

Original Research Article

**Performance Analysis of Parabolic Trough
Collector in Hot Climate**

ABSTRACT

The potential for reduction of greenhouse gases emissions is an important issue for the wide spread of renewable energy systems. In most existing buildings cooling and heating loads lead to high primary energy consumption and consequently high CO₂ emissions. These can be substantially decreased with suitable energy concepts using appropriately integrated renewable systems. In the present study a numerical model is developed to study the effect of different collector parameters and operating conditions on the performance of parabolic trough solar collector (PTSC) in Kuwait climate. The present model takes into consideration the thermal interaction between absorber-envelope, and envelope-envelope for thermal radiation losses which have been neglected in the existing models. To account for the thermal interaction between adjacent surfaces, a comprehensive radiative analysis was implemented in the present study for the heat losses in the absorber and the glass envelope. A review of the correlations for convective heat transfer losses was performed as well and new correlations were are in the present model. The effects of heat conduction in the collector tube wall and the mixed convection in the inner tube which have been neglected in the previous studies are also taken into account in the present developed model. In addition a case study is carried out adapting parabolic trough collectors to satisfy nominal space heating load, water heating load and cooling load of a typical Kuwaiti house. Finally, the environmental impact of solar heating and cooling systems under Kuwait climate conditions is investigated. Present results indicate that the performance of parabolic trough collector can be significantly enhanced by optimizing its parameters as well as operating conditions. It is found that convection loss from absorber tube to supporting structures is the largest among the other losses (conduction and radiation). Also, at noon time PTSC has the smallest incident angle and the highest efficiency and when a vacuum is present in the annulus between the receiver surface and the glass envelope, the conduction and the convection across the annulus are effectively eliminated. In addition, space heating load and domestic water heating load can be completely provided by PTSC and the minimum required collector area is about 82 m² to supply cooling loads of a typical residential house in all hot climate conditions of Kuwait. Finally, a total avoid CO₂ emission of about 12.3 tonne/year can be achieved as a result of adapting parabolic trough solar collector in the house.

Keywords: Parabolic trough collector, building integrated renewable systems, solar fraction, avoided CO₂ emission.

1. INTRODUCTION

Energy consumption worldwide is increasing rapidly due to the increasing global population and the industrialization processes in many countries. The production and use of conventional fossil fuel energy resources accounts for a high percentage of air pollution

20 leading to harmful impact on our environment. In contrast, renewable energy sources can be
 21 adapted to produce energy sources with little-if any pollution. Widespread commercialization
 22 of renewable energy systems especially in hot climate regions like Kuwait can significantly
 23 reduce energy consumption of conventional fuels. In return, this will help greatly in reducing
 24 environmental pollution and maintain our environment healthy and clean. Concentrating
 25 solar power (CSP) technologies now constitute feasible commercial options for large scale
 26 power plants as well as for smaller electricity and heat generating devices. There are
 27 currently four basic commercially available CSP technologies. These plants typically consist
 28 of three main circuits: the Solar Field, through which the heat transfer fluid (HTF) circulates,
 29 the Power Block, which circulates water and steam, and the TES system. The HTF and
 30 water-steam circuits and the HTF and Thermal Storage System (TES) circuits can exchange
 31 energy at the corresponding heat exchangers.

32 Parabolic trough collector is a type of high temperature solar concentrator collector that has
 33 a linear parabolic shaped reflector that focuses the sun's radiation on a receiver at the focus
 34 of the reflector. The parabolic troughs are long parallel rows of curved glass mirrors usually
 35 coated silver or polished aluminum focusing the sun's energy on an absorber pipe located
 36 along its focal line. Parabolic trough collectors use the high temperatures to heat up a fluid in
 37 absorber tubes arranged in the focal line. The heat transfer fluid (HTF), oil, is circulated
 38 through the pipes. Under normal operation the heated HTF leaves the collector with a
 39 specified collector outlet temperature and is pumped to a central power plant area. There,
 40 the HTF is passed through several heat exchangers where its energy is transferred to the
 41 power plant's working fluid, which is usually steam. The heated steam is used in turn to drive
 42 a turbine generator to produce electricity. Thermal storage can be used in these systems to
 43 provide electricity during peak hours or when the sun intensity is low. Solar collector tube is
 44 a key component in parabolic trough solar thermal generation system, used to convert solar
 45 radiation to thermal energy. Optimizing its performance and improving its efficiency have
 46 important effects on the thermal-electricity conversion efficiency.

47 Kalogirou et al. [1] studied the performance of parabolic trough collector using a theoretical
 48 model. They predicted the quantity of steam produced by the system. The optimum flash
 49 vessel diameter and inventory obtained from this analysis are 65 mm and 0.7 l, respectively.
 50 The current state of the art of parabolic trough solar power technology is reviewed by Price
 51 et al. [2]. They described the research efforts that are in progress to enhance this
 52 technology. They claimed that since the last commercial parabolic trough plant was built,
 53 substantial technological progress has been realized. The review presented shows how the
 54 economics of future parabolic trough solar power plants are expected to improve. Klaus et
 55 al. [3] developed two methods to measure the solar flux in the focal region of parabolic
 56 trough collector. Intercept losses are detected and assigned to the locations on the parabola
 57 from which they are caused. Direct steam generation in parabolic trough collectors is studied
 58 by Eck et al. [4]. They presented the advantages, disadvantages, and design considerations
 59 of a steam cycle operated with saturated steam. A simulation study of solar lithium bromide–
 60 water absorption cooling system with parabolic trough collector is carried out by Mazloumi et
 61 al. [5]. The system has been designed to supply the cooling load of a typical house where
 62 the cooling load peak is about 17.5 kW (5 tons of refrigeration), which occurs in July. The
 63 minimum required collector area was estimated to be about 57.6 m², which could supply the
 64 cooling loads for the sunshine hours of the design day for July.

65 A reflux heat transfer storage concept for producing high-temperature superheated steam in
 66 the temperature range 350–400 °C was developed and tested by Adinberg et al. [6]. The
 67 thermal storage medium is a metallic substance, Zinc–Tin alloy, which serves as the phase
 68 change material. A high-temperature heat transfer fluid is added to the storage medium in
 69 order to enhance heat exchange within the storage system and the associated heat

70 exchangers serving for charging and discharging the storage. The tests produced
71 satisfactory results in terms of thermal stability and compatibility of the utilized storage
72 materials up to a working temperature of 400 °C. Garcia et al. [7] presented an overview of
73 the parabolic-trough collectors that have been built and marketed during the past century, as
74 well as the prototypes currently under development. In addition, they presented a survey of
75 systems which could incorporate this type of concentrating solar system to supply thermal
76 energy up to 400 °C, especially steam power cycles for electricity generation.

77 Sansoni et al. [8] investigated parabolic trough collectors and applied the study results to a
78 plant prototype installed in Florence (Italy) for residential supply. Their work summarizes the
79 results of several studies and analyzes the interactions between collection efficiency,
80 angular misalignments, mirror deformations, sun tracking and trough placement. They
81 concluded that the placement of parabolic trough collector and sun tracking should be
82 matched to optimize the collection performance. Daniel et al [9] employed selective coatings
83 with evacuated and non evacuated glass tubes to control radiative and convective heat
84 losses. They developed a numerical technique to investigate the performance of the vacuum
85 shell. They found that the performance of the evacuated tube with selective coating is the
86 highest among all configurations.

87 A detailed heat transfer analysis and modeling of a parabolic trough solar receiver was
88 carried out by Forristall [10]. One and two dimensional energy balances were used for short
89 and long receivers respectively. This model was used to determine the thermal performance
90 of parabolic trough collectors under different operating conditions. Jones et al. [11] carried
91 out a comprehensive model of the 30 MWe SEGS VI parabolic trough plant. The model was
92 created to accurately predict the SEGS VI plant behavior and to examine transient effects
93 such as start up, shutdown and system response. Li Xu et al. [12] developed a
94 comprehensive physical model to predict the thermal performance of parabolic trough
95 collectors during transient operations. Four sets of outdoor tests were conducted to verify the
96 model using a 100 m long row of parabolic trough solar collectors. Comparison of the
97 predicted outlet temperature, energy output and thermal efficiency with measured data
98 shows that the model accurately predicts the transient thermal performance for both heating
99 and cooling processes.

