extraction can be through a U-pipe, heat pipe or direct liquid contact.

Evacuated tube collectors usually have much greater efficiencies than the conventional flat plate collectors especially at low temperature and isolation [1-3]. The performance of the heat-pipe collectors have been studied and reported before [4-7]. Badar et al. [8] developed an analytical steady-state model to study the thermal performance of an individual single-walled evacuated-tube with coaxial piping incorporating both single and two-phase flows. A combination of single-walled evacuated-tubes comprising heat pipes with an external or internal concentrator is presented by Nkwetta et al. [9]. Tang et al [10,11] presented a mathematical procedure to determine the optimal choice of tilt and zenith angles. The thermal performance comparisons in two types of the flat plate and vacuum tubes solar collectors have been carried out by Zambolin et al [12]. They concluded that, in the steady-state conditions, the slope of the linear regression instantaneous efficiency with increasing heat losses in flat plate collector is greater than the water-in-glass ETCs. An improved procedure for the experimental characterization of optical efficiency in evacuated tube solar collectors has been introduced by Zambolin et al [13]. At the same environmental conditions, Ayompe et al. [14] conducted an experimental study to compare the performance of both FPC and a heat pipe ETC for domestic water heating system application. The collector efficiencies were found to be 46.1% and 60.7% and the system efficiencies were found to be 37.9% and 50.3% for FPC and heat pipe ETC, respectively. The thermal performance of four differently shaped absorbers of the evacuated-tube is analyzed numerically and experimentally by Kim and Seo [15].

Zhao et al. [16] conducted an experimental and theoretical research on a prototype of a looped heat pipe single walled evacuated tube water heating system. Yin and Harding [17] studied the effect of a range of tube inclinations, manifold flow rates and inlet temperatures. They concluded that for a wide range of operating conditions, buoyancy effects alone resulted in efficient heat transfer. Morrison et al. [18] presented a numerical study on a 45° inclined evacuated tube collector. They found the possible presence of a stagnant region at the bottom of very long tubes. Evacuated tube collectors is preferably used for high temperature applications such as desalination of sea water, air conditioning, refrigeration, and industrial heating processes since their performance is better than that of FPC [19]. The effects of thermal flow and mass rate on forced circulation solar hot water system are experimentally investigated by Gao etal. [20]. Two types of ETC namely water in glass and U pipe evacuated collectors was adapted. Results showed that U pipe evacuated collectors have 25–35% higher energy storage than water in glass. In addition, they concluded that the energy storage and also pump operations are influenced by the flow rate and fluid thermal mass. It is noted that the performance of energy collection will be reduced for higher flow rate. Mangal [21] stated that ETC are strong and long lasting. If any tube is broken, it is just replaced which is considered as a cheaper option compared to flat plate collector (FPC) which require the replacement of the whole collector. Optical and heat loss characteristics of water in glass evacuated tube solar

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heaters is investigated by Budihardjo and Morrison [22].The domestic water heating system was compared with FPC and the performance of 2 panel flat plate arrays was found to be higher than 30 evacuated tube arrays. Shukla et al. [23] mentioned in the review of recent advances in the solar water heating system that the performance of an ETSC is better than mostly used FPC due to its ability to produce high temperature but ETSCs are not widely used because of its high initial cost.

A Dewar tube is made of two thin borosilicate glass walls that form the inner and outer tubes. A selective absorbance coating is deposited on the outside wall of the inner tube to collect solar energy, and the layer between the inner and outer tubes is evacuated to reduce heat loss. However, the water-in-glass (direct liquid contract) tube is more popular than the Dewar tube with a U-pipe or heat pipe inserted because of its lower price. The thermal performance of the Dewar evacuated tube solar collector has been investigated concerning energy balance by Tian [24]. Louise and Simon [25] studied heat transfer and flow structure employing computational fluid dynamics and an optimum inlet flow rate of 0.006–0.015 kg/s has been recommended. The thermal performance of single-phase ETCs with different tilt angles was carried out by Selvakumar et al [26]. They found that the daily efficiency of inclined solar water heater at 22°C is relatively equal to that of inclined solar water heater at 46°C. So, they concluded that the result shows that the thermal performance of water-in-glass solar domestic hot water system is independent of the collector slope.

Ayompe and Duffy [27] analyzed the thermal performance of solar water heating system using heat pipe evacuated tube collector. Measurement data are collected over one year period from a forced circulation solar water heating system. Water was used as the working fluid in the system and the maximum outlet temperature of water was recorded as 70.3° while 59.5° was recorded at the bottom of the hot water tank. Measurements obtained revealed that the heat pipe ETCs are more efficient than FPCs of a solar water heating system. The characteristics and the performance of different types of ETSCs for solar water heating system sthrough out the year was investigated by Arefin et al. [28]. They evaluated the system feasibility by calculating the payback time. They also reported that all glass evacuated tubes are the cheapest and simplest and the heat loss is less than heat pipe collectors as the glass tube collectors are directly connected with the tank. In addition, they determined the operating temperature of the system to be 50° C which is good enough for domestic purposes and their cost analysis showed that the solar water heater using an ETSC is more cost effective than the electric water heater. Morthy [29] studied the performance of solar air conditioning system using ETC. The efficiency of heat pipe evacuated tube varies from 26% to 51% and the overall system has efficiency from 27% to 48%.

Pappis et al. [30] conducted economical and environmental comparison between FPC and ETC. They stated that the ETC are the best choice from the environmental point of view because of the least impact generated during the manufacturing process. On the other hand, FPC is preferable from the economic point of view as it is much cheaper than ETC.

Existing heating systems for industrial process heat are based on steam or hot water from a boiler, which mainly uses fossil fuels like oil, gas and coal or electricity generated by different sources. There are several potential fields of application for solar thermal energy at medium and high temperature level. Heat production for industrial processes is considered as one of the most important application in that concern. Many industrial sectors are considered as preferable technology for the application of solar energy. The most important industrial processes using heat at a mean temperature level are: sterilising, pasteurising, drying, distillation and evaporation, washing and cleaning. A system without heat storage is economically feasible. In this case, the solar heat is forward directly into the desirable process. However, this system is not economically feasible when heat is needed at the early or late hours of the day or at nighttimes. Most of the process heat is used in beverages, food and textile industry for different processes as drying, cleaning, extraction and other different processes. Temperature level for these applications may vary from ambient to low-pressure steam which can be provided either from flat plate or evacuated tube collector. The solar industrial process heat system uses the same system components and mode of operation as solar domestic hot water system. Heat storage is recommended to be added to the system so that the system can operate in periods of low irradiation and/or nighttime. Hot water or low pressure steam at medium temperatures (<150°C) can be used either for preheating of water used for processes or for steam generation.

The most common applications of industrial process heat has been introduced by Schweiger et al. [31]. In addition, he presented the history of solar industrial and agricultural process applications as well as describing practical examples. Integration of solar heat systems into industrial applications requires storage and control strategies to handle the non-continuous supply of solar energy [32]. Benz et al. [33] presented the details of two solar thermal systems installed for producing process heat for a brewery and a dairy in Germany. The solar generation in both industrial processes is found to be comparable to the energy generated from solar systems for domestic solar water heating or space heating.

