

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

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 (a) 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 (due) to the angle of incidence. These losses occur owing to (because of) additional reflection and absorption by the glass envelope when (as) 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 ? (yields):

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

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 (the) 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 (proposed) 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 (environment).

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

$$235 \quad A_{i,a} \rho_f C_{p,f} \frac{\delta T_f}{\delta t} = \dot{m}_f \frac{\delta}{\delta z} \left(C_{p,f} T_f + \frac{V_f^2}{2} \right) + \dot{Q}_{a-f,conv} \quad (4)$$

$$236 \quad V_f = \frac{\dot{m}_f}{\rho_f A_{i,a}} \quad (5)$$

237 where $A_{i,a}$ is the internal cross sectional area of the absorber, T_f is the fluid temperature,
 238 and $\dot{Q}_{a-f,conv}$ is the heat transfer by convection from the absorber to HTF per unit length and is
 239 given by:
 240

$$241 \quad \dot{Q}_{a-f,conv}' = \pi Nu_f k_f (T_a - T_f) \quad (6)$$

242
 243 where Nu_f is the Nusselt number, k_f is the thermal conductivity of HTF, and T_a is the absorber
 244 wall temperature. The correlation equation for fully developed turbulent flow ($Re_D > 2300$)
 245 convective heat transfer in circular ducts [30] is adapted:
 246

$$247 \quad Nu_{uf} = \frac{(C_f / 2)(Re_D - 1000) Pr}{1 + 12.7(C_f / 2)^{1/2}(Pr^{2/3} - 1)} \left(\frac{Pr}{Pr_w} \right)^{0.11} \quad (7)$$

$$248 \quad Re_D = \frac{V_f D_{i,a}}{\nu_f} \quad (8)$$

249 where C_f is the friction coefficient (Fanning friction factor), Re_D is the Reynolds number
 250 based on the absorber inner pipe diameter ($D_{i,a}$), Pr is the Prandtl number, and ν_f is the
 251 kinematic viscosity of HTF. This correlation equation is valid for $2300 \leq Re_D \leq 5 \times 10^6$ and
 252 $0.5 \leq Pr \leq 2000$. The thermal properties should be evaluated at the bulk mean heat transfer
 253 fluid temperature, except Pr_w which is evaluated at the absorber wall temperature.

254 The convection heat transfer coefficient for rough tubes can be calculated
 255 approximately by estimated using the friction coefficient determined from the Colebrook and
 256 White equation [29]:
 257

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

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

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 [29] has shown that heat transfer losses are (is) independent of on 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 (the) 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 (absorber) cross sectional area of the absorber, $A_a = (\pi/4)(D_{o,a}^2 - D_{i,a}^2)$, (D stands for Diameter?) $\dot{Q}'_{a,abs}$ 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 (wind guided) (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 (. 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 (the) control volume leads to (results in) 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:

295

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

297

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

299

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

302

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

304

305 The solar receiver is surrounded by either the collector or the sky. In order to simplify the
306 model, it is assumed that half of the receiver surface is surrounded by the mirror and the
307 other half by the sky. The heat flux and radiosity are calculated for each surface and a set of
308 energy balance equations for internal and external surfaces are written. In this work, it is
309 assumed that the collector mirror temperature is approximately the ambient temperature.
310 When the collector surface is not included in the analysis, maximum radiation heat transfer
311 loss occurs, and this means that the glass envelope is assumed to be totally covered by the
312 sky. So, the heat flux ($\dot{q}_{esi}$) for the area of analysis (considered) is expressed by using
313 Stefan's Boltzmann's equation: (explain terms)

314

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

316

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

321

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

323

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

331

332 3. NUMERICAL MODEL

333

334 The previous (set of) partial differential equations (PDE) were (was) discretized for steady
335 state conditions by using the finite difference method and taking into account the
336 dependence of the thermal properties on the temperature. For (Turning to) the heat transfer

fluid, the discretization by using backward difference (differencing) gives creates a set of algebraic equations. For the absorber and the envelope, the discretization is carried out by using the central difference and (thus,) another set of algebraic equations are (is) obtained. Finally, the boundary conditions for each element are set down. The present (proposed) model takes into consideration (includes) the thermal interaction between absorber-envelope, and envelope-envelope, for (These) thermal radiation losses which have been neglected were not included 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 (equations) for convective heat transfer losses was performed as well, and new correlations (equations) were utilized in (incorporated to) 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 (explain or show us). (Full stop)//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: (explain terms)

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

The thermal losses from the collector receiver are (is related to the) functions of operating temperature. The thermal losses through PTSC are (can be) changed in different ways depending on the receiver's configuration and operational conditions. In general, (T) the thermal losses are given by (can be expressed as): (explain terms)