100 A thermal economical model called Solar Advisor Model (SAM) was developed by NREL and
101 Sandia National Laboratory [13,14]. This model calculates costs, finances, and performance
102 of concentrating solar power and also allows examining the impact of variation in the
103 physical parameters on the costs and overall performance. A more recent methodology for
104 the economic optimization of the solar area in either parabolic trough or complex solar plants
105 was carried out by Montes et al. [15]. An analytic model for a solar thermal electric
106 generating system with parabolic trough collectors was carried out by Rolim et al. [16]. A
107 simplified numerical methodology for designing of parabolic trough solar energy systems
108 was proposed by Stine and Harrigan [17]. This method proposed a design chart called
109 storage sizing graph, which obtains the optimum collector area for certain location.

110 Edenburn [18] predicted the efficiency of a PTC by using an analytical heat transfer model
111 for evacuated and non evacuated cases. The results showed good agreement with
112 measured data obtained from Sandia National Laboratories collector test facility [19]. Ratzel
113 et al. [20] carried out both analytical and numerical study of the heat conduction and
114 convective losses in an annular receiver for different geometries. Three techniques were
115 proposed to reduce the conduction heat loss: evacuation, oversizing the annular receiver
116 while keeping the Rayleigh number below 1000 over the range of operation and use of
117 gases with low thermal conductivity. Dudley et al. [21] developed an analytical model of

118 SEGS LS-2 parabolic solar collector. The thermal loss model for the heat collection element
119 was one dimensional steady state model based on thermal resistance analysis.

120 Stuetzle [22] proposed an unsteady state analysis of solar collector receiver to calculate the
121 collector field outlet temperature. The results obtained showed that the overall match
122 between the calculated and measured outlet temperatures was good. García-Valladares and
123 Velásquez [23] developed a detailed numerical model for 51a single pass solar receiver and
124 validated it. Their results showed that the proposed configuration enhances the thermal
125 efficiency of the solar collector compared with the single pass. Heidemann et al. [24]
126 formulated a two dimensional heat transfer model for calculating the absorber wall
127 temperature of a parabolic trough collector under steady and unsteady conditions. The
128 numerical solution showed that a sudden drop of irradiation induces a very high temperature
129 gradient inside the absorber tube in a short period of time. Odeh et al. [25] carried out a
130 model for heat loss of parabolic trough collector in terms of wall temperature rather than
131 working fluid temperature. The results were compared with the Sandia test data and the
132 results showed that the model underestimates the measured loss. Recently, three
133 dimensional heat transfer analysis of PTCs was performed by combining the Monte Carlo
134 Ray Trace Method (MCRT) and CFD analysis [26-28]. The results indicated that the support
135 bracket and bellow under non-vacuum conditions bring a high conductive heat loss.

136 The foregoing literature review shows that a lot of studies have been carried out to model the
137 performance of parabolic trough collector. However, the existing models published
138 previously assume that there is no thermal interaction between the neighboring surfaces
139 (absorber-envelope, and envelope-envelope) for thermal radiation losses. Although these
140 assumptions simplify the analysis, it underestimates the radiation losses at high absorber
141 temperatures. To account for the thermal interaction between adjacent surfaces, a
142 comprehensive radiative analysis was implemented in the present study for the heat losses
143 in the absorber and the glass envelope. A review of the correlations for convective heat
144 transfer losses was performed as well and new correlations were used in the present model.
145 The effects of heat conduction in the collector tube wall and the mixed convection in the
146 inner tube which have been neglected in the previous studies are also taken into account in
147 the present developed model. The present model can predict the effects of annuli space,
148 tube diameter ratio and heat conductivity coefficient of inner tube wall on the heat transfer
149 performance. Parametric study are carried out for different: collector absorber areas, overall
150 heat loss coefficients, optical losses, collector inlet temperatures, mass flow rates of the heat
151 transfer fluid, angles of incidence and concentration ratios. This will certainly lead to the
152 optimum parabolic trough performance as well as the maximum annual energy yield under
153 Kuwait climate conditions.

154 In addition a case study is carried out adapting parabolic trough collectors to satisfy nominal
155 space heating load, water heating load and cooling load of a typical Kuwaiti house as a
156 model for existing buildings that can be moved toward nearly net zero energy buildings in
157 Kuwait. Finally, the environmental impact of solar heating and cooling systems under Kuwait
158 climate conditions is investigated.

159

160 **2. THEORETICAL ANALYSIS**

161

162 **2.1 Solar Radiation**

163 The prediction of collector performance requires knowledge of the absorbed solar energy by
164 the collector absorber plate. The solar energy incident on a tilted collector consists of three
165 different components: beam radiation, diffuse radiation, and ground-reflected radiation. The
166 details of the calculation depend on which diffuse-sky model is used. Using isotropic sky
167 model, the absorbed radiation on the absorber plate can be given by:

$$G = G_b R_b (\tau\alpha)_b + G_d (\tau\alpha)_d \left\{ \frac{1 + \cos\beta}{2} \right\} + (G_b + G_d) (\tau\alpha)_g \rho_g \left\{ \frac{1 + \cos\beta}{2} \right\} \quad (1)$$

where the subscripts b, d, and g represent beam, diffuse, and ground-reflected radiation, respectively. G is intensity of radiation on a horizontal surface, $(\tau\alpha)$ the transmittance-absorptance product that represents the effective absorptance of the cover-plate system, and β the collector slope, ρ_g is the diffuse reflectance of ground and the geometric factor R_b is the ratio of beam radiation on the tilted surface to that on a horizontal surface. The angle of incidence (θ) represents the angle between the beam radiation and the plane normal to the surface. In addition to losses due to the angle of incidence, there are other losses from the collectors that can be correlated to the angle of incidence. These losses occur due to additional reflection and absorption by the glass envelope when the angle of incidence increases. The incidence angle modifier (IAM) corrects for these additional reflection and absorption losses.

The calculation can be simplified by defining equivalent angles that give same transmittance as for diffuse and ground-reflected radiation. Performing the integration of the transmittance over the appropriate incident angle with an isotropic sky model and suggested the equivalent angle of incidence for diffuse radiation:

$$\theta_{d,e} = 59.7 - 0.1388\beta + 0.001497\beta^2 \quad (2)$$

where β is the tilted angle of solar collector. For ground-reflected radiation, the equivalent angle of incidence is given by:

$$\theta_{g,e} = 90 - 0.5788\beta + 0.002693\beta^2 \quad (3)$$

2.2 Parabolic Trough Collector

A performance model is developed for solar thermal collector based on a linear, tracking parabolic trough reflector focused on a surface-treated metallic pipe receiver enclosed in an evacuated transparent tube. The developed PTSC model is a steady-state model and based on the energy balance and heat transfer mechanism within PTSC's receiver tube. It could calculate the fraction of the incident solar energy recovered in the fluid circulating through the pipe receiver, the detail thermal losses from collector, the pressure drop of that fluid over the receiver, and the temperatures of various collector components and of the fluid throughout the receiver under different conditions. The model could be used for optimizing PTSC design by varying the size of receiver tube, the size of glass envelope, receiver tube material, coating material and so forth. The present model also considers the effects of fluid properties, ambient conditions, and operating conditions on the performance of the collector.

The heat collection element (HCE) consists of an absorber surrounded by a glass envelope. The absorber is typically stainless steel tube with a selective absorber surface which provides the required optical and radiative properties. Selective surfaces combine a high absorptance for solar radiation with low emittance for the temperature range in which the surface emits radiation. The glass envelope is an anti reflective evacuated glass tube which protects the absorber from degradation and reduces the heat losses. The vacuum enclosure is used primarily to reduce heat losses at high operating temperatures and to protect the

212 solar-selective absorber surface from oxidation. The heat transfer model is based on an
213 energy balance between the heat transfer fluid and the surroundings.
214

