

Performance Analysis of Parabolic Trough Collector in Hot Climate

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ABSTRACT

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 appropriate 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. A review of the correlations for convective heat transfer losses was performed as well and new correlations are implemented 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 convection loss from the 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 climatic conditions of Kuwait. Finally, a total avoided 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 account for a high percentage of air pollution leading to harmful impact on our environment. In contrast, renewable energy sources can be adapted to produce energy sources with little-if any pollution. Widespread commercialization of renewable energy systems, especially in hot weather regions like Kuwait can significantly reduce energy consumption of conventional fuels. In return, this will help greatly in reducing environmental pollution and maintain our environment healthy and clean. Concentrating

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27 solar power (CSP) technologies now constitute feasible commercial options for large scale
28 power plants as well as for smaller electricity and heat generating devices. There are
29 currently four basic commercially available CSP technologies. These plants typically consist
30 of three main circuits: the Solar Field, through which the heat transfer fluid (HTF) circulates,
31 the Power Block, which circulates water and steam, and the TES system. The HTF and
32 water-steam circuits and the HTF and Thermal Storage System (TES) circuits can exchange
33 energy at the corresponding heat exchangers.

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

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

68 A reflux heat transfer storage concept for producing high-temperature superheated steam in
69 the temperature range 350–400 °C was developed and tested by Adinberg et al. [6]. The
70 thermal storage medium is a metallic substance, Zinc–Tin alloy, which serves as the phase
71 change material. A high-temperature heat transfer fluid is added to the storage medium in
72 order to enhance heat exchange within the storage system. In the same time, the associated
73 heat exchangers serve for charging and discharging the storage. The tests produced
74 satisfactory results in terms of thermal stability and compatibility of the utilized storage
75 materials up to a working temperature of 400 °C. Garcia et al. [7] presented an overview of
76 the parabolic-trough collectors that have been built and marketed during the past century, as

77 well as the prototypes currently under development. In addition, they presented a survey of
78 systems which could incorporate this type of concentrating solar system to supply thermal
79 energy up to 400 °C, especially steam power cycles for electricity generation.

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

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

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

113 Edenburn [18] predicted the efficiency of a PTC by using an analytical heat transfer model
114 for evacuated and non evacuated cases. The results showed good agreement with
115 measured data obtained from Sandia National Laboratories collector test facility [19]. Ratzel
116 et al. [20] carried out both analytical and numerical studies of the heat conduction and
117 convective losses in an annular receiver for different geometries. Three techniques were
118 proposed to reduce the conduction heat loss: evacuation, oversizing the annular receiver
119 while keeping the Rayleigh number below 1000 over the range of operation and use of
120 gases with low thermal conductivity. Dudley et al. [21] developed an analytical model of
121 SEGS LS-2 parabolic solar collector. The thermal loss model for the heat collection element
122 was one dimensional steady state model based on thermal resistance analysis.

123 Stuetzle [22] proposed an unsteady state analysis of the solar collector receiver to calculate
124 the collector field outlet temperature. The results obtained showed that the overall match

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

The foregoing literature review shows that a lot of studies have been carried out to model the performance of parabolic trough collectors. However, the existing models published previously assume that there is no thermal interaction between the neighboring surfaces (absorber-envelope, and envelope-envelope) for thermal radiation losses. The previous assumptions greatly simplify the analysis. However, these assumptions underestimate the radiation losses at high absorber temperatures. To account for the thermal interaction between adjacent surfaces, a comprehensive radiative analysis is 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 are used 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 previous studies are also taken into consideration in the present developed model. The present model can predict the effects of annuli space, tube diameter ratio and heat conductivity coefficient of the inner tube wall on the heat transfer performance. Parametric studies are carried out for different: collector absorber areas, overall heat loss coefficients, optical losses, collector inlet temperatures, mass flow rates of the heat transfer fluid, angles of incidence and concentration ratios. This will certainly lead to the optimum parabolic trough performance as well as the maximum annual energy yield under Kuwait climatic conditions.

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 as a model for existing buildings that can be moved toward nearly net zero energy buildings in Kuwait. Finally, the environmental impact of solar heating and cooling systems under Kuwait climatic conditions is investigated.