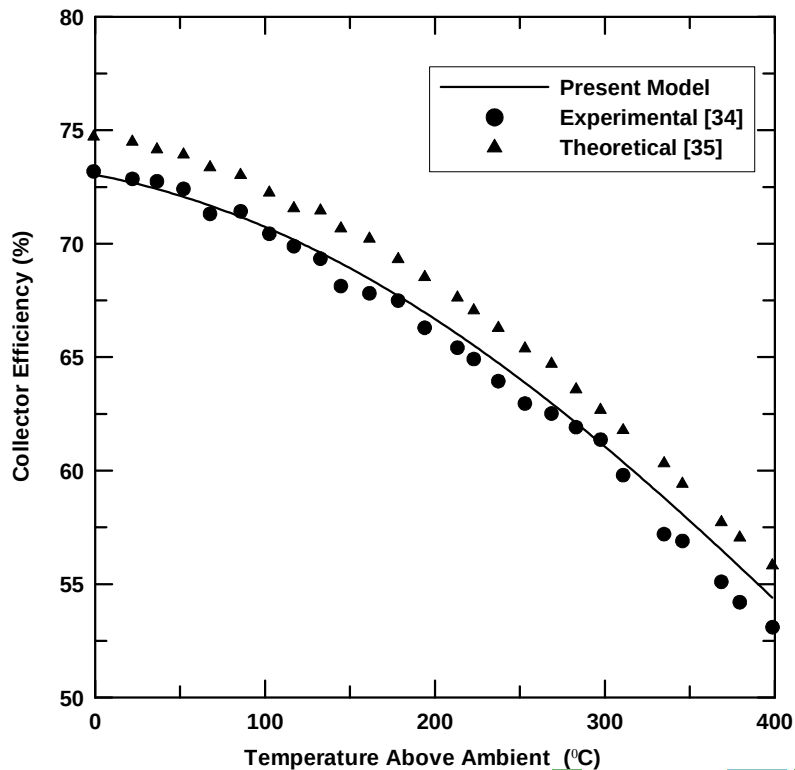
$$\text{Thermal loss} = \left[\sum_i (\dot{Q}'_{a-e, \text{conv}})_i + \sum_i (\dot{Q}'_{a-e, \text{rad}})_i + \sum_j (\dot{Q}'_{\text{cond, bracket}})_j \right] \frac{\Delta z}{I_b A_{ap}} \quad (19)$$

A simulation model compatible with the transient simulation program (TRNSYS) is (was) developed to determine the thermal performance of a typical parabolic trough configuration under Kuwait weather conditions. A TRNSYS studio analysis project is designed by setting up the connecting components graphically in the simulation study [33]. (include authors The Parameters studied are: treated were; the area of the absorber of the collector area of collector's absorber, overall heat loss coefficient from the absorber, reflectivity of the reflecting surface and absorptivity and emissivity of the absorber. The parametric study has been made was conducted 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. measured in what units?.

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] (include authors). In addition, to corroborate the improvement in the correlations (equations) and radiation analysis proposed in this work (here), the numerical model was also compared with other solar receiver heat transfer models [35] (include authors). The experimental results are (were performed) for a solar collector assembly module (LS-2) placed on the an AZTRAK rotating platform at the SNL. Two different selective coatings were used in this test: black chrome and cermet.

384 Cermet has better radiative properties (low emissivity) at high temperatures than black
 385 chrome, and does not oxidize if the vacuum is lost. (//Full stop) The Sandia test was
 386 performed for both full sun and no sun condition and different scenarios for the annulus of
 387 the heat collection element (HCE): vacuum intact (the *in vacuo* (evacuated) annulus
 388 pressure was 10^{-4} Torr), lost vacuum (annulus filled with ambient air), and glass cover
 389 completely removed (bare tube). Figure 1 presents the results obtained for the collector
 390 efficiency. As seen from in the figure, the current calculated collector efficiency follows the
 391 trends of the experimental values and all of the results are always inside (within) the
 392 experimental error bars. (I can see no error bar) As it was expected, the higher efficiencies
 393 are obtained when the annulus is under vacuum (*in vacuo*), but in both cases (air and
 394 vacuum) at high temperature, the collector efficiency drops gradually. This behavior which
 395 is more notable noticeable for the black chrome coating.
 396



397
 398 **Fig. 1. Comparison of collector efficiency as calculated from with the present**
 399 **proposed model with to experimental data [34] and NREL model [35] (add author to**
 400 **references)**
 401

402 For the case of Cermet coating, the model developed by Garcia-Valladares and Velasquez
 403 [35] shows some discrepancies at low temperatures in the collector efficiency. These
 404 discrepancies may be attributed to their assumptions: negligible conduction at the ends of
 405 each trough and only radiation heat losses takes place between the receiver and the glass
 406 envelope for the case of vacuum in the (*in vacuo*) annular space. Those (These)
 407 assumptions were not made (neither) in the present model or (nor in the) NREL model. So,
 408 the present model and NREL models [34] (include authors) give almost similar collector
 409 efficiencies values.