215 The solar energy reflected by the mirrors is absorbed by the glass envelope mainly $\dot{Q}_{e-abs}$
216 and the absorber surface $\dot{Q}_{a-abs}$. A part of the energy absorbed in the absorber is transferred
217 to the heat transfer fluid by forced convection $V_{a-f;conv}$, remaining energy is transferred back
218 to the glass envelope by radiation $\dot{Q}_{a-e;rad}$ and natural convection $\dot{Q}_{a-e;conv}$ and lost through the
219 support brackets by conduction $\dot{Q}_{cond;bracket}$ as well. The heat loss coming from the absorber
220 (radiation and natural convection) passes through the glass envelope by conduction and
221 along with the energy absorbed by the glass envelope $\dot{Q}_{e-abs}$ is lost to the environment by
222 convection $\dot{Q}_{e-sa;conv}$ and to sky by radiation $\dot{Q}_{e-s;rad}$. In order to obtain the partial differential
223 equations that govern the heat transfer phenomena, an energy balance is applied over a
224 section of the solar receiver. After applying the energy balance on a control volume,
225 assuming unsteady state and incompressible fluid, the following partial differential equation
226 can be obtained:

$$227 \quad A_{i,a} \rho_f C_{p,f} \frac{\delta T_f}{\delta t} = \dot{m}_f \frac{\delta}{\delta z} \left(C_{p,f} T_f + \frac{V_f^2}{2} \right) + \dot{Q}_{a-f,conv} \quad (4)$$

$$228 \quad V_f = \frac{\dot{m}_f}{\rho_f A_{i,a}} \quad (5)$$

229 where $A_{i,a}$ is the internal cross sectional area of the absorber, and $\dot{Q}_{a-f,conv}$ is the heat
230 transfer by convection from the absorber to HTF per unit length and is given by:
231

$$232 \quad \dot{Q}'_{a-f,conv} = \pi Nu_f k_f (T_a - T_f) \quad (6)$$

233 where Nu_f is the Nusselt number, k_f is the thermal conductivity of HTF, and T_a is the absorber
234 wall temperature. The correlation for fully developed turbulent flow ($Re_D > 2300$) convective
235 heat transfer in circular ducts [29] is adapted:
236
237

$$238 \quad Nu_{uf} = \frac{(C_f/2)(Re_D - 1000)Pr}{1 + 12.7(C_f/2)^{1/2}(Pr^{2/3} - 1)} \left(\frac{Pr}{Pr_w} \right)^{0.11} \quad (7)$$

$$239 \quad Re_D = \frac{V_f D_{i,a}}{v_f} \quad (8)$$

240 where C_f is the friction coefficient (Fanning friction factor), Re_D is the Reynolds number
241 based on the absorber inner pipe diameter, Pr is the Prandtl number, and v_f is the kinematic
242 viscosity of HTF. This correlation is valid for $2300 \leq Re_D \leq 5 \times 10^6$ and $0.5 \leq Pr \leq 2000$. The
243 thermal properties should be evaluated at the bulk mean heat transfer fluid temperature,
244 except Pr_w which is evaluated at the absorber wall temperature. The convection heat
245 transfer coefficient for rough tubes can be calculated approximately by using the friction
246 coefficient determined from the Colebrook and White equation [30]:
247

$$248 \quad \frac{1}{\sqrt{C_f}} = 3.48 - 1.7372 \ln \left(\frac{2\varepsilon}{D_{i,a}} + \frac{9.35}{Re_D \sqrt{C_f}} \right) \quad 5 \leq Re_\varepsilon \leq 70 \quad (9)$$

$$\text{With } Re_\varepsilon = \frac{V_f \varepsilon}{\nu_f} \quad (10)$$

For laminar flow, ($Re_D \leq 2300$), the Nusselt number on walls with uniform temperature is given by $Nu_f = 3.66$. Two heat transfer mechanisms occur between the absorber and the glass envelope: convection heat transfer and thermal radiation. Convection heat transfer depends on the annulus pressure; experimental work has shown that heat transfer losses are independent of the annulus vacuum pressure for pressures above 1 Torr. At pressures below 1 Torr, molecular conduction is the heat transfer mechanism while for pressure above 1 Torr, natural convection takes place. Applying the energy balance on control volume:

$$A_a \rho_a C_{p,a} \frac{\delta T_a}{\delta t} = A_a \frac{\delta}{\delta z} \left(k_a \frac{\delta T_a}{\delta z} \right) + \dot{Q}'_{a,abs} - \dot{Q}'_{a-f,conv} - \dot{Q}'_{a-e,conv} - \dot{Q}'_{a-e,rad} - \dot{Q}'_{cond,bracket} \quad (11)$$

where A_a is the cross sectional area of the absorber, $A_a = (\pi/4)(D_{o,a}^2 - D_{i,a}^2)$, $\dot{Q}'_{a,abs}$ is the solar absorption in the absorber per receiver length, $\dot{Q}'_{a-f,conv}$ is the heat transfer by convection from absorber to heat transfer fluid per unit length, $\dot{Q}'_{a-e,conv}$ is the heat transfer by convection from absorber to envelope per unit length, $\dot{Q}'_{a-e,rad}$ is the heat transfer by radiation from absorber to glass envelope per unit length, and $\dot{Q}'_{cond,bracket}$ is the heat conduction through support brackets per unit length. Stainless steel is normally used as the absorber tube material. Thermal radiation analysis for one surface implies that all surfaces that can exchange radiative energy with the surface must be considered simultaneously. How much energy two surfaces exchange depends on their size, separation distance, and orientation. In order to carry out the radiative heat transfer analysis, the view factors for a short annulus proposed by Siegel and Howell [31] are employed. The heat transfer from the glass envelope to the surroundings is by convection and radiation. Convection heat transfer has two cases: with wind (forced convection) and no wind (natural convection). The radiation heat transfer is basically between the glass envelope and either the sky or the collector surface, but the maximum radiation heat losses take place when the solar receiver is assumed to be surrounded only by the sky surface. The energy balance on a control volume leads to the following partial differential equation (PDE):

$$A_e \rho_e C_{p,e} \frac{\delta T_e}{\delta t} = A_e \frac{\delta}{\delta z} \left(k_e \frac{\delta T_e}{\delta z} \right) + \dot{Q}'_{e,abs} + \dot{Q}'_{a-e,conv} + \dot{Q}'_{a-e,rad} - \dot{Q}'_{e-sa,conv} - \dot{Q}'_{e-s,rad} \quad (12)$$

where $\dot{Q}'_{e,abs}$ is the solar absorption in the envelope per receiver length, $\dot{Q}'_{a-e,conv}$ is the heat transfer by convection from the glass envelope to the surrounding air per unit length, and $\dot{Q}'_{e-s,rad}$ is the heat transfer by radiation from the glass envelope to the sky per unit length. The heat transfer by convection per unit length from the glass envelope to the surrounding air is calculated as:

$$\dot{Q}'_{e-sa,conv} = h_e \pi D_{o,e} (T_e - T_\infty) \quad (13)$$

$$\text{with } h_e = \frac{Nu_e k_e}{D_{o,e}} \quad (14)$$

For no wind conditions, the following expression [30] is recommended for horizontal cylinder under natural convection:

$$Nu_e = \left[0.60 + 0.387 \left\{ \frac{Ra_D}{[1 + (0.559/Pr)^{9/16}]^{6/9}} \right\}^{1/6} \right]^2 \quad (15)$$

The solar receiver is surrounded by either the collector or the sky. In order to simplify the model, it is assumed that half of the receiver surface is surrounded by the mirror and the other half by the sky. The heat flux and radiosity are calculated for each surface and a set of energy balance equations for internal and external surfaces are written. In this work, it is assumed that the collector mirror temperature is approximately the ambient temperature. For the case when the collector surface is not included in the analysis (maximum heat transfer losses by radiation), which means that glass envelope is assumed to be totally covered by the sky surface. So, the heat flux for the area of analysis is expressed as:

$$\dot{q}_{esi} = \sigma \epsilon_{esi} (T_{esi}^4 - T_{sky}^4) \quad (16)$$

Several relations have been proposed to relate T_{sky} , for clear skies, to other measured meteorological variables. In the absence of meteorological data such as: relative humidity, dew point temperature, etc, the simple relation given by Swinbank [32] can be used:

$$T_{sky} = 0.0553 T_{\infty}^{1.5} \quad (17)$$

The energy absorbed in the solar receiver is affected by the optical properties and imperfections of the solar collector ensemble. For a concentrating collector, the effective optical efficiency is defined as long as the direct beam radiation is normal to the collector aperture area. When the beam radiation is not normal, the incident angle modifier (IAM) is included to account for all optical and geometric losses due to an incident angle greater than 0°. The IAM depends on the geometry and the optical characteristics of the solar collector.