The main objective of the present work is to investigate thermal performance of evacuated tube solar water collector in hot and harsh climate like Kuwait climate. An experimental test facility is installed at the College of Technological Studies, Kuwait to record the thermal performance of ETC through one year period. Linear regression analysis is adapted to correlate the experimental data to collector thermal efficiency. In the present work, a detailed developed nonlinear model for evacuated tube solar collectors is introduced with more comprehensive optical and thermal analysis. The variation of the temperature along both the circumferential (fin) and the longitudinal (tube) directions is considered in the present model. The predicted numerical values are compared with the experimental values obtained from experimental test facility. The optimum design parameters; collector tube size, mass flow rate and collector tilt angle are determined based on year round ETC thermal performance under the meteorological condition of Kuwait. Finally, the integration of evacuated tube collector to provide heat for industrial processes like soft drink industry is evaluated.

Experimental Setup

Figure 1 shows a schematic diagram of the collector test facility, designed and constructed to conduct this work. It consists of evacuated tube collector (1), radiation pyranometer (2), anemometer (3), storage tank (4) of 100 liters capacity, cross flow heat exchanger (5), and constant temperature circulator (6). A circulator pump (7) is employed to overcome the pressure resistance of the system, pressure relief valve (8), air vent (9). Flow meter (10) to measure mass flow rate, a control valve (11) is used to regulate the flow rate through the circuit with the aid of a valve in the pump by-pass line. Filters (12) and several non-return valves (13) are fitted in the pipeline to define the flow direction.

1. solar collector
2. radiation pyranometer

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3. anemometer
4. storage tank
5. const. temp. circulator
6. cross flow heat exchanger
7. centrifugal pump
8. pressure relief valve
9. air vent
10. flow meter
11. flow control valve
12. filter
13. non return valve

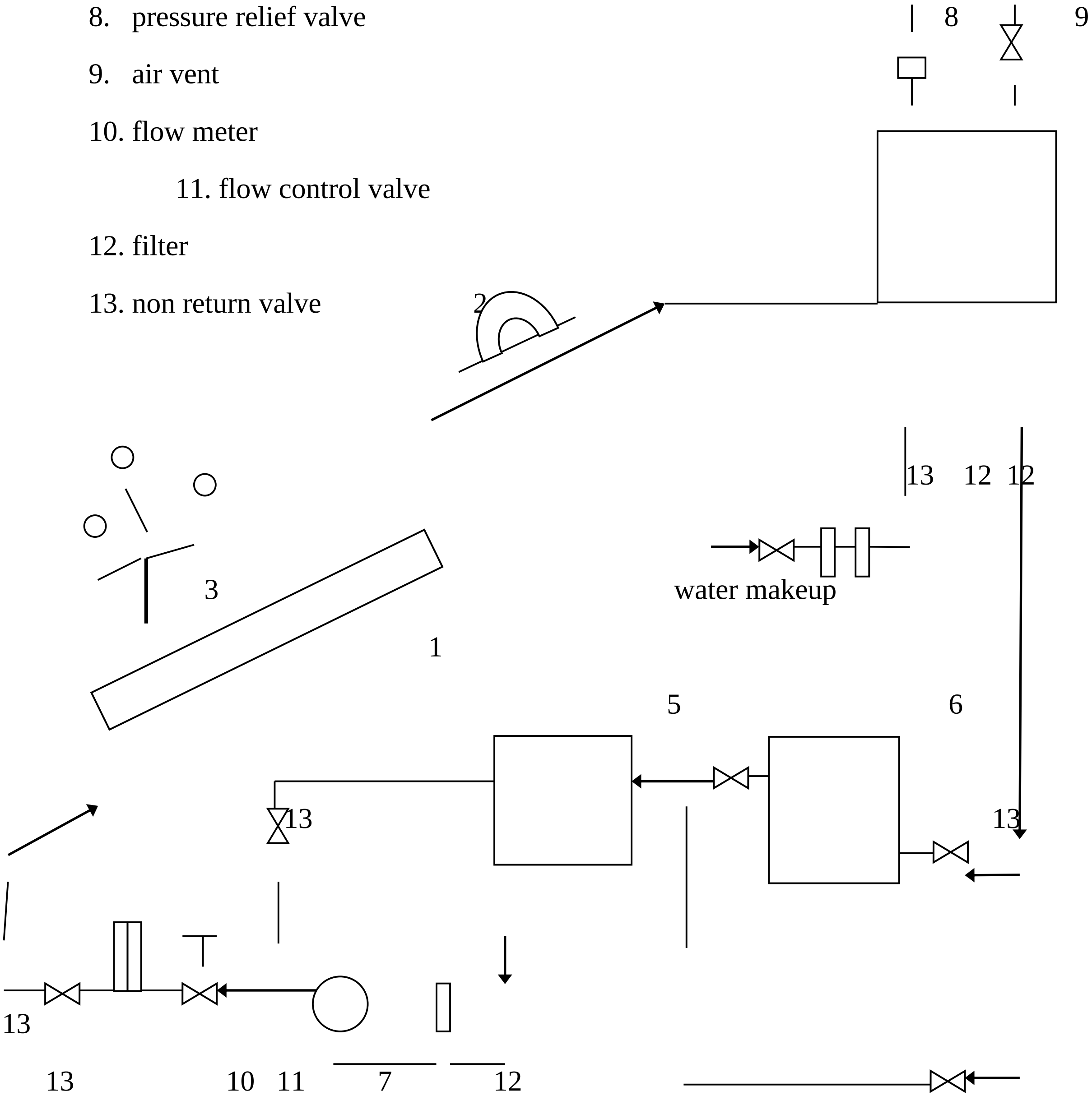

Figure 1. Schematic diagram of the collector test facility

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The circulator is fitted with standard RS232 interface enabling temperature control through the data acquisition system. The heat exchanger is made from finned copper tubes with two fans to force the air across the tubes. The intensity of the global and diffuse solar radiation incident on the collector surface (30° tilted) are measured and recorded by two Precision Spectral Pyranometers. The pyranometer used to measure the diffuse solar radiation is fitted with a shading ring such that the detector is shielded from direct solar radiation to measure the diffuse radiation only.Three standard resistance thermometer detectors (RTD-PT100) are used to monitor the surrounding ambient temperature, inlet and outlet fluid temperatures of the collector.

The water flow rate through the collector is measured using turbine flow meter suitable for 0.2 to 5 liters/min with accuracy of 3%. The evacuated tube collector has an aperture area of 1.8 m^2 and is inclined 30° on the horizontal.