410 The incidence angle modifier, $K_{\text{at}}(\theta_i)$, enables the performance of the collector to be
 411 predicted for solar angles of incidence other than 0° (normal). Simulations using the present
 412 numerical model are carried out at a setting up a value of θ_i and then calculating K_{at} . //It

413 should be mentioned that the The incident angle of incidence modifier represents the ratio
414 between thermal efficiency values at a specified value and the peak efficiency of the
415 collector at zero incidence. Results of simulation are presented in Figure 2. Regression
416 analyses provided the following equations for $K_{\alpha\tau}$ as a function of θ_i :
417

$$418 \quad K_{\alpha\tau U} = -2.235 \times 10^{-6} (\theta_i)^3 + 1.2146 \times 10^{-4} (\theta_i)^2 - 3.884 \times 10^{-3} (\theta_i) + 1.125 \quad (20)$$

$$419 \quad K_{\alpha\tau G} = 9.272 \times 10^{-7} (\theta_i)^3 - 1.561 \times 10^{-4} (\theta_i)^2 + 1.773 \times 10^{-3} (\theta_i) + 1.078 \quad (21)$$

421
422 Coefficients of determination for equations (20) and (21) are 0.958 and 0.961. Up to an
423 incidence angle angle of incidence of approximately 25° the glass-shielded receiver
424 performed slightly better, but beyond that its performance declined more rapidly and was
425 inferior to that of the unshielded receiver. //Full stop From In Figure 3, the calculated value of
426 $K_{\alpha\tau}$ was 0.74 for the glass-shielded receiver at the maximum tested incidence angle of 54°.
427 This was , an 8.9% lower than that for the unshielded receiver. At the same maximum angle
428 of incidence, the simplified cosine model under-predicted $K_{\alpha\tau}$ by about 25% for the
429 unshielded receiver and by 17.7% for the glass shielded unit. Two factors are primarily
430 responsible for the decline in performance of a PTSC with increasing θ_i , the first factor is the
431 geometric reduction in irradiance falling on the aperture as θ_i increases which is referred to
432 as the or 'cosine effect'. The second factor is and the change in optical efficiency as light
433 interacts differently (due to differences in light interaction) with the reflective surface of the
434 collector, the glass shield (if present) and the absorber. Nothing can be done to account for
435 the first effect apart from (besides of) tilting or rotating the PTSC constantly so as to keep it
436 oriented perpendicularly (oriented) to the sun's rays.
437