3. NUMERICAL MODEL

The previous partial differential equations (PDE) were discretized for steady state conditions by using the finite difference method and taking into account the dependence of the thermal properties on the temperature. For the heat transfer fluid, the discretization by using backward difference gives a set of algebraic equations. For the absorber and the envelope, the discretization is carried out by using central difference and another set of algebraic equations are obtained. Finally, the boundary conditions for each element are set down. The present model takes into consideration the thermal interaction between absorber-envelope, and envelope-envelope for thermal radiation losses which have been neglected in the existing models. To account for the thermal interaction between adjacent surfaces, a comprehensive radiative analysis was implemented in the present study for the heat losses in the absorber and the glass envelope. A review of the correlations for convective heat

transfer losses was performed as well and new correlations were are in the present model. The effects of heat conduction in the collector tube wall and the mixed convection in the inner tube which have been neglected in the previous studies are also taken into account in the present developed model. The resulting set of nonlinear algebraic equations is solved simultaneously using numerical techniques. Collector thermal efficiency (η) is defined as the ratio of energy collected by the working fluid to the direct normal solar radiation incident upon the collector aperture. It is typically determined by testing a collector over a range of high temperatures and is expressed as:

$$\eta_c = \frac{\dot{m}(h_{o,f} - h_{i,f})}{I_b A_{ap}} \quad (18)$$

The thermal losses from the collector receiver are functions of operating temperature. The thermal losses through PTSC are changed in different ways depending on the receiver's configuration and operational conditions. In general, the thermal losses are given by:

$$\text{Thermal loss} = \left[\sum_i (\dot{Q}'_{a-e,conv})_i + \sum_i (\dot{Q}'_{a-e,rad})_i + \sum_j (\dot{Q}'_{cond,bracket})_j \right] \frac{\Delta z}{I_b A_{ap}} \quad (19)$$

A simulation model compatible with the transient simulation program (TRNSYS) is developed to determine the thermal performance of a typical parabolic trough configuration under Kuwait climate conditions. A TRNSYS studio project is designed by setting up the connecting components graphically in the Simulation Studio [33]. The parameters studied are: area of the absorber of the collector, overall heat loss coefficient from the absorber, reflectivity of the reflecting surface, absorptivity of the absorber, emissivity of the absorber. The parametric study has been made for different mass flow rates and concentration ratios using hourly solar radiation data for Kuwait. The processed solar insulation data consists of the beam radiation and diffuse radiation as the total radiation on a tilted surface.

4. RESULTS AND DISCUSSIONS

In order to validate the present heat transfer model, it was first compared with experimental data obtained from Sandia National Laboratory [34]. In addition, to corroborate the improvement in the correlations and radiation analysis proposed in this work, the numerical model was also compared with other solar receiver heat transfer models [35]. The experimental results were for a solar collector assembly (LS-2) module placed at the AZTRAK rotating platform at the SNL. Two different selective coatings were used in this test: black chrome and cermet. Cermet has better radiative properties (low emissivity) at high temperatures than black chrome, and does not oxidize if the vacuum is lost. The Sandia test was performed for both full sun and no sun condition and different scenarios for the annulus of the heat collection element (HCE): vacuum intact (the evacuated annulus pressure was 10^{-4} Torr), lost vacuum (annulus filled with ambient air), and glass cover completely removed (bare tube). Figure 1 presents the results obtained for the collector efficiency. As seen from figure, the current calculated collector efficiency follows the trends of the experimental values and all the results are always inside the experimental error bars. As it was expected, the higher efficiencies are obtained when the annulus is under vacuum, but in both cases (air and vacuum) at high temperatures the collector efficiency drops gradually, which is more notable for the black chrome coating.

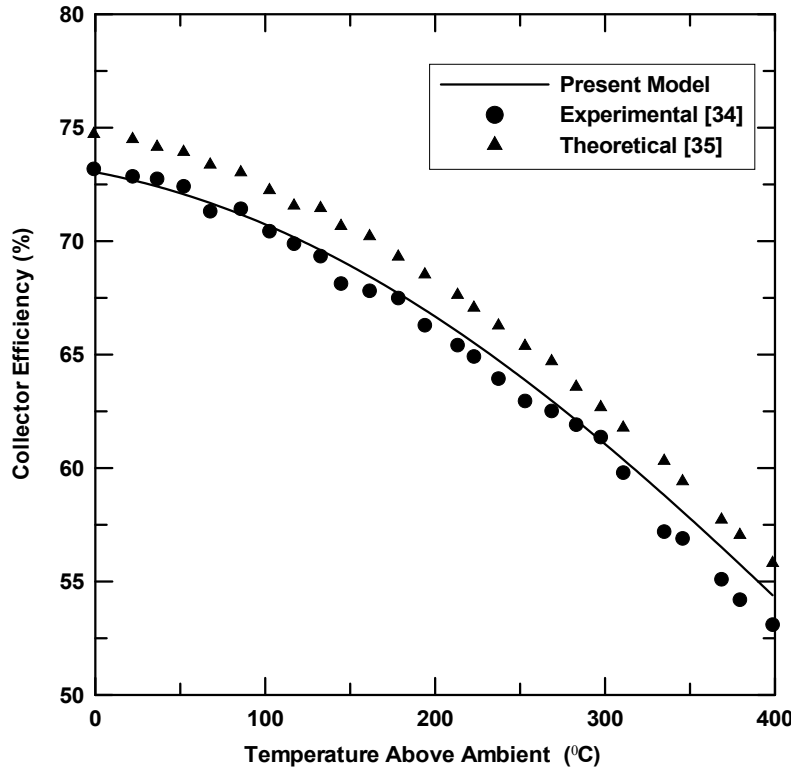


Fig. 1. Comparison of collector efficiency calculated from the present model with experimental data [34] and NREL model [35]

For the case of Cermet coating, the model developed by Garcia-Valladares and Velasquez [35] shows some discrepancies at low temperatures in the collector efficiency due to their assumptions: negligible conduction at the ends of each trough and only radiation heat losses take place between the receiver and the glass envelope for the case of vacuum in the annular space. Those assumptions were not made in the present model or NREL model. The present model and NREL model [34] give almost similar collector efficiencies values. The incidence angle modifier, $K_{\alpha\tau}(\theta_i)$, enables the performance of the collector to be predicted for solar angles of incidence other than 0° (normal). Simulations using the present numerical model are carried out at a set value of θ_i and then calculating $K_{\alpha\tau}$. It should be mentioned that the incident angle modifier represents the ratio between thermal efficiency values at a specified value and the peak efficiency of the collector at zero incidence. Results of simulation are presented in Figure 2. Regression analyses provided the following equations for $K_{\alpha\tau}$ as a function of θ_i :

$$K_{\alpha\tau U} = -2.235 \times 10^{-6}(\theta_i)^3 + 1.2146 \times 10^{-4}(\theta_i)^2 - 3.884 \times 10^{-3}(\theta_i) + 1.125 \quad (20)$$

$$K_{\alpha\tau G} = 9.272 \times 10^{-7}(\theta_i)^3 - 1.561 \times 10^{-4}(\theta_i)^2 + 1.773 \times 10^{-3}(\theta_i) + 1.078 \quad (21)$$

The coefficients of determination for Equations (20) and (21) are 0.958 and 0.961. Up to an incidence angle of approximately 25° the glass-shielded receiver performed slightly better, but beyond that its performance declined more rapidly and was inferior to that of the unshielded receiver. From Figure 3, the calculated value of $K_{\alpha\tau}$ was 0.74 for the glass-shielded receiver at the maximum tested incidence angle of 54° . This was 8.9% lower than for the unshielded receiver. At the same maximum angle of incidence, the simplified cosine

model under-predicted K_{gr} by about 25% for the unshielded receiver and by 17.7% for the glass shielded unit. Two factors are primarily responsible for the decline in performance of a PTSC with increasing θ_i : the geometric reduction in irradiance falling on the aperture as θ_i increases, called the 'cosine effect' and the change in optical efficiency as light interacts differently with the reflective surface of the collector, the glass shield (if present) and the absorber. Nothing can be done to counter the first effect apart from tilting or rotating the PTSC constantly so as to keep it oriented perpendicular to the sun's rays.