A data acquisition system capable of recording 40 channels is used to record the instantaneous measurements of temperatures, flow rates and solar intensities and to control the collector inlet fluid temperature through the constant temperature water circulator. The data acquisition system has a resolution better than 0.01°C for thermocouple readings and for 4-wires RTD readings. The experiments are carried out for global solar radiation between 600 and 1000 W/m^2, on a 30°-tilted collector surface with average ambient temperatures from 20 to 40°C. The inlet water temperature is changed from around the ambient temperature up to 85°C in 10°C steps. The experimental procedure is started by flushing the system. Then, the system is filled with water and the flow rate is adjusted to the required value. The solar collector is allowed to run for over 30 minutes (about 4 times the collector time constant) to achieve quasi-steady-state conditions before the data collections were started. Each experiment continued for 90 minutes, after that the inlet fluid temperature is changed and a new experiment is started. The collected data are examined to ensure that it presents quasi steady state conditions according to the recommendations outlined by ASHRAE [34]. Knowing the inlet and outlet fluid temperatures and the mass flow rate of water, the useful energy can be evaluated using equation (1). The useful energy may also be expressed in terms of the energy gained by the absorber and the energy lost from the absorber as given by equation (2).

$$Q_u = \dot{m}\, c_p\, (T_o - T_i) \tag{1}$$

$$Q_u = (\tau\alpha)\, A_a\, G - U_L\, A_a\, (T_i - T_a) \tag{2}$$

The instantaneous collector efficiency relates the useful energy to the total radiation incident on the collector surface by equation (3) or (4).

$$\eta = \frac{Q_u}{A_a G} = \frac{\dot{m}\, c_p\, (T_o - T_i)}{A_a G} \tag{3}$$

$$\eta = F_R\, [\,(\tau\alpha) - U_L\, (T_i - T_a)/G\,] \tag{4}$$

The uncertainty analysis shows an experimental error of about 1.1, 1.2 and 2.9% for $F_R U_L$, $F_R(\tau\alpha)$, and the collector efficiency, η, respectively. The uncertainty analysis, for the experimental data fitted by equation 4, revealed that the optical efficiency, $\tau\alpha$, is more sensitive to experimental error than the heat loss coefficient, U_L.

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Theoretical Modeling of ETC

Evacuated tube collector receiver consists of a copper U-tube inside a glass vacuumed tube. The copper tube is surrounded by a cylindrical aluminum fin pressed on it. This fin enhances the heat transfer area between the inner glass absorber surface and the U-tube. The working fluid enters the collector inlet pipe, then it is evenly distributed to the U-tubes, absorbs heat and, at the end, it is returned to the outlet header pipe. The outer cylindrical glass transmits the rays to the inner glass tube, which conducts the energy to the absorber fin. The energy transformed into heat is conducted by the fin to the copper U-tube and finally absorbed by the working fluid, which is water in this case. Figure 2 shows a detailed schematic diagram of the evacuate tube and its cross section view.

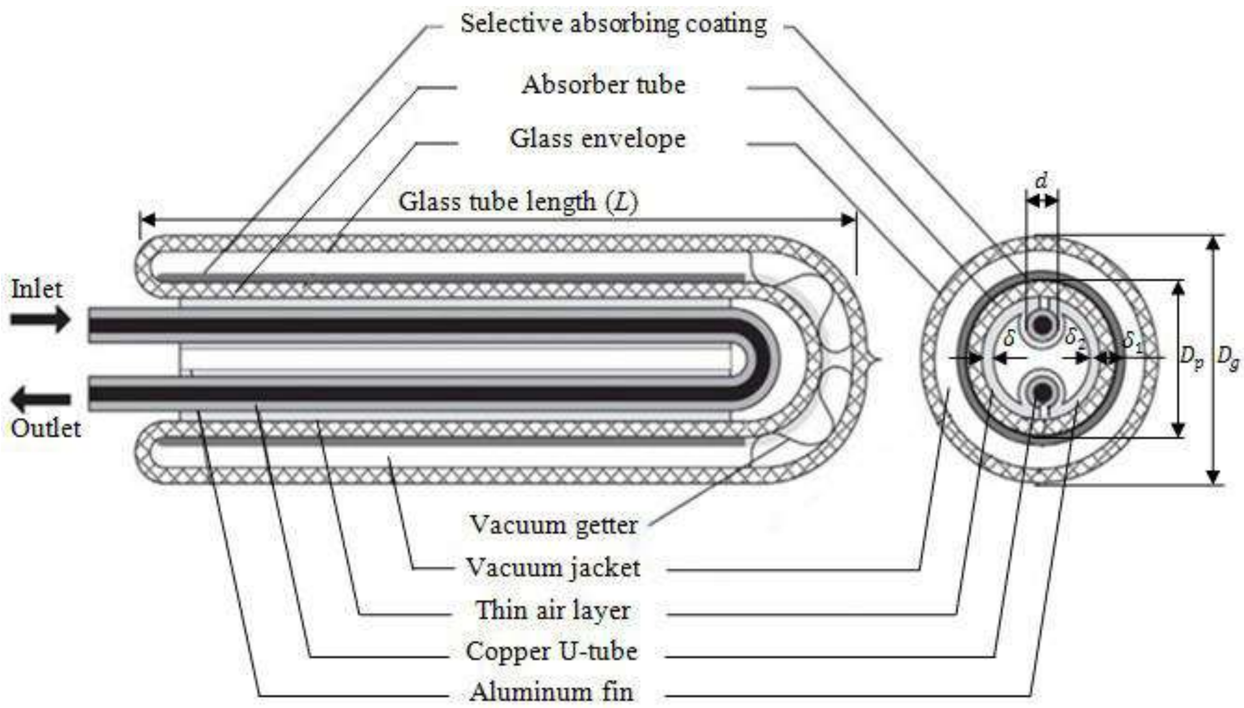

Figure 2. Schematic diagram of the glass evacuated tube solar collector with U-tube

One dimensional analysis for the fin of a single unit of the glass evacuated tube solar collector is carried out. To simplify the model, the following assumptions are considered: i) steady-state conditions are considered with normal incidence angle of solar radiation; ii) thermal resistance of the outer glass tube thickness is negligible; iii) perfect vacuum is assumed between the two glass tubes, thus gas conduction is neglected; iv) an air layer of small thickness is considered between aluminum fin and the absorber glass tube. The absorbed solar power (S), is equal to the incident solar power times the optical losses and can be expressed as:

$$S = (\tau\alpha)GA_a \quad (5)$$

where τ is the glass cover transmittance, α is the glass cover absorptance, G is the global solar irradiance on the collector surface and A_a is the collector aperture area. The theoretical model is based on the heat balance equations for each part of the collector as follows:

Outer glass tube

$$h_{r-g}L_{r-g}\left(T_r - T_g\right) + h_{g-a}L_g\left(T_a - T_g\right) = 0 \quad (6)$$

where T_r, T_g, T_a are the inner glass tube temperature, outer glass tube temperature and ambient temperature; h_{r-g} is the heat transfer coefficient between the inner and outer glass tubes; and h_{g-a} is the heat transfer coefficient between the outer glass tube and the ambient environment. Heat conduction can be ignored because the space between the inner and outer tubes is a very narrow vacuum layer, in which the vacuum level is 10^{-4} Pa and the heat conduction coefficient is less than 0.27 x 10^{-5} W/m °C [34]. Considering $h_{r-g} = h_{r-g,rad} + h_{r-g,cond}$, where $h_{r-g,rad}$ and $h_{r-g,cond}$ are the radiation and conduction heat transfer coefficients between the outer and inner glass tube Also, $h_{g-a} = h_{g-a,rad} + h_{g-a,conv}$, where $h_{g-a,rad}$ and $h_{g-a,conv}$ are the radiation and convection heat transfer coefficients between the outer tube and the ambient environment, and $h_{g-a,conv}$ is a function of the wind velocity. $L_{r-g} = (L_r + L_g)/2$, where L_r and L_g are the perimeters of the inner and outer glass tubes.