2. THEORETICAL ANALYSIS

2.1 Solar Radiation

Prediction of collector performance requires knowledge of the absorbed solar energy by the collector absorber plate. The solar energy incident on a tilted collector consists of three different components: beam radiation, diffuse radiation, and ground-reflected radiation. The details of the calculation depend on which diffuse-sky model is utilized. Using an isotropic sky 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 owing 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 the 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 the 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. The reflector is 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. The model can calculate the fraction of the incident solar energy recovered in the fluid circulating through the pipe receiver, the detailed thermal losses from the collector. The model also calculates the pressure drop of that flowed over the receiver, and the temperatures of various collector components and of the fluid throughout the receiver under different conditions. So, the model can be used for optimizing PTSC design by varying the size of the receiver tube, the size of the glass envelope, receiver tube material, coating material and so forth. In addition, the present model 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 a stainless steel tube with a selective absorber surface which provides the required optical and radiative properties. Selective surfaces combine a high absorptance of 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 solar-selective absorber surface from oxidation. The heat transfer model is based on an energy balance between the heat transfer fluid and the surroundings.

220 The solar energy reflected by the mirrors is absorbed by the glass envelope mainly $\dot{Q}_{e-abs}$
 221 and the absorber surface $\dot{Q}_{a-abs}$. A part of the energy absorbed in the absorber is transferred
 222 to the heat transfer fluid by forced convection $\dot{Q}_{a-f,conv}$, remaining energy is transferred back
 223 to the glass envelope by radiation $\dot{Q}_{a-e,rad}$ and natural convection $\dot{Q}_{a-e,conv}$ and lost through the
 224 support brackets by conduction $\dot{Q}_{cond,bracket}$ as well. The heat loss coming from the absorber
 225 (radiation and natural convection) passes through the glass envelope by conduction and
 226 along with the energy absorbed by the glass envelope $\dot{Q}_{e-abs}$ is lost to the environment by
 227 convection $\dot{Q}_{e-sa,conv}$ and to sky by radiation $\dot{Q}_{e-s,rad}$. In order to obtain the partial differential
 228 equations [29] that govern the heat transfer phenomena, an energy balance is applied over a
 229 section of the solar receiver. After applying the energy balance on a control volume,
 230 assuming unsteady state and incompressible fluid, the following partial differential equation
 231 can be obtained:

$$232 \quad A_{i,a} \rho_f C_{p,f} \frac{\delta T_f}{\delta t} = \dot{m}_f \frac{\delta}{\delta z} \left(C_{p,f} T_f + \frac{V_f^2}{2} \right) + \dot{Q}_{a-f,conv} \quad (4)$$

$$233 \quad V_f = \frac{\dot{m}_f}{\rho_f A_{i,a}} \quad (5)$$

234 where $A_{i,a}$ is the internal cross sectional area of the absorber, T_f is the fluid temperature,
 235 and $\dot{Q}_{a-f,conv}$ is the heat transfer by convection from the absorber to HTF per unit length and is
 236 given by:
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$$238 \quad \dot{Q}_{a-f,conv}' = \pi Nu_f k_f (T_a - T_f) \quad (6)$$

239 where Nu_f is the Nusselt number, k_f is the thermal conductivity of HTF, and T_a is the absorber
 240 wall temperature. The correlation for fully developed turbulent flow ($Re_D > 2300$) convective
 241 heat transfer in circular ducts [30] is adapted:
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$$244 \quad Nu_{uf} = \frac{(C_f / 2)(Re_D - 1000) Pr}{1 + 12.7(C_f / 2)^{1/2}(Pr^{2/3} - 1)} \left(\frac{Pr}{Pr_w} \right)^{0.11} \quad (7)$$

$$245 \quad Re_D = \frac{V_f D_{i,a}}{v_f} \quad (8)$$

246 where C_f is the friction coefficient (Fanning friction factor), Re_D is the Reynolds number
 247 based on the absorber inner pipe diameter, Pr is the Prandtl number, and v_f is the kinematic
 248 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
 249 thermal properties should be evaluated at the bulk mean heat transfer fluid temperature,
 250 except Pr_w which is evaluated at the absorber wall temperature. The convection heat
 251 transfer coefficient for rough tubes can be calculated approximately by using the friction
 252 coefficient determined from the Colebrook and White equation [29]:
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$$254 \quad \frac{1}{\sqrt{C_f}} = 3.48 - 1.7372 \ln \left(\frac{2\varepsilon}{D_{i,a}} + \frac{9.35}{Re_D \sqrt{C_f}} \right) \quad 5 \leq Re_\varepsilon \leq 70 \quad (9)$$

$$255 \quad \text{With } Re_\varepsilon = \frac{V_f \varepsilon}{v_f} \quad (10)$$