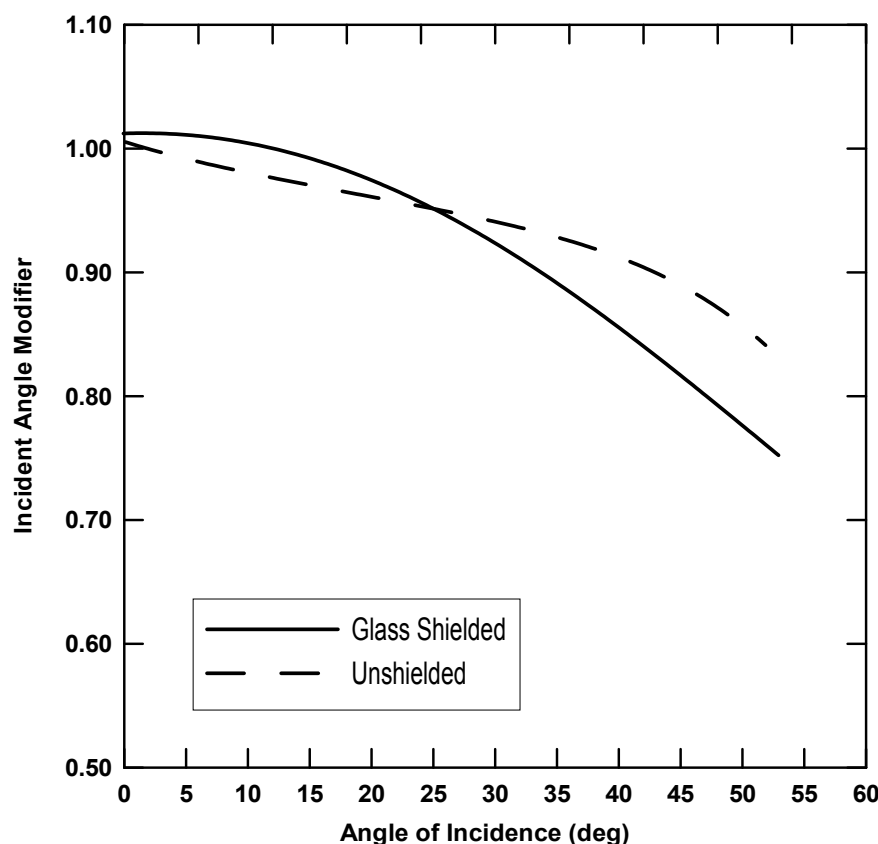


Fig. 2. Incidence angle modifier for unshielded and glass-shielded receivers

The thermal losses from the collector receiver are functions of operating temperature. The thermal losses through PTSC are changed in different ways depending on the receiver's configuration and operational conditions. As shown in Figure 3, among three losses: conduction, convection and radiation, the convection loss from absorber tube to supporting structures is the largest; the next is the radiation loss from glass envelope to ambient air; the relative smaller is the convection loss from glass envelope to ambient air. The thermal losses are always existing (if there is a temperature difference between receiver and ambient) whenever solar radiation is available or not.

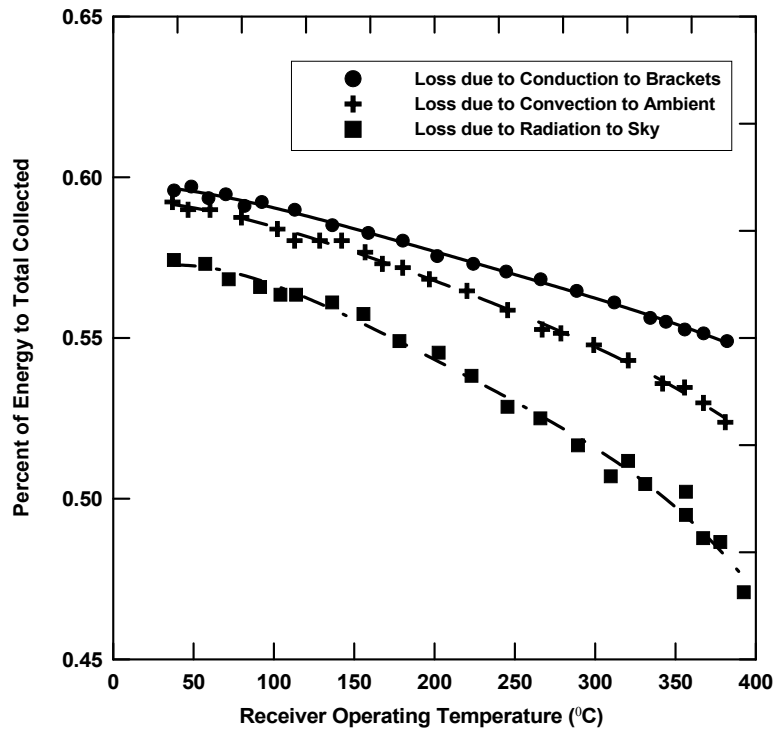


Fig. 3. The heat losses of an absorber tube

Increasing solar radiation results in increasing the solar energy absorbed by the collector. It should be noted that the thermal losses also increases due to the increased collector temperature. However this increase is much smaller than the enhanced absorbed solar energy. As shown in Figure 4, PSTC efficiency increases with increasing in solar radiation.

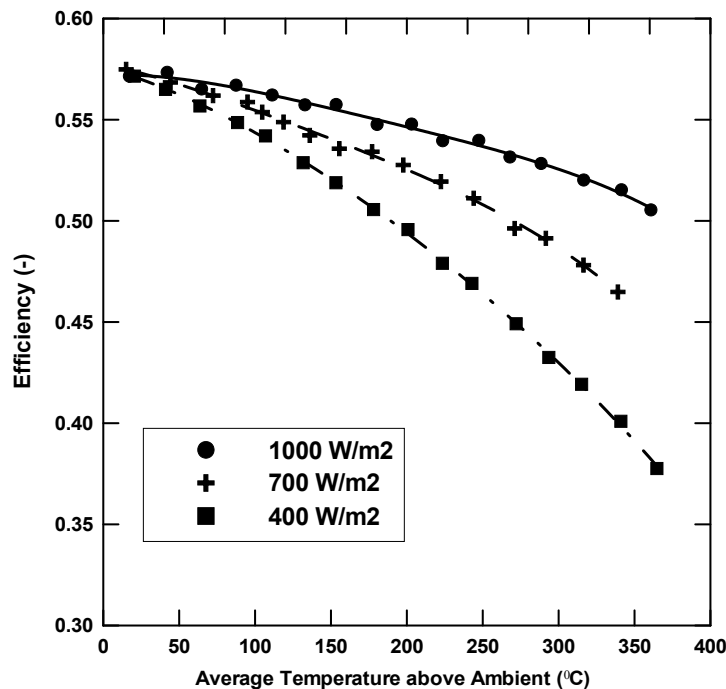
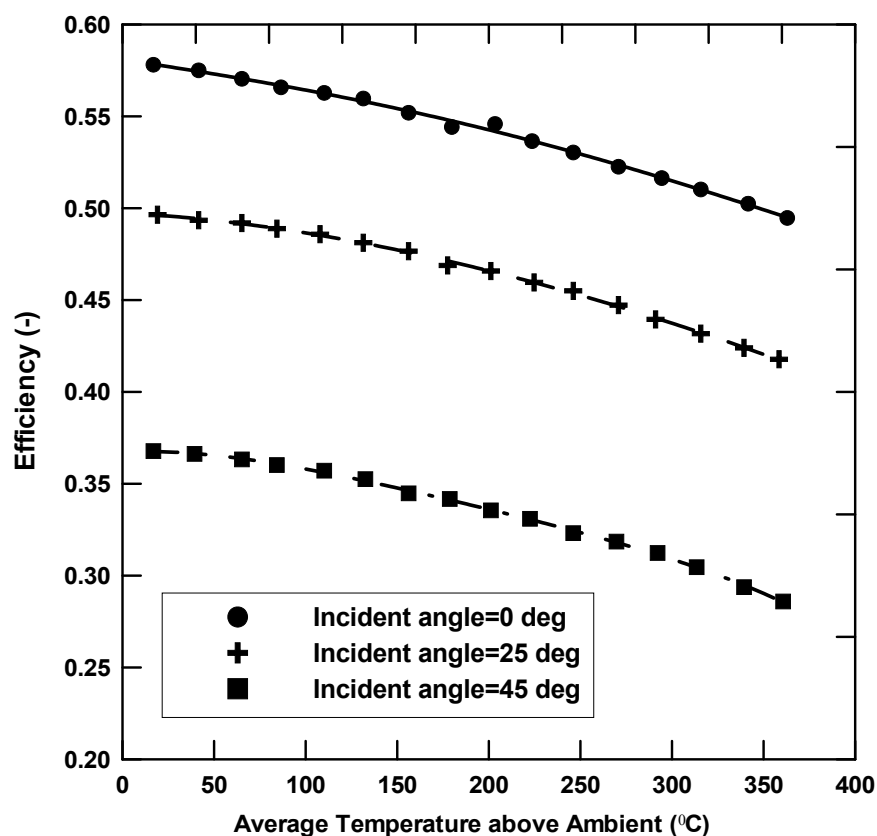


Fig. 4. Variation of parabolic trough collector efficiency with solar radiation

434 The incident angle modifier is a very important factor impacting on the solar efficiency. The
 435 incident angle modifier is defined as the ration of the transmittance-absorptance product for
 436 angle of the incidence of radiation and for the normal incidence radiation and it is
 437 proximately equal to the cosine of the incident angle. It changes depending on the time of
 438 day, the day of the year, the location and orientation of the aperture, and whether it is
 439 stationary or tracks the sun movement about one or two axes. Collector efficiency is at a
 440 maximum only when the incident angle is zero. The efficiency of a PTSC decreases when
 441 the solar beam incident angle increases shown in Figure 5. The end effect of incident angle
 442 is to reduce radiation arriving at the absorber tube. Therefore at noon time PTSC has the
 443 smallest incident angle and the highest efficiency.
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Fig. 5. Variation of collector efficiency with incident angle

When a vacuum is present in the annulus between the receiver surface and the glass envelope, the conduction and the convection across the annulus are effectively eliminated. Once air is introduced into the vacuum space, measured losses increase significantly since conduction and convection begin to transfer heat to the glass envelope as shown in Figure 6. Radiation loss from the heated receiver's metal surface to the glass envelope is not changed very much by the presence of air in the annulus.