Inner glass tube

$$h_{r-g}L_{r-g}\left(T_g - T_r\right) + h_{r-Al}L_{r-Al}(T_{Al} - T_r) + Q_e + \tau_g\alpha_r D_g G = 0 \quad (7)$$

where T_{Al} is the aluminum fin temperature; and h_{r-Al} is the total heat transfer coefficient between the inner glass tube and the aluminum fin, and $h_{r-Al} = h_{r-Al,rad} + h_{r-Al,cond}$, where $h_{r-Al,rad}$ and $h_{r-Al,cond}$ are the radiation and conduction heat transfer coefficients between inner glass tube and aluminum fin, τ_g is the transmission coefficient of the outer glass tube; τ_r is the absorption coefficient of the selective coating on the inner glass tube; d_g is the diameter of the outer glass tube; G is the total solar irradiance on the collector aperture surface; $L_{r-Al} = (L_r + L_{Al})/2$, where L_{Al} is the perimeter of aluminum fin, and Q_e is the heat loss of the tube edge to the manifold.

Aluminum Fin

$$h_{r-Al}L_{r-Al}(T_r - T_{Al}) + h_{Al-Cu}L_{Cu}(T_{Cu} - T_{Al}) = 0 \quad (8)$$

where T_{Cu} is the U copper pipe temperature; and h_{Al-Cu} is the total heat transfer coefficient between the aluminum fin and U copper pipe, $h_{Al-Cu} = h_{Al-Cu,rad} + h_{Al-Cu,cond}$, where $h_{Al-Cu,rad}$ and $h_{Al-Cu,cond}$ are the radiation and conduction heat transfer coefficients between the aluminum fin and U copper pipe, and L_{Cu} is the perimeter of the copper pipe.

U copper pipe

$$h_{Al-Cul}L_{Cu}(T_{Al} - T_{Cu}) + h_f L_{Cu}(T_f - T_{Cu}) = 0 \quad (9)$$

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where h_f is the fluid convective heat transfer coefficient inside the U copper pipe and it varies with flow velocity.

Working fluid

$$h_f L_{Cu}(T_{Cu} - T_f)\, x + \dot{m} c_p (T_i - T_o) = 0 \tag{10}$$

where T_i and T_o are the inlet and outlet temperatures of working fluid in the U copper pipe and x is the interval of the fluid along the tube axis. $\dot{m}$ and c_p are the mass flow rate and specific heat capacity of the working fluid.

The radiation heat transfer coefficient ($h_{r-g,rad}$) between the outer and inner glass tube is given by:

$$h_{r-g,rad} = \frac{\sigma\left(T_r^2 + T_g^2\right)\left(T_r + T_g\right)}{\frac{1}{\varepsilon_r} + \frac{1}{\varepsilon_g} - 1} \tag{11}$$

where ε_r and ε_g are the emission coefficients of the inner tube selective coating and glass of the outer glass tubes, respectively. Under the conditions given by the industrial temperature range, the ambient air can be treated as a transparent body of heat radiation, so that $h_{g-a,rad}$ can be ignored.

The convection heat transfer coefficient (h_f) between the fluid and the U-pipe wall can be expressed as [35]:

$$h_f = \frac{\lambda_f Nu}{D} = \frac{\lambda_f}{D} 1.75 \left[Gz + 5.6 \times 10^{-4}\left(GrPr\frac{L_T}{D}\right)^{0.7}\right]^{\frac{1}{3}} \left(\frac{\mu_{av}}{\mu_{wall}}\right)^{0.14} \tag{12}$$

where Nu is Nusselt number, Gz is Graetz number, Pr is Prandtl number, L_T is the length of the tube, D is equivalent diameter for the theoretical model, μ_{av} and μ_{wall} are the viscosity of fluid at average fluid temperature and wall temperature and λ_f is the thermal conductivity of the fluid.

The previous equations are adapted to develop a theoretical model to predict the performance of evacuated tube collector under different climatic conditions.

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RESULTS AND DISCUSSIONS

Evacuated Tube Collector Performance

A comparison between the ETC performance predicted by the present model and the corresponding performance resulted from linear regression of experimental data is presented in Figure 3 for reduced temperature, $(T_i\text{-}T_a)/G$, ranging from 0 to 0.08 m^2 K/W.

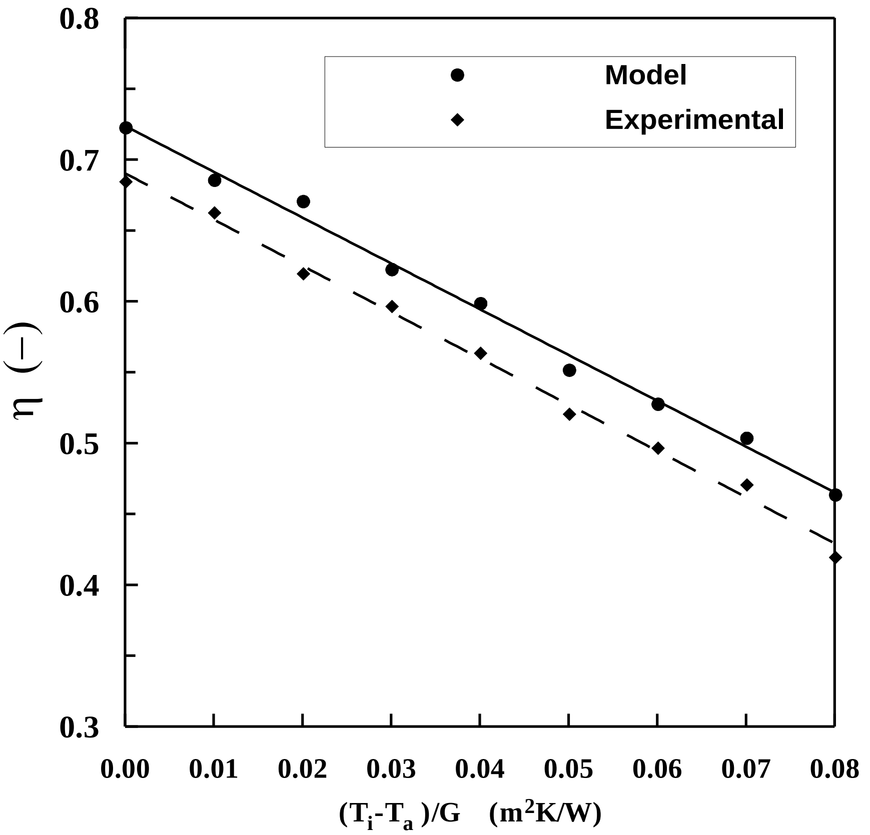

Figure 3. Comparison between simulated and experimental ETC thermal efficiency

Simulations are carried out under different test conditions to obtain the collector efficiency curve shown in Figure3. The weather data adapted for present work is the hourly values of global radiation and ambient temperature measured for a period of two years in the College of Technological Studies, Kuwait.