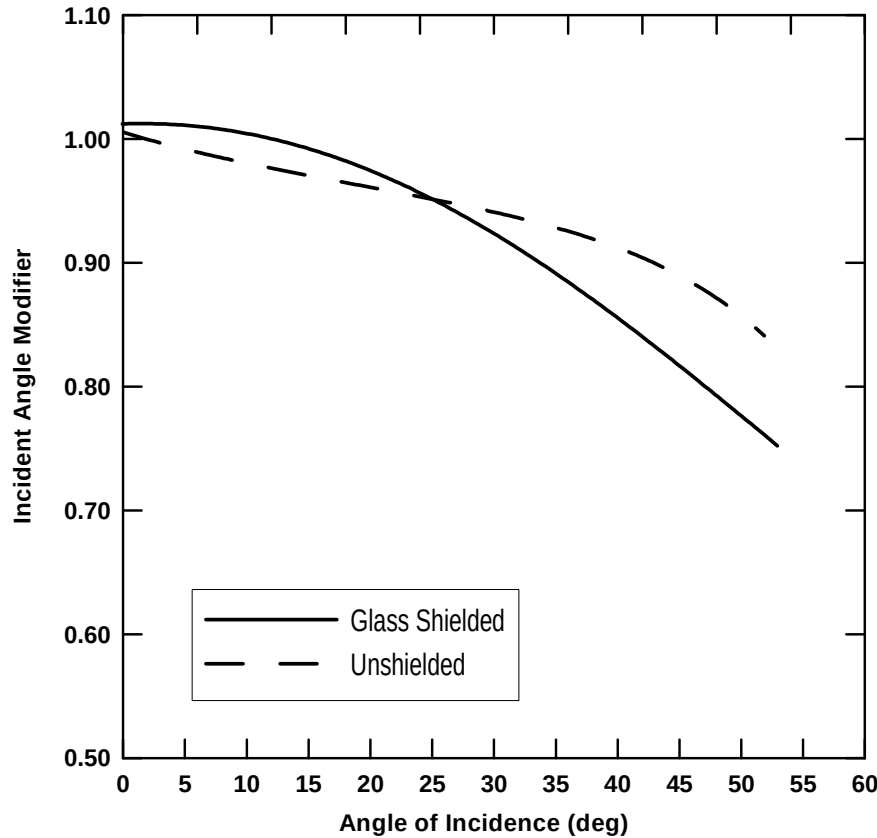


Fig. 2. Incidence angle modifier for unshielded and glass-shielded receivers

The thermal losses from (of) the collector receiver are functions of (depend on) 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 (Follows) 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 (present) existing (if there is a temperature difference between receiver and ambient) whenever solar radiation is available or not.

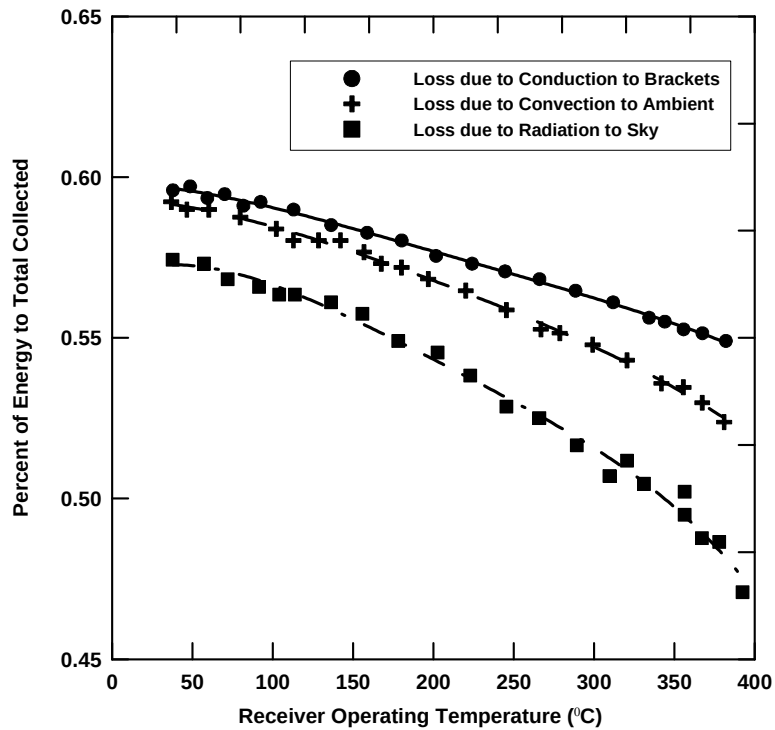
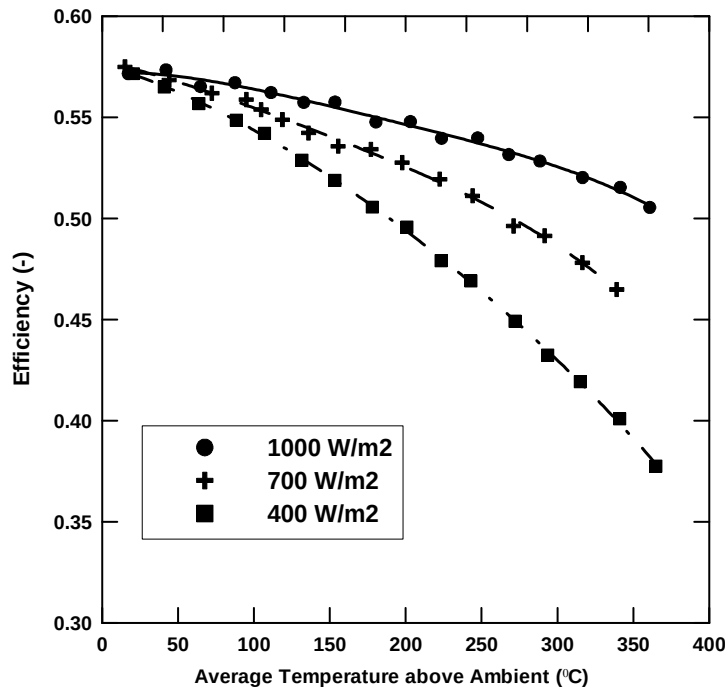