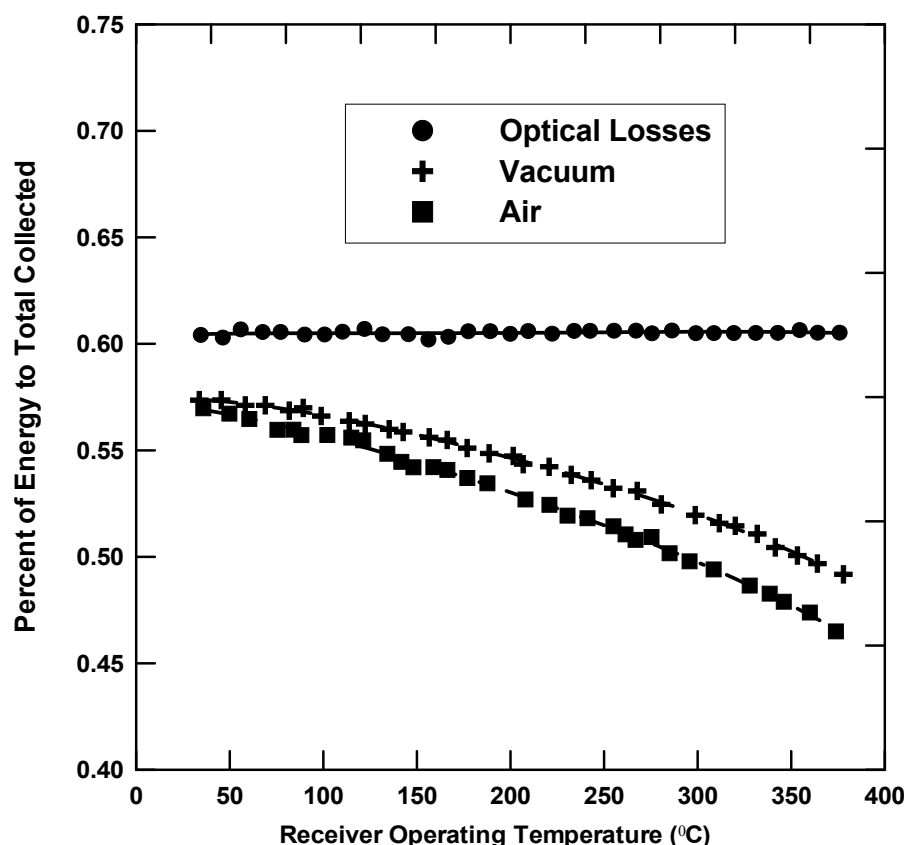


Fig. 6. Collector efficiency and air in annular

5. CASE STUDY

A case study is carried out to investigate the performance of solar heating and cooling system in Kuwait climate. The solar system is designed to satisfy a significant part of nominal space heating load, water heating load and cooling load of a typical Kuwaiti house. A schematic diagram of the system studied is represented in Figure 7. The system consists of parabolic trough collectors integrated in the roof of the building, a storage tank, an absorption chiller, heat exchanger, and auxiliary units. The main purpose of this state is to investigate the feasibility of integrating parabolic trough collector in an existing building in an attempt to convert it to nearly net zero energy building. Buildings are typically responsible for a significant part of the total primary energy consumption. So, it is the aim goal of researcher and building designers to develop efficient buildings that can use the energy produced from renewable energy sources. Recently many new and existing buildings incorporate solar energy technologies to attain environmental benefits in comparison to the conventional energy in terms of reducing global warming and greenhouse gases emissions, mainly CO₂, and preventing toxic gas emissions [36-39]. The system operates in four different modes. When solar energy is available for collection and there is a load demand, heat is supplied directly from the collector to the heating or cooling unit. When solar energy is available for collection and there is no heat or cooling demand, heat is stored in the storage unit. On the other hand, if solar energy is not available for collection and there is a load demand, storage then supplies heat to the heating or cooling unit. However, if storage temperature is not sufficient, the heating or cooling load is supplied by the auxiliary source.

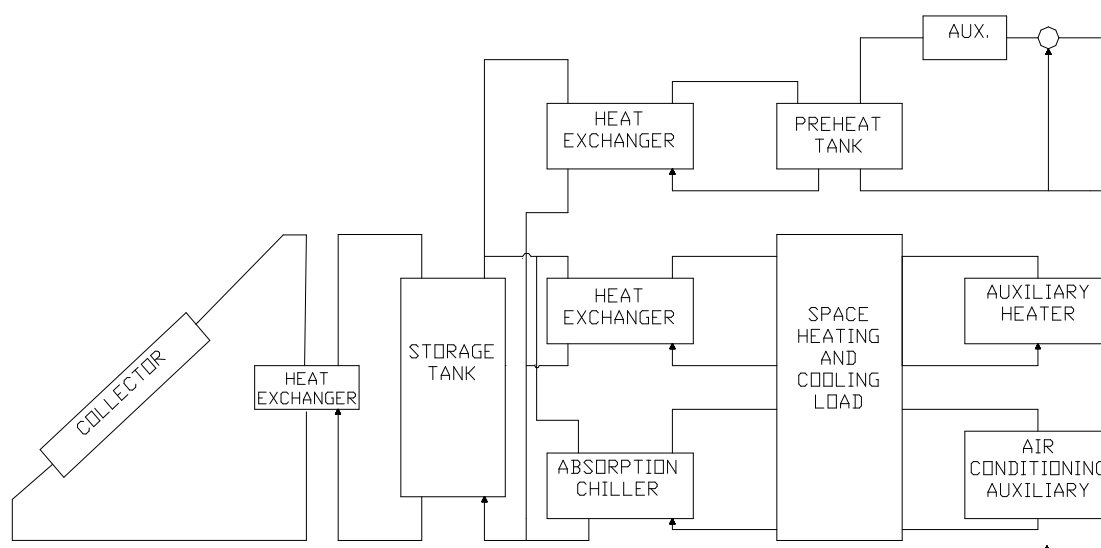


Fig. 7. Schematic diagram of the solar heating and cooling system

The solar cooling system consists of single-effect lithium bromide water absorption chiller. The chiller model is based on a commercially available LiBr-H₂O absorption chiller system, Arkla model WF-36. The Arkla chiller has a nominal cooling capacity of three tons (37980 kJ/h). Units of different capacity are approximated by scaling the Arkla performance. Hot water is supplied to the air conditioner at a temperature of 87°C (minimum), 93°C (maximum) and leaves this unit 10°C cooler than the supply and returns to the storage (or to the auxiliary heater if storage is below 77°C). Whenever hot water from storage is cooler than 87°C, the auxiliary heat is supplied to raise its temperature to 87°C. When storage is cooler than 77°C, it is not used, and the auxiliary heater carries the full cooling load. The performance of air conditioning systems is expressed by their coefficient of performance (COP). COP determines how many units of cooling/heating one gets for every unit of energy he puts. In addition, a constant COP model of a vapor compression air conditioner is included as a secondary cooling auxiliary so that the energy required to meet space cooling is provided even if the absorption machine cannot meet the full load.

5.1 Building load

The building studied is one of the typical Kuwaiti houses in a remote area. The computer program is provided with an estimate of typical loads encountered in the house. The two-story house consists of two living rooms, five bedrooms, dining room and two kitchens. An hourly load schedule, which repeats each day, is written to a file to be called during calculation. The program is provided with an estimation of the number of hours that the load is used during a typical day for each load. The building roof area is large enough so that the parabolic trough collectors can be spaced widely to minimize shading losses. The domestic water demand is based on a figure of 60 kg/day/person. The system has been designed to supply the cooling load of a typical Kuwaiti house where the cooling load peak is about 21 kW (6 tons of refrigeration), which occurs in August.