The efficiency curves obtained from the present model and experimental data fitting are:

$$\eta_{model} = 0.718 - 3.19\frac{(T_i - T_a)}{G} \quad (13)$$

$$\eta_{experiment} = 0.699 - 3.30\frac{(T_i - T_a)}{G} \quad (14)$$

Error analysis revealed that the maximum deviation between model and experimental optical efficiency is about 2.6% while the maximum deviation between model and experimental

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overall heat loss coefficient is approximately 3.3%. The above graph and these predictions clearly demonstrate the reliability of the present developed ETC numerical model.

After validation and checking reliability, the theoretical model is employed to determine optimum collector parameters. Figure 4 presents the variation of the annual energy production from collector for various collector orientations. The slope of the collector is changed from 0° to 60° (i.e. latitude ± 30°). In addition, different azimuth angles are examined ranging from 0° (due south) to 40° west of south. Each combination of slope and azimuth angle is examined in a single simulation period which is one year.

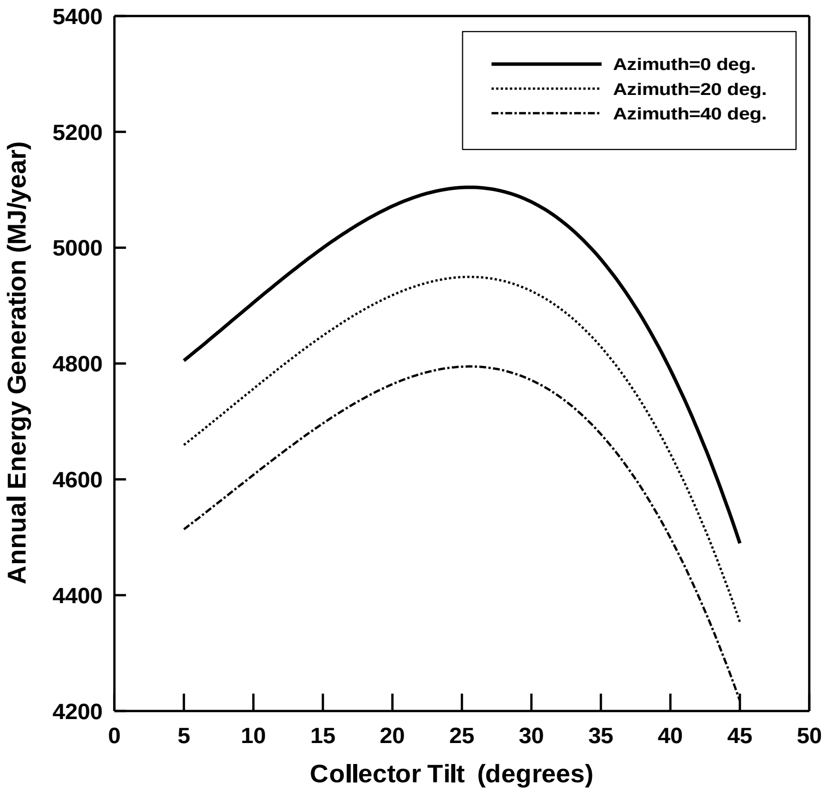

Figure 4. Variation of Annual energy generation with collector tilt

As seen from the figure, the energy production changes with both collector orientation and azimuth angle. It is obvious that the maximum energy generation from the collector corresponds to tilt equal to 25° (i.e. latitude -5°) and for collector facing south (azimuth angle=0°). Maximum energy production at angles greater than latitude is in accordance with the fact that more solar energy is available in summer than in winter in Kuwait. So, annual energy production can be maximized by using collector sloped at an angle of 25°.

In spite of the significant effect of tube length and tube diameter on the thermal efficiency and the manufacturing costs of evacuated tube collectors, few studies are carried out to analyze this effect. So, it is of high importance to analyze the thermal efficiency variation as function of the tube size. The variation of the thermal efficiency with different tube lengths is studied.

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Figure 5 shows this variation with the smallest and the biggest tube lengths studied (30 mm and 60 mm).

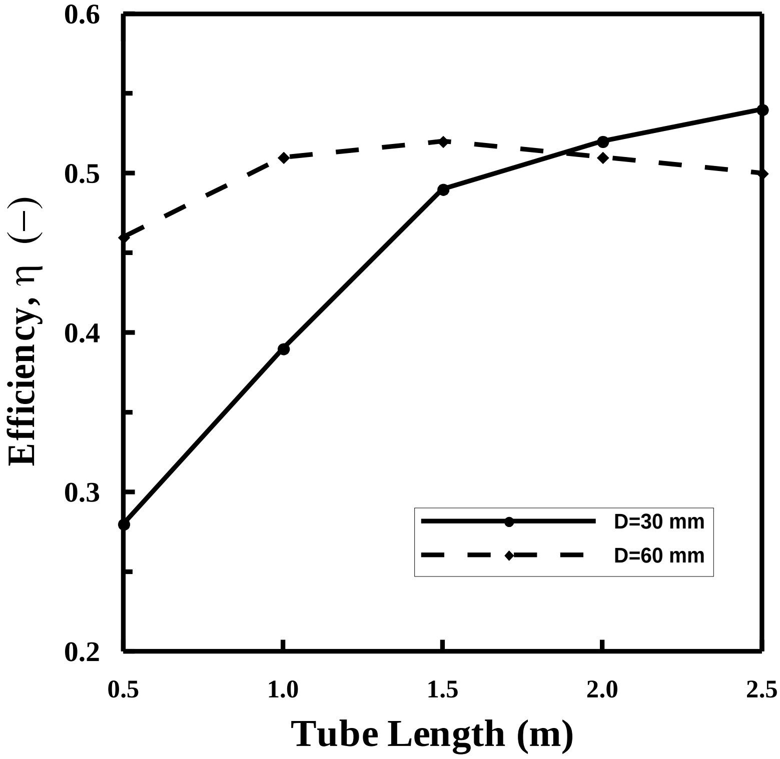

Figure 5. Variation of thermal efficiency with tube size

As seen from Figure 5, there is a significant increase in the thermal efficiency with increase in tube length for the small tube diameter of 30 mm. Thermal efficiency at tube diameter of 30 mm is almost doubled when tube length is increased from 0.5 m to 2.5 m. On the other hand, for the bigger tube size (60 mm) the variation in thermal efficiency with tube length is not significant. The efficiency for the bigger tube size initially slightly increases with tube length then decreases for tube lengths greater than 1.5 m. The results indicate that the optimum tube length is 1.5 m as at this length a significant enhancement can be achieved in thermal efficiency for different tube diameters. It should be noted that both large and small tube lengths are not desirable for good thermal efficiency. That is mainly due to the fact that energy storage increases with tube length increase, because tube outlet temperature and pump running time increase. On the other hand, energy storage decreases with tube length decrease because of the increase in heat losses. So, this let us conclude that the optimum tube length is 1.5 m, as also collector manufacturing cost, would greatly increase for higher tube lengths.