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 \[29\]](#) 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, $A_a = (\pi/4)(D_{0,a}^2 - D_{1,a}^2)$, $\dot{Q}'_{a,abs}$ 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 takes 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:

$$\dot{Q}'_{e-sa,conv} = h_e \pi D_{0,e} (T_e - T_\infty) \quad (13)$$

$$\text{with } h_e = \frac{Nu_e k_e}{D_{0,e}} \quad (14)$$

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

$$\text{Nu}_e = \left[0.60 + 0.387 \left\{ \frac{\text{Ra}_D}{\left[1 + (0.559/\text{Pr})^{9/16} \right]^{16/9}} \right\}^{1/6} \right]^2 \quad (15)$$

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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 the glass envelope is assumed to be totally covered by the sky surface. So, the heat flux for the area of analysis is expressed using **Stefan's Boltzmann equation** as:

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

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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:

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

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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

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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 the 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 utilized 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 previous studies are also taken into consideration 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 weather conditions. A TRNSYS studio project is designed by setting up the connecting components graphically in the Simulation Studio [33]. The parameters studied are: the 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 utilizing hourly solar radiation data for Kuwait. The processed solar insolation data consist of the beam radiation and diffuse radiation as well 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 are for a solar collector assembly (LS-2) module placed on 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 the 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.

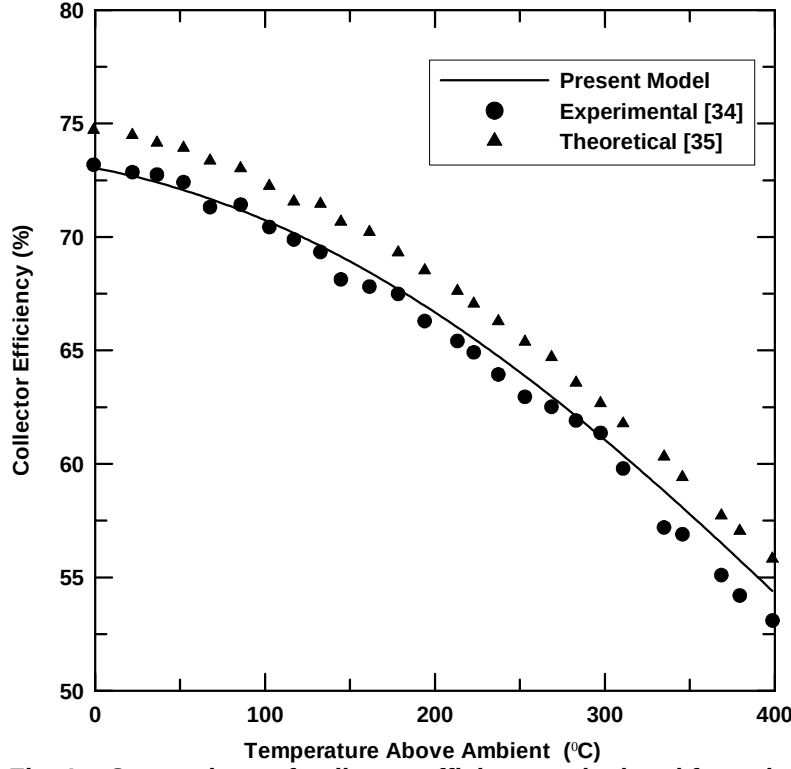


Fig. 1. Comparison of collector efficiency calculated from the present model with experimental data [34] and NREL model [35]

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. These discrepancies may be attributed 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. So, 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 :

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

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

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 first factor is the geometric reduction in irradiance falling on the aperture as θ_i increases which is referred to as the 'cosine effect'. The second factor is 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 account for the first effect apart from tilting or rotating the PTSC constantly so as to keep it oriented perpendicular to the sun's rays.