Different collector parameters and operating conditions are varied to maximize the annual energy generated by the parabolic trough collector. For building simulation, type 56 included in version 16 of TRNSYS [33] is employed. This component models the thermal behavior of

building having multiple thermal zones. The building description is ready by this component from a set of external files. The files can be generated based on user supplied information by running the preprocessor program called TRNBuild. From the simulation done with the TRNbuild (Type 56) of TRNSYS to determine the demand for air conditioning in the house under study, we obtained results that indicate that this demand starts from April to October, with critical periods for the months of June, July and August in which occurs the maximum load. In this study, the air conditioning system of the house consumes about 83% of the total building energy consumption. The remaining energy consumption is distributed between space heating 11%, and domestic water heating, 6%. For hot thermal storage, a stratified liquid storage tank, with two inlet and two outlet flows (type 60) is adapted. Several data files of absorption chillers of single-effect employing Li Br – H₂O solution as working fluid, (type 107) according to Yazaki chiller WFC SH10 are used. Other types used were: Type 15-2-TMY2, type 3b for simulation of pumps.

5.2 Solar Fraction

The thermal performance of a solar system is usually measured by the solar fraction (F). Solar fraction is defined as the fraction of load met by solar energy. Figure 8 shows the variation of the solar fraction of space heating (F_s), domestic water heating (F_D), and cooling load (F_{AC}) with collector area. As seen from the figure, the solar space heating load and domestic water heating load is completely satisfied at areas around 29 m². Conversely, the space cooling load requires much greater areas. The minimum required collector area is about 82 m² which can supply the cooling load of a typical residential house for sunshine hours in all hot climate conditions of Kuwait where maximum cooling load is within 21 KW (6 ton refrigeration).

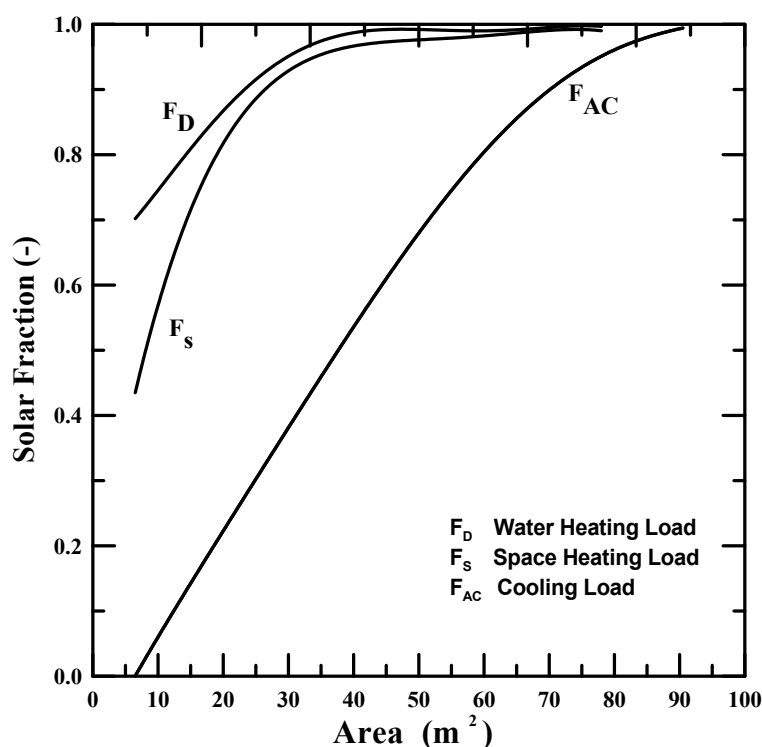
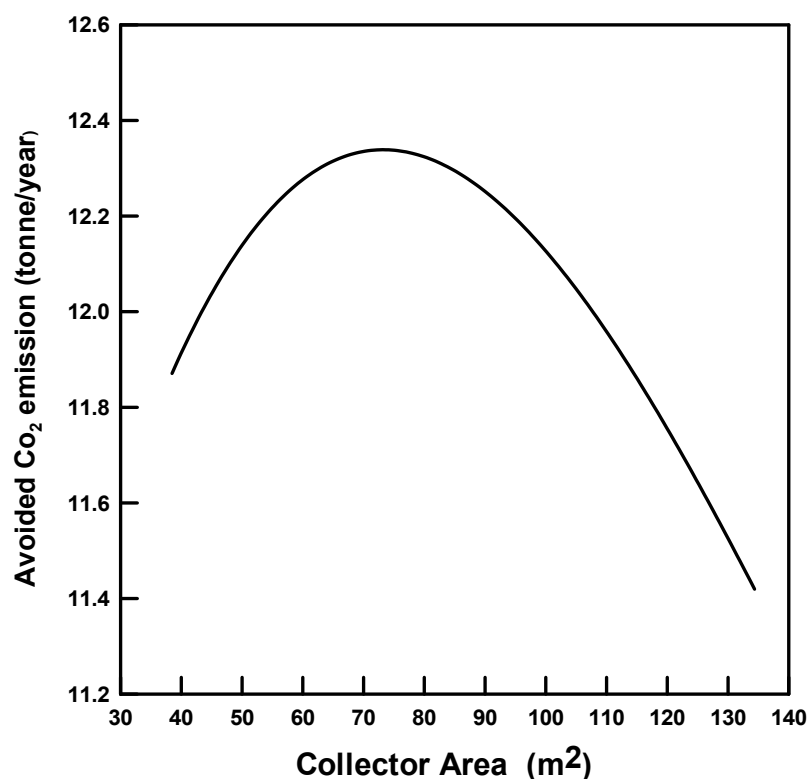


Fig. 8 . Solar fraction variation with collector area

546 The coefficient of performance (COP) of the absorption chiller is approximately 0.66 which is
 547 within the accepted practical values of the conventional lithium bromide system.
 548 A program is written to determine the annual avoided CO₂ emission as a result of integrating
 549 parabolic trough collectors at the house roof. The variation of annual avoided CO₂ emission
 550 with collector area is presented in Figure 9. As shown from the figure at the required
 551 collector area to satisfy the house cooling load, the avoided CO₂ emission approximately
 552 equals to 12.3 tonne/year. In addition, the costs due to the application of the Kyoto Protocol,
 553 which penalizes the emissions of green house effect gases should be added to the costs of
 554 conventional energy resources. Considering application of this protocol will enhance the
 555 economical and environmental aspects of solar systems in Kuwait much more. The present
 556 results should encourage governments for wide installation of solar heating and cooling
 557 systems in different applications which will significantly reduce energy consumption of
 558 conventional fuel as well as reducing environmental pollution to keep our community healthy
 559 and clean.
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 563 **Fig. 9. Avoided CO₂ emission variation with collector area (azimuth angle=0)**
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565 6. CONCLUSIONS

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 567 This work investigates the performance of parabolic trough collector as well as solar heating
 568 and cooling systems in Kuwait climate. A theoretical model takes into consideration the
 569 factors ignored in previous models is developed. Based on the present results, the following
 570 conclusions can be drawn:

- 571 • The performance of parabolic trough collector can be significantly enhanced by
- 572 optimizing its parameters as well as operating conditions.
- 573 • Convection loss from absorber tube to supporting structures is the largest among
- 574 the other losses (conduction and radiation).