Figure 6 shows the variation of collector thermal efficiency with mass flow rate for the optimum tube size (tube length =1.5 m, tube diameter=30 mm). Efficiency increases with mass flow rate up to an optimal value of 30 kg/h.m^2 where thermal efficiency reaches its highest maximum value of 0.53. Then efficiency starts to decrease for higher mass flow rates. Optimum mass flow rate is an important factor in collector design. For smaller flow rates, collector operates at a higher average temperature causing an increase in heat loss. On the other

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side, higher flow rates will result in unstable collector outlet temperatures leading to increase in heat loss. That is shows the importance of determining the optimal flow rate.

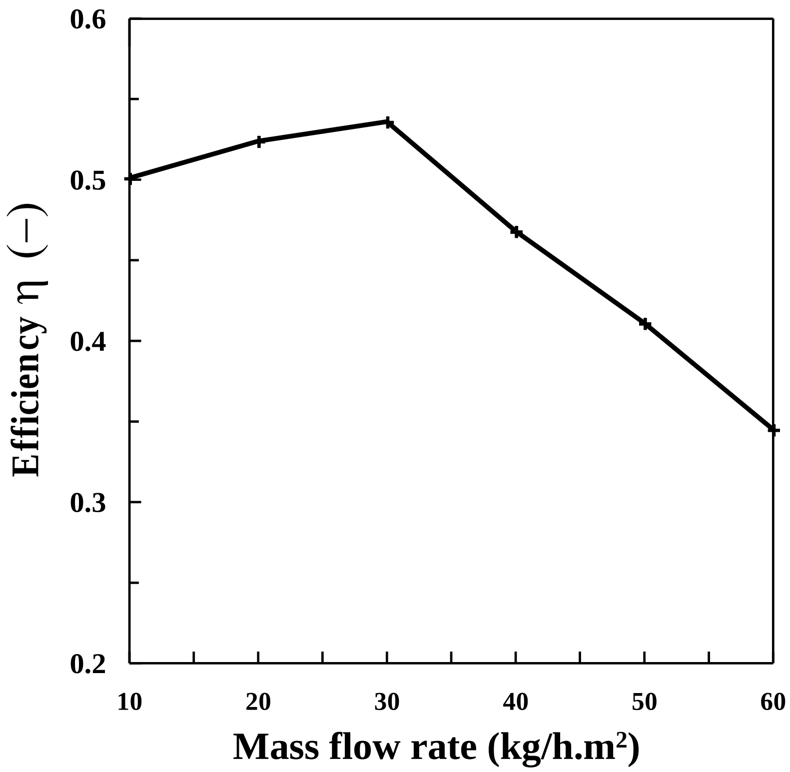

Figure 6. Variation of thermal efficiency with mass flow rate

Integration of Solar Thermal Systems for Industrial Processes Heat

In the present work, the optimum evacuated tube parameters obtained from the theoretical model is employed to design and size solar thermal system to generate the heat required for soft drinks industry. Transient simulation program (TRNSYS) is adapted to determine the required solar collector area, energy generation, the corresponding solar fraction as well as the life cycle savings of the solar system using climatic data of Kuwait. The life cycle savings of a solar heating system over a conventional heating system is defined as the difference between the reduction in fuel costs and the increase in expenses resulting from the cost of solar system components. Heat generated from evacuated tube collector is utilized for direct heating of a circulating fluid in industrial processes as shown in Figure 7.

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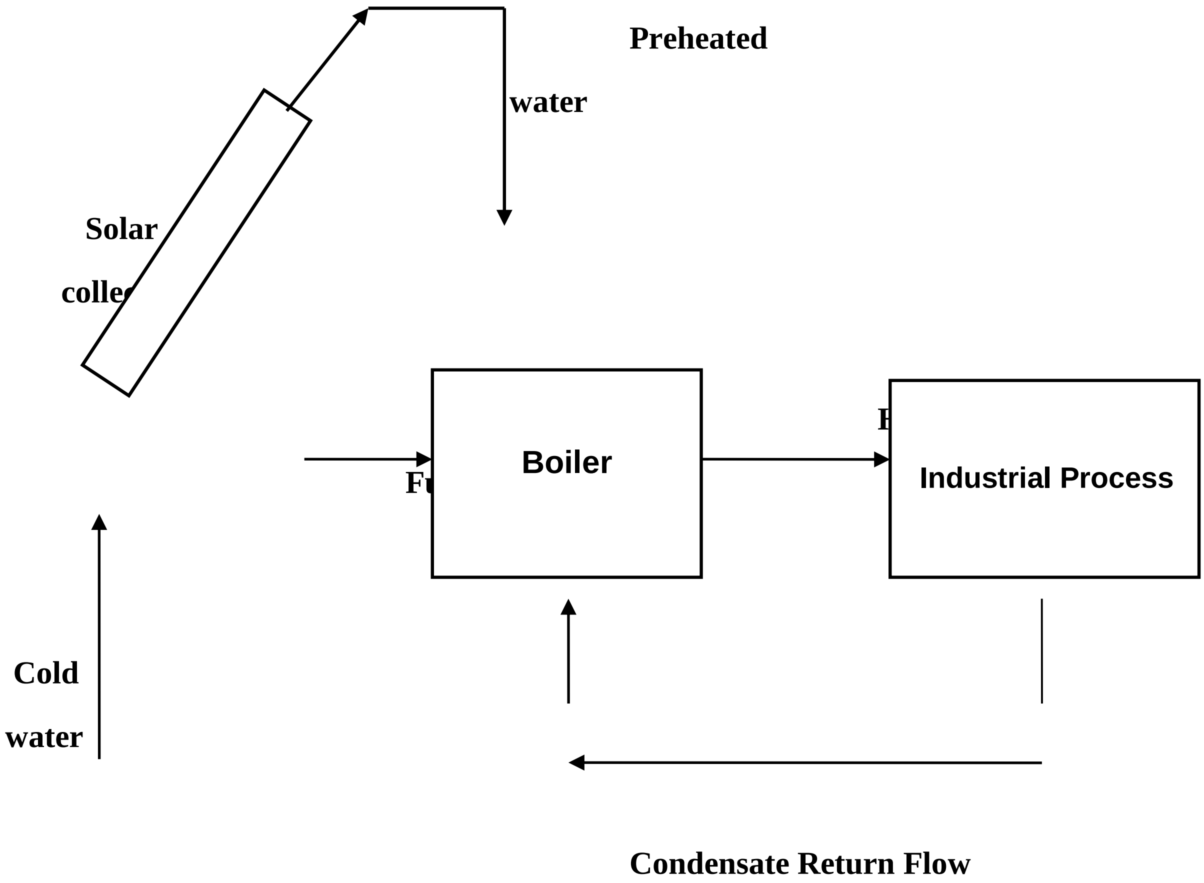

Figure 7. Solar Thermal System for Industrial Processes

In soft drinks industry, the consumption of hot water is mainly in the preparation of the syrup (water and sugar) and washing of the bottles. Steam is used in both processes to give the heating effects. In the preparation room, a mixture of water, sugar and syrup is heated together in a tank with steam passing in a coil around the tank. The washing process includes two steps, in the first step the bottles are washed with a mixture of sodium hydroxide and water. The second step is to rinse the bottles with water only. Most of the energy consumed in soft drinks is used to produce hot water at temperature approximately equal to 80°C.