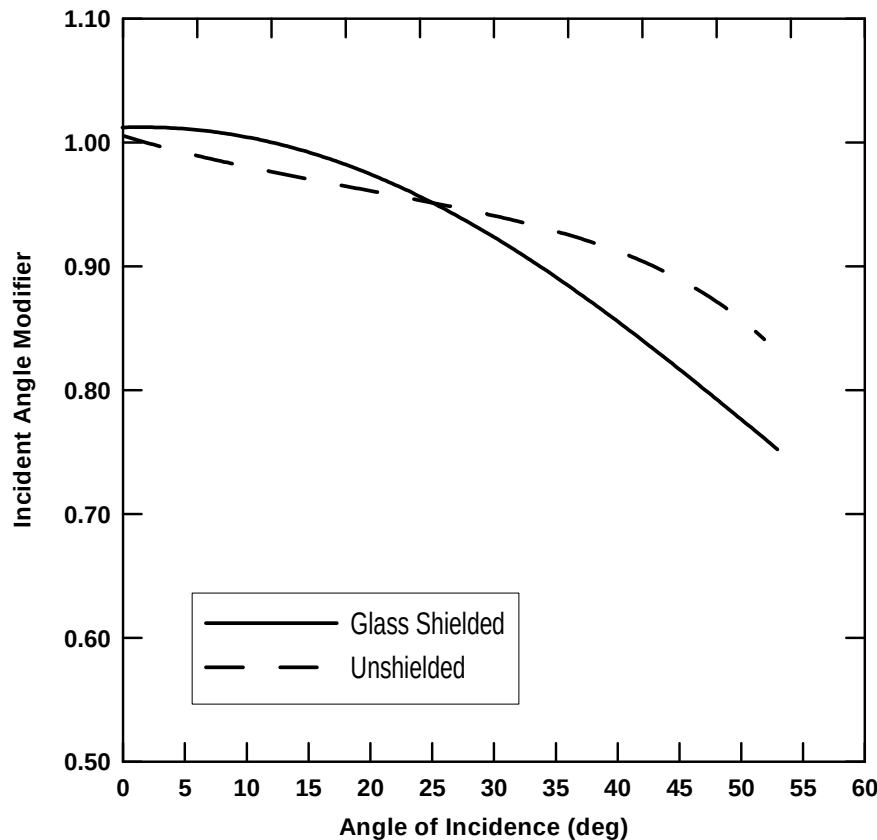


Fig. 2. Incidence angle modifier for unshielded and glass-shielded receivers

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 the absorber tube to supporting structures is the largest. The next is the radiation loss from glass envelope to ambient air; while the relative smaller loss is the convection loss from glass envelope to ambient air. Thermal losses are always existing (if there is a temperature difference between receiver and ambient) whenever solar radiation is available or not.