- 575 • The incident angle modifier is a very important factor impacting on the solar
576 efficiency.
- 577 • At noon time PTSC has the smallest incident angle and the highest efficiency.
- 578 • When a vacuum is present in the annulus between the receiver surface and the
579 glass envelope, the conduction and the convection across the annulus are
580 effectively eliminated.
- 581 • Space heating load and domestic water heating load can be completely provided by
582 parabolic trough collectors.
- 583 • The minimum required parabolic trough collector area is about 82 m² to supply
584 cooling loads of a typical residential house in all hot climate conditions of Kuwait.
- 585 • Total avoid CO₂ emission of about 12.3 tonne/year can be achieved.
- 586 • The results of the present study should encourage wide utilization of solar energy
587 systems which will help in keeping our environment healthy and clean.
588

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594 **REFERENCES**

- 595 [1] Kalogirou S, Lloyd S, Ward J. Modeling, Optimization and Performance Evaluation of a
596 Parabolic Trough Solar Collector Steam Generation. *Solar Energy*. 1997; 60(1): 49-59.
- 597 [2] Price H, Lufert E, Kearney D, Zarza E, Cohen G, Gee R, Mahoney R. Advances in
598 Parabolic Trough Solar Power Technology. *Journal of Solar Energy Engineering*. 2002;
599 124:109-125.
- 600 [3] Riffelmann K, Neumann A, Ulmer S. Performance Enhancement of Parabolic Trough
601 Collectors by Solar Flux Measurement in the Focal Region. *Solar Energy*. 2006; 80:
602 1303-1313.
- 603 [4] Eck M, Zarza E. Saturated Steam Process with Direct Steam Generating Parabolic
604 Troughs. *Solar Energy*. 2006; 80:1424-1433.
- 605 [5] Mazloumi M, Naghashzadegan M, Javaherdeh K. Simulation Study of Solar lithium
606 bromide-water absorption cooling system with parabolic trough collector. *Energy*
607 *Conversion and Management*. 2008;49:2820-2832.
- 608 [6] Adinberg R, Zvegilsky D, Epstein M. Heat transfer efficient thermal energy storage for
609 steam generation. *Energy Conversion and Management*. 2010;51:9-15.
- 610 [7] Garcí'a A, Zarza E, Valenzuela L, Rez M. Parabolic-Trough Solar Collectors and Their
611 Applications. *Renewable and Sustainable Energy Reviews*. 2010;14:1695-1721.
- 612 [8] Sansoni P, Fontani D, Francini, F, Giannuzzi A, Sani E, Mercatelli L, Jafrancesco D.
613 Optical Collection Efficiency and Orientation of a Solar Trough Medium-Power Plant
614 Installed in Italy. *Renewable Energy*. 2011;36:2341-2347.
- 615 [9] Daniel P, Joshi Y, Das A. Numerical Investigation of Parabolic Trough Receiver
616 Performance with Outer Vacuum Shell. *Solar Energy*. 2011;85:1910-1914.
- 617 [10] Forristall R. Heat Transfer Analysis and Modeling of a Parabolic Trough Solar
618 Receiver Implemented in Engineering Equation Solver. *National Renewable Energy*
619 *Laboratory (NREL), Colorado*;2003.
- 620 [11] Jones S, Pitz-Paal R, Schwarzbözl P, Hohe L, Blair N, and Cable R. TRNSYS
621 Modeling of the SEGS VI Parabolic Trough Solar Electric Generating Sys- tem. 2011;
622 In *Proceedings of Solar Forum: Solar Energy: The Power to Choose*, Washington,
623 DC, USA, p. 405.

- 624 [12] Li Xu, Wang Z, Xin Li, YuanG, Sun F, Lei D. Dynamic test model for the transient
625 thermal performance of parabolic trough solar collectors. *Solar Energy*. 2013;95:65-78.
- 626 [13] Blair N, Mehos M, and Christensen C. Sensitivity of Concentrating Solar Power
627 trough performance, cost, and financing with the Solar Advisor Model. Tech. Rep.
628 NREL/CD-550-42709, NREL;2008.
- 629 [14] National Renewable Energy Laboratory (NREL), Solar Advisor Model CSP
630 Reference Manual for Version 3.0;2009.
- 631 [15] Montes M, Abánades A, Martínez-Val J, and Valdés M. Solar multiple optimization for
632 a solar-only thermal power plant, using oil as heat transfer fluid in the parabolic
633 trough collectors. *Solar Energy*. 2009; 83(12):2165 – 2176.
- 634 [16] Rolim M M, Fraidenraich N, and Tiba C. Analytic modeling of a solar power plant
635 with parabolic linear collectors. *Solar Energy*. 2009;126–133.
- 636 [17] Harrigan R, Handbook for the conceptual design of parabolic trough solar energy
637 systems process heat applications. Tech. rep., NASA STI/Recon Technical Report;
638 1981.
- 639 [18] Edenburn M W. Performance analysis of a cylindrical parabolic focusing collector and
640 comparison with experimental results. *Solar Energy*. 1976;18(5):437 – 444.
- 641 [19] Pope R, and Schimmel. An analysis of linear focused collectors for solar power. In
642 Eighth Intersociety Energy Conversion Engineering Conference , Philadelphia, PA.
643 1973:353–359.
- 644 [20] Ratzel A, Hickox C, and Gartling D. Techniques for reducing thermal conduction and
645 natural convection heat losses in annular receiver geometries. *J Heat Transfer Trans*
646 *ASME*. 1979;101(1):108–113.
- 647 [21] Dudley V, Kolb G, Sloan M, and Kearney D. SEGS LS2 Solar Collector-Test Results.
648 Report of Sandia National Laboratories, SAN94-1884, 1996.
- 649 [27] Thomas A, and Thomas S. Design data for the computation of thermal loss in the
650 receiver of a parabolic trough concentrator. *Energy Conversion and Management*.
651 1994; 35(7): 555–568.
- 652 [22] Stuetzle T. Automatic control of the 30 MWe SEGS VI parabolic trough plant. Master's
653 thesis, University of Wisconsin-Madison, College of Engineering, 2002.
- 654 [23] García-Valladares O, and Velázquez N. Numerical simulation of parabolic trough solar
655 collector: Improvement using counter flow concentric circular heat ex- changers.
656 *International Journal of Heat and Mass Transfer*. 2009; 52(3-4): 597–609.
- 657 [24] Heidemann W, Spindler K, and Hahne E. Steady-state and transient temperature
658 field in the absorber tube of a direct steam generating solar collector. *International*
659 *Journal of Heat and Mass Transfer*. 1992; 35(3):649–657.
- 660 [25] Odeh S, Morrison G, and Behnia M. Thermal analysis of parabolic trough solar
661 collectors for electric power generation. In *Proceedings of ANZSES 34th annual*
662 *conference, Darwin, Australia*. 1996; 460–467.
- 663 [26] Cheng Z, He Y, Xiao J, Tao Y, and Xu R. Three-dimensional numerical study of heat
664 transfer characteristics in the receiver tube of parabolic trough solar collector.
665 *International Communications in Heat and Mass Transfer*. 2010;37(7):782–787.
- 666 [27] He Y, Xiao J, Cheng Z, and Tao Y. A MCRT and FVM coupled simulation method for
667 energy conversion process in parabolic trough solar collector. *Renewable Energy*.
668 2011;36:976–985.

- 669 [28] Gong G, Huang X, Wang J, and Hao M. An optimized model and test of the China's
670 first high temperature parabolic trough solar receiver. Solar Energy. doi:
671 10.1016/j.solener.2010.08.003, 2010.
- 672 [29] Kakaç S, Shah R K, and Aung W. Handbook of Single-Phase Convective Heat
673 Transfer. John Wiley & Sons, New York., 1987.
- 674 [30] Rohsenow W M, Hartnett J P, and Cho Y I. Handbook of Heat Transfer, 3 ed.
675 McGraw-Hill Professional May;1998.
- 676 [31] Siegel R, and Howell, J R. Thermal Radiation Heat Transfer. McGraw-Hill, New York;
677 1971.
- 678 [32] Swinbank W. Long-wave radiation from clear skies. Quarterly Journal of the Royal
679 Meteorological Society. 1963; 89(381): 339–348.
- 680 [33] Klein SA et al. : TRNSYS, : A Transient Simulation Program", Version 16, University of
681 Wisconsin-Madison ;2006.
- 682 [34] Dudley V, Kolb G, Sloan M, Kearney D. 1994. SEGS LS2 Solar Collector-Test Results.
683 Report of Sandia National Laboratories. SAN94-1884;1994.
- 684 [35] Garcí'a-Valladares O, Velazquez N. Numerical simulation of parabolic trough solar
685 collector: improvement using counter flow concentric circular heat exchangers. Int J
686 Heat Mass Trans. 2009; 52: 597–609.
- 687 [36] Zhai X Q, Wang R., Dai YJ, Wu JY. Experience on integration of solar thermal
688 technologies with green buildings. Renewable Energy. 2008 ;33 :1904–1910.
- 689 [37] Yin H M, Yang D J, Kelly G, Garant J. Design and performance of a novel building
690 integrated PV / thermal system for energy efficiency of buildings. Solar Energy. 2013 ;
691 87 :184-195.
- 692 [38] Thomas D, Duffy J. Energy performance of net-zero and near net-zero energy homes in
693 New England. Energy and Building. 2013 ; 67 :551-558.
- 694 [39] Bourrelle J, Andresen I, Gustavsen A. Energy payback: An attributional and
695 environmentally focused approach to energy balance in net zero energy buildings.
696 Energy and Building. 2013; 65: 84-92.