The variation of both annual solar fraction (F) and life cycle savings (LCS) with evacuated tube collector area is shown in figure 8 for the optimum parameters obtained from the theoretical mode. The graph data are calculated for a medium size soft drink industry for hot water demand of 4000 kg/day. The economic parameters adapted for calculations are: cost per unit area 500 $\$/m^2$, area independent cost \$2000, annual increase in maintenance 5%, annual mortgage interest rate 7% and period of economic analysis 20 years. As seen from figure, as the collector area increases, solar fraction consequently increases. As seen hot water load demand can be completely satisfied at collector area of about 150 m^2 as solar fraction at this area reaches 1. Also, life cycle savings increases with increasing collector area until reaching its maximum value at an optimum collector area of 110 m^2 corresponding to solar fraction of 0.9. As the

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collector area is further increased, the excessive system costs force the solar savings to decrease as shown from the figure. These results indicate that solar system can provide 90% of the heat required for soft drink industry with life cycle savings of approximately $18,000 (i.e. about $900/year).

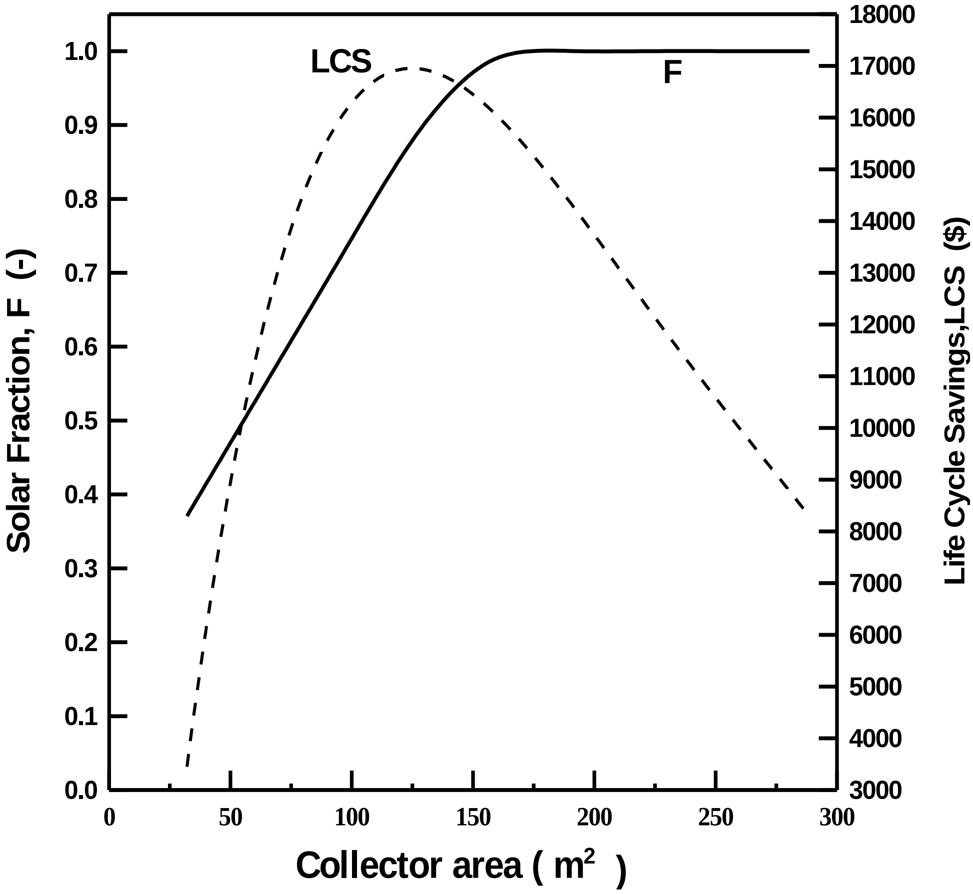

Figure 8. Variation of solar fraction and life cycle savings with collector area

CONCLUSIONS

An experimental rig facility is installed to monitor the thermal performance of ETC through one year period. A detailed developed nonlinear model for evacuated tube solar collectors with more comprehensive optical and thermal analysis is presented. In addition, integration of solar system in industrial process heat is examined. Based on the results obtained from current study, the following the following conclusions can be drawn:

- The predicted numerical values agree well with the experimental values indicating the reliability of the present developed model.
- The maximum energy generation from the collector corresponds to tilt equal to 25° (i.e. latitude - 5°) and for collector facing south (azimuth angle=0°).

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- The optimum tube length is 1.5 m as at this length a significant enhancement can be achieved in thermal efficiency for other tube diameters studied.
- The optimal mass flow rate under Kuwait climatic conditions is 30 kg/h.m^2 as thermal collector efficiency reaches its highest maximum value of 0.53 at this optimum mass flow rate.
- Integration of evacuated tube collector in soft drink industry can provide 90% of the heat required for soft drink industry with life cycle savings of approximately $18,000 (i.e. about $900/year).

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Nomenclature

A_a	aperture area	m^2
c_p	specific heat	J/kgK
D_g	glass tube diameter	m
F'	collector efficiency factor	—
F_R	heat removal factor	—
$h_{Al\text{–}Cu}$	total heat transfer coefficient between aluminum fin and U copper pipe	W/m^{2}K
$h_{g\text{–}a}$	heat transfer coefficient between glass tube and ambient	W/m^{2}K
$h_{g\text{–}a,conv}$	convection heat transfer coefficient between outer tube and ambient	W/m^{2}K
$h_{g\text{–}a,rad}$	radiation heat transfer coefficient between outer tube and ambient	W/m^{2}K
h_f	fluid convective heat transfer coefficient	W/m^{2}K
$h_{r\text{–}Al}$	total heat transfer coefficient between inner glass tube and aluminum fin	W/m^{2}K
G	global solar radiation on collector surface	W/m^2
Gz	Graetz number	—
L_g	perimeter of outer glass tube	m
L_r	perimeter of inner glass tube	m
L_T	Total tube length	m
$\dot{m}$	mass flow rate of water	kg/s
Nu	Nusselt number	—
Pr	Prandtl number	—
Q_u	rate of useful energy gained	W
T_a	ambient temperature	K
T_{Al}	aluminum fin temperature	K
T_{Cu}	U copper pipe temperature	K
T_r	outer glass tube temperature	K
T_i	inlet temperature	K
T_o	outlet temperature	K
T_r	inner glass tube temperature	K
S	absorbed solar power	W
U_L	overall heat loss coefficient	W/m^{2}K

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Greek Letters

α	absorbitivity	—
ε_g	emission coefficient of outer glass tubes	—
ε_r	emission coefficients of inner tube selective coating	—
η	collector efficiency	—
λ_f	thermal conductivity of the fluid	W/mK
μ_{av}	viscosity of fluid at average fluid temperature	$N.s/m^2$
μ_{wall}	viscosity of fluid at wall temperature	$N.s/m^2$
τ	transmissivity	—