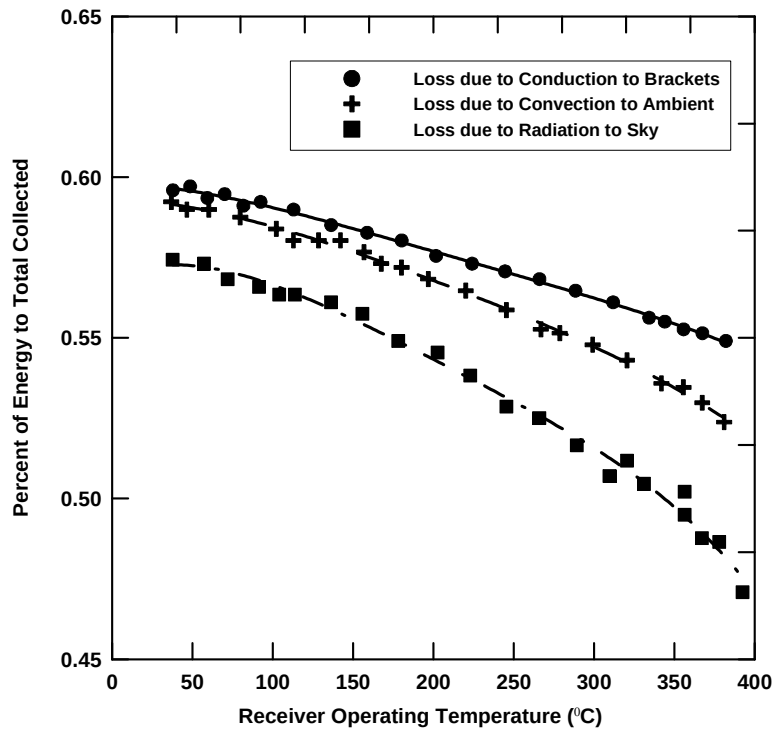


Fig. 3. Heat losses of an absorber tube

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

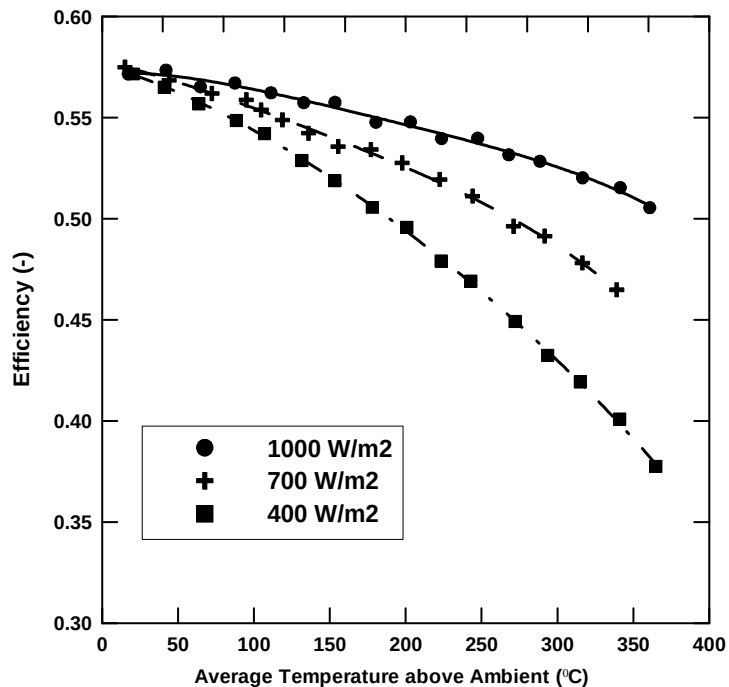
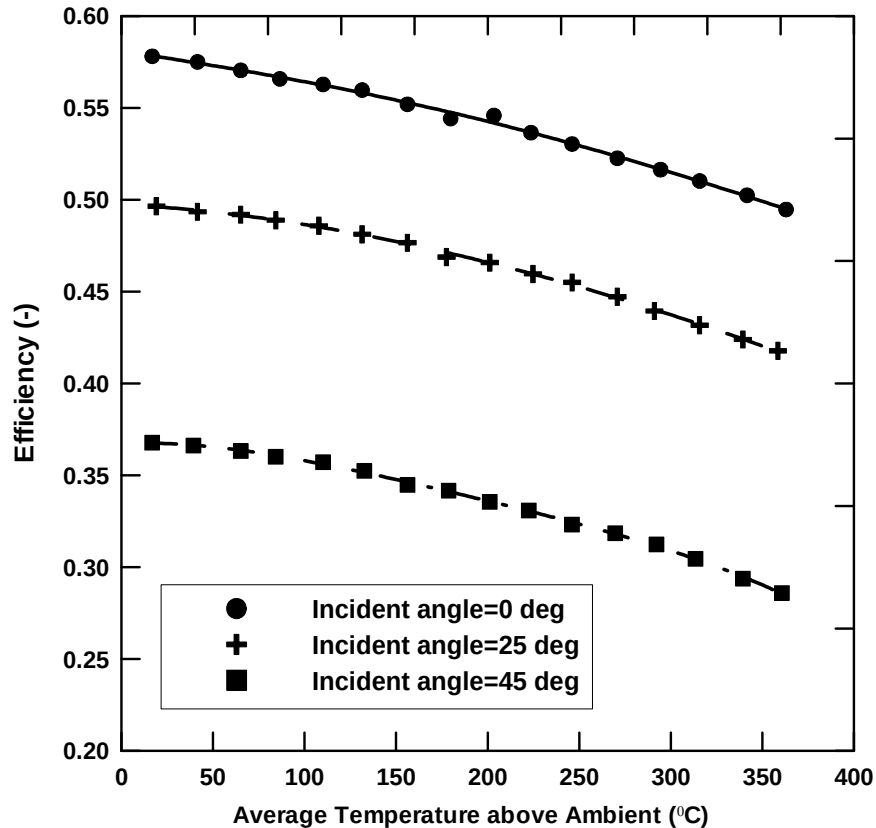


Fig. 4. Variation of parabolic trough collector efficiency with solar radiation

444 The incident angle modifier is a very significant factor impacting on the solar efficiency. The
 445 incident angle modifier is defined as the ration of the transmittance-absorptance product for
 446 the angle of the incidence of radiation and for the normal incident radiation. It is
 447 approximately equal to the cosine of the incident angle. The incident angle modifier changes
 448 depending on the time of day, the day of the year, the location and orientation of the
 449 aperture, and whether it is stationary or tracks the sun movement about one or two axes.
 450 Collector efficiency reaches a maximum value only when the incident angle is zero. The
 451 efficiency of a PTSC decreases when the solar beam incident angle increases as shown in
 452 Figure 5. The end effect of the incident angle is to reduce radiation arriving at the absorber
 453 tube. Therefore, at noon time PTSC has the smallest incident angle and the highest
 454 efficiency.
 455



456
 457

458 **Fig. 5. Variation of collector efficiency with incident angle**
 459

460 When a vacuum is present in the annulus between the receiver surface and the glass
 461 envelope, the conduction and the convection across the annulus are effectively eliminated.
 462 Once air is introduced into the vacuum space, measured losses increase significantly since
 463 conduction and convection begin to transfer heat to the glass envelope as shown from
 464 Figure 6. Radiation loss from the heated receiver's metal surface to the glass envelope is not
 465 changed very much by the presence of air in the annulus.
 466

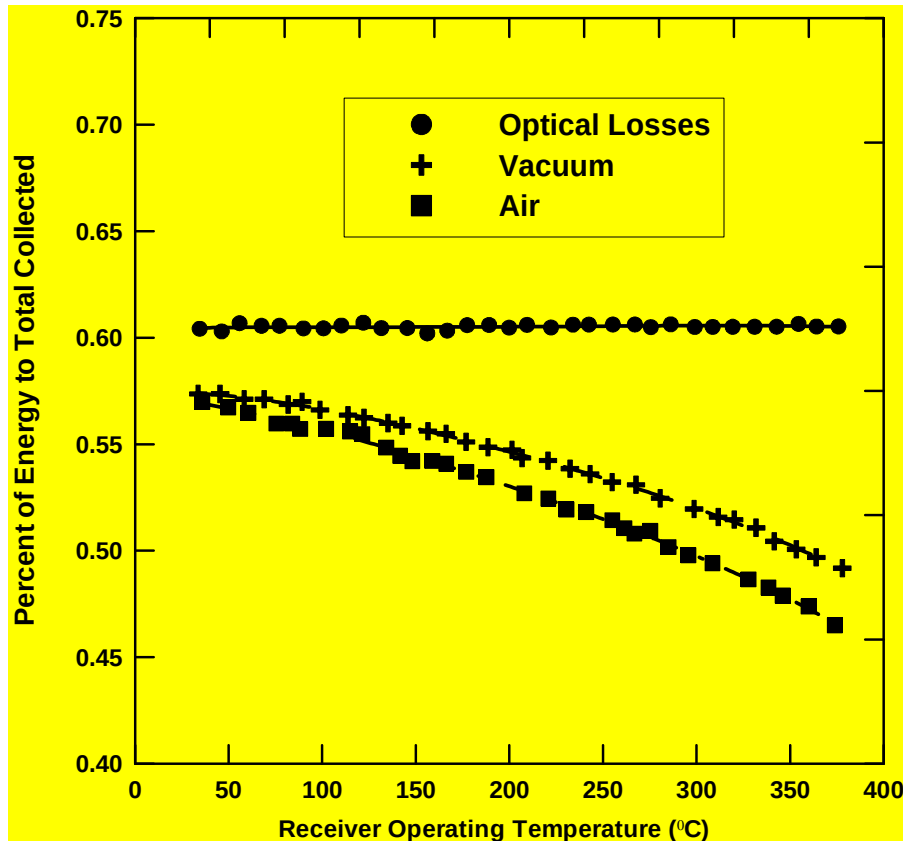


Fig. 6. Collector efficiency and air in annular

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 the 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 into 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 a parabolic trough collectors 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 [36-41]. 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 the storage temperature is not sufficient, the heating or cooling load is supplied by the auxiliary source.