REFERENCES

[1] Morrison G, Budihardjo I, Behnia M. Water-in-glass evacuated tube solar water heaters. Sol Energy 2004;76:135–40.
[2] Zubriski S E, Dick K. Measurement of the efficiency of evacuated tube solar collectors under various operating conditions. College Publishing; 2012.p. 114–130.
[3] Kalogirou S A. Solar thermal collectors and applications .Prog Energy Combust Sci 2004; 30:231–95.
[4] L.M. Ayompe, A. Duffy, "Thermal performance analysis of a solar water heating system with heat pipe evacuated tube collector using data from a field trial", Solar Energy. Dublin. Ireland, vol. 90, pp. 17-28, January 2013.
[5] D. A.G.Redpath, P. C.Eames, S. N.G.Lo and P. W.Griffiths, "Experimental investigation of natural convection heat exchange within a physical model of the manifold chamber of a thermosyphon heat-pipe evacuated tube solar water heater", Solar Energy. Coventry. England, January 2009, Vol. 83, pp. 988-997,.
[6] S.B. Riffat, X. Zhao, P.S. Doherty, "Developing a theoretical model to investigate thermal performance of a thin membrane heat-pipe solar collector", Applied Thermal Engineering. Nottingham. UK, August 2004, Vol. 25, pp. 899-915.
[7] C.C. Chien, C.K. Kung, C.C. Chang, W.S. Lee, C.S. Jwo and S.L. Chen, "Theoretical and experimental investigations of a two-phase thermosyphon solar water heater", Energy. Taipei. Taiwan, October 2010, Vol. 36, pp. 415-423.
[8] Badar, A.W., Buchholz, R., Ziegler, F., 2012. Single and two-phase flow modeling and analysis of a coaxial vacuum tube solar collector. Sol. Energy 86 (1), 175–189.
[9] Nkwetta, D.N., Smyth, M., Zacharopoulos, A., Hyde, T., 2013. Experimental field evaluation of novel concentrator augmented solar collectors for medium temperature applications. Appl. Therm. Eng. 51 (1–2), 1282–1289.
[10] Tang R, Gao W, Yu Y, Chen H. "Optimal tilt-angles of all-glass evacuated tube solar collectors", Energy.2009, 34: 1387-1395.
[11] Tang R, Yang Y, Gao W. "Comparative studies on thermal performance of water-in-glass evacuated tube solar water heaters with different collector tilt-angles.", Solar Energy.2011, 85: 1381-1389.
[12] Zambolin E, Del Col D. "Experimental analysis of thermal performance of flat plate and evacuated tube solar collectors in stationary standard and daily conditions", Solar Energy. 2010, 84: 1382-1396.

ISSN 2055-0197(Print), ISSN 2055-0200(Online)

[13] Zambolin E, Del Col D. “An improved procedure for the experimental characterization of optical efficiency in evacuated tube solar collectors”, Renewable Energy. 2012, 43: 37-46.
[14] Ayompe L, Duffy A, McKeever M, Conlon M, McCormack S. Comparative field performance study of flat plate and heat pipe evacuated tube collectors (ETCs) for domestic water heating systems in a temperate climate. Energy 2011; 36:3370–8.
[15] Kim, Y., Seo, T., 2007. Thermal performances comparisons of the glass evacuated tube solar collectors with shapes of absorber tube. Renew. Energy 32, 772–795.
[16] Zhao, X.D., Wang, Z.Y., Qi, T., 2010. Theoretical investigation of the performance of a novel loop heat pipe solar water heating system for use in Beijing China. Appl. Therm. Eng. 30, 2526–2536.
[17]Yin, Z.Q., Harding, G.L., 1984. Water in glass manifolds for heat extraction form evacuated solar collector tubes. Sol. Energy 32, 223–230.
[18] Morrison, G.L., Budihardjo, I., Behnia, M., 2004. Water-in-glass evacuated tube solar water heaters. Sol. Energy 76, 135–140.
[19] Budihardjo I, Morrison G, Behnia M. Development of TRNSYS models for predicting the performance of water-in-glass evacuated tube solar water heaters in Australia. Proceedings of ANZSES annual con .; 2003.
[20] Gao Y, Zhang Q, Fan R, Lin X, Yu Y. Effects of thermal mass and flow rate on forced circulation solar hot water system: comparison of water in glass and U-pipe evacuated tube solar collectors. Sol Energy 2013; 98:290–301.
[21]Mangal D, Lamba D K, Gupta T, Jhamb K. Acknowledgement of evacuated tube solar water heater over flat plate solar water heater. IntJEng 2010;4:279.
[22] Budihardjo I, Morrison G. Performance of water-in-glass evacuated tube solar water heaters. SolEnergy 2009; 83:49–56.
[23] Shukla R, Sumathy K, Erickson P, Gong J .Recent advances in the solar water heating systems: a review. RenewSustEnergRev 2013;19:173–90.
[24] Tian, Q., 2007. Thermal performance of the U-type evacuated glass tubular solar collector. Build. Energy Environ. 26 (3), 51–54.
[25] Louise, J.S., Simon, F., 2007. Theoretical flow investigations of an all glass evacuated tubular collector. Sol. Energy 81 (6), 822–828.
[26] Selvakumar P, Somasundaram P. “Effect of Inclination Angle on Temperature Characteristics of Water in-Glass Evacuated Tubes of Domestic Solar Water Heater”, International Journal of Engineering and Innovative Technology, 2012, 1(4): 1-4.
[27] Ayompe L, Duffy A, McKeever M, Conlon M, McCormack S. Comparative field performance study of flat plate and heat pipe evacuated tube collectors (ETCs) for domestic water heating systems in a temperate climate. Energy 2011; 36:3370–8.
[28] Arefin M, Hasan M, Azad A. Characteristics and cost analysis of an automatic solar hot water system in Bangladesh. Int Proc Chem Biol Environ Eng 2011;6.
[29] Moorthy M. Performance of solar air-conditioning system using heat pipe evacuated tube collector. National conference in mechanical engineering research and postgraduate studies; Pahang: UMP Pekan, 2010.
[30] Pappis F, Friderichs A, Serafini S, Foletto E. Economic-environmental comparison between Flat Plate and Evacuated Tube Solar Collectors. Global NESTJ 2014:16.
[31] Schweiger H, Mendes JF, Benz N, Hennecke K, Prieto G, Gusi M, Goncalves H. The potential of solar heat in industrial processes. a state of the art review for Spain and Portugal. Proceedings of Eurosun’2000 Copenhagen, Denmark, (2000)..

[32] Schramm, S., Adam, M. "Storage in Solar Process Heat Applications", Energy Procedia, Proceedings of the 2nd International Conference on Solar Heating and Cooling for Buildings and Industry (SHC 2013), 2014, V. 48, pp. 1202-1209.
[33] Benz N, Gut M, Rub W. Solar process heat in breweries and dairies. Proceedings of EuroSun 98, Portoroz, Slovenia, (1998).
[34] ASHRAE, ASHRAE Standard 93-86. Methods of Testing to Determine the Thermal Performance of Solar Collectors. Atlanta, Georgia, USA, (1986).

ISSN 2055-0197(Print), ISSN 2055-0200(Online)