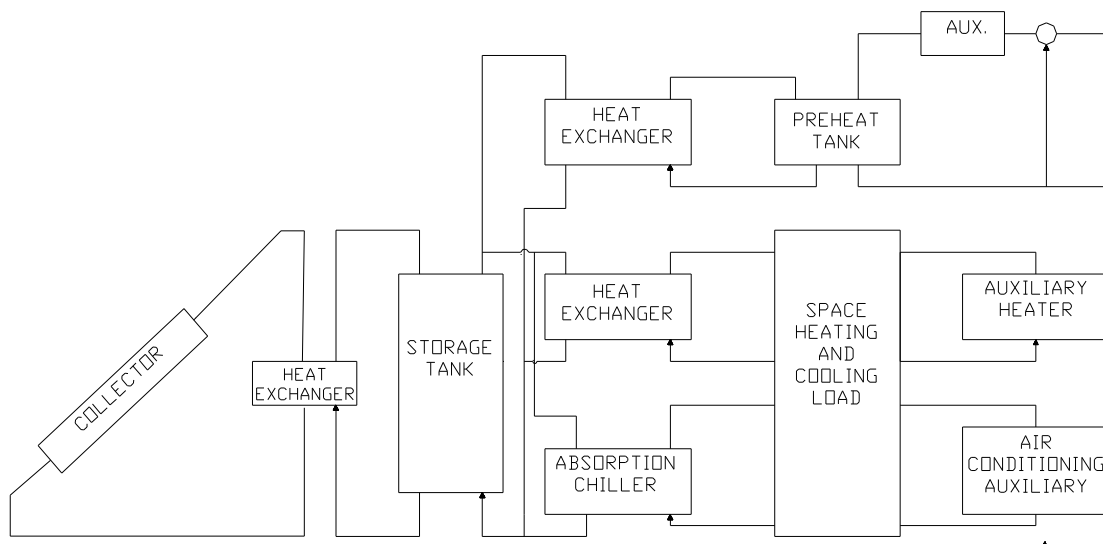


Fig. 7. Schematic diagram of the solar heating and cooling system

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 the 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 always 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-storey 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 estimate of the number of hours that the load is utilized 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.

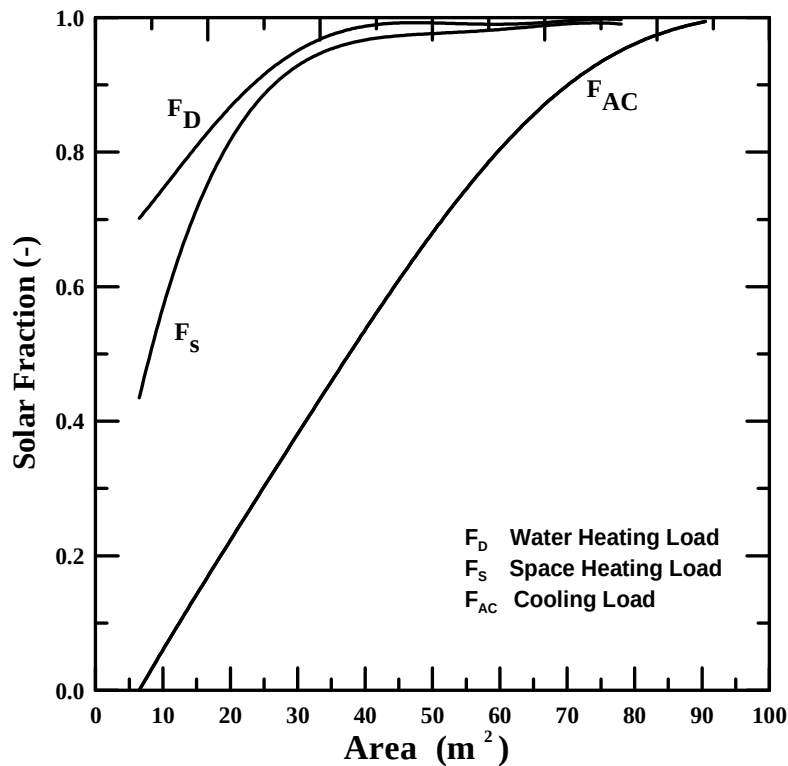
Different collector parameters and operating conditions are varied to maximize the annual energy generated by the parabolic trough collector. For building simulation, type 56 included in version 16 of TRNSYS [33] is employed. This component models the thermal behavior of

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

541 5.2 Solar Fraction

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

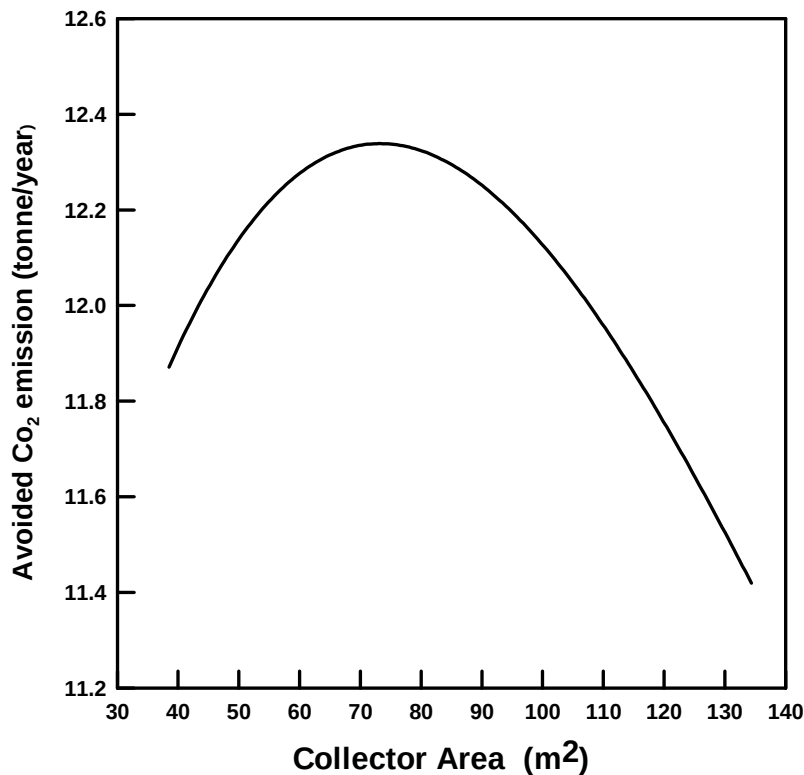
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Fig. 8. Solar fraction variation with collector area

558 The coefficient of performance (COP) of the absorption chiller is approximately 0.66 which is
 559 within the accepted practical values of the conventional lithium bromide system.
 560 A program is written to determine the yearly avoided CO₂ emission as a result of integrating
 561 parabolic trough collectors on the house roof. Variation of annual avoided CO₂ emission with
 562 collector area is presented in Figure 9. As shown from the figure at the required collector
 563 area to satisfy the house cooling load, the avoided CO₂ emission approximately equals to
 564 12.3 tonne/year. In addition, the costs due to the application of the Kyoto Protocol, which
 565 penalizes the emissions of green house effect gases, should be added to the costs of
 566 conventional energy resources. Considering an application of this protocol will enhance the
 567 economic and environmental aspects of solar systems in Kuwait much more. The present
 568 results should encourage governments for wide installation of solar heating and cooling
 569 systems in different applications which will significantly reduce energy consumption of
 570 conventional fuel as well as reducing environmental pollution to keep our community healthy
 571 and clean.
 572



573
 574
 575 **Fig. 9. Avoided CO₂ emission variation with collector area (azimuth angle=0)**
 576

577 6. CONCLUSIONS

578
 579 This work investigates the performance of parabolic trough collectors as well as solar
 580 heating and cooling systems in Kuwait climate. A theoretical model takes into consideration
 581 the factors ignored in previous models is developed. Based on the present results, the
 582 following conclusions can be drawn:

- 583 • The performance of parabolic trough collectors can be significantly enhanced by
 584 optimizing its parameters as well as operating conditions.
- 585 • Convection loss from the absorber tube to the supporting structures is the largest
 586 among the other losses (conduction and radiation).

- The incident angle modifier is a very important factor impacting on the solar efficiency.
- At noon time, PTSC has the smallest incident angle and the highest efficiency.
- 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.
- Space heating load and domestic water heating load can be completely provided by parabolic trough collectors.
- The minimum required parabolic trough collector area is about 82 m² to supply cooling loads of a typical residential house in all hot climatic conditions of Kuwait.
- Total avoided CO₂ emission of about 12.3 tonne/year can be achieved.
- The results of the present study should encourage widespread utilization of solar energy systems which will help in keeping our environment healthy and clean.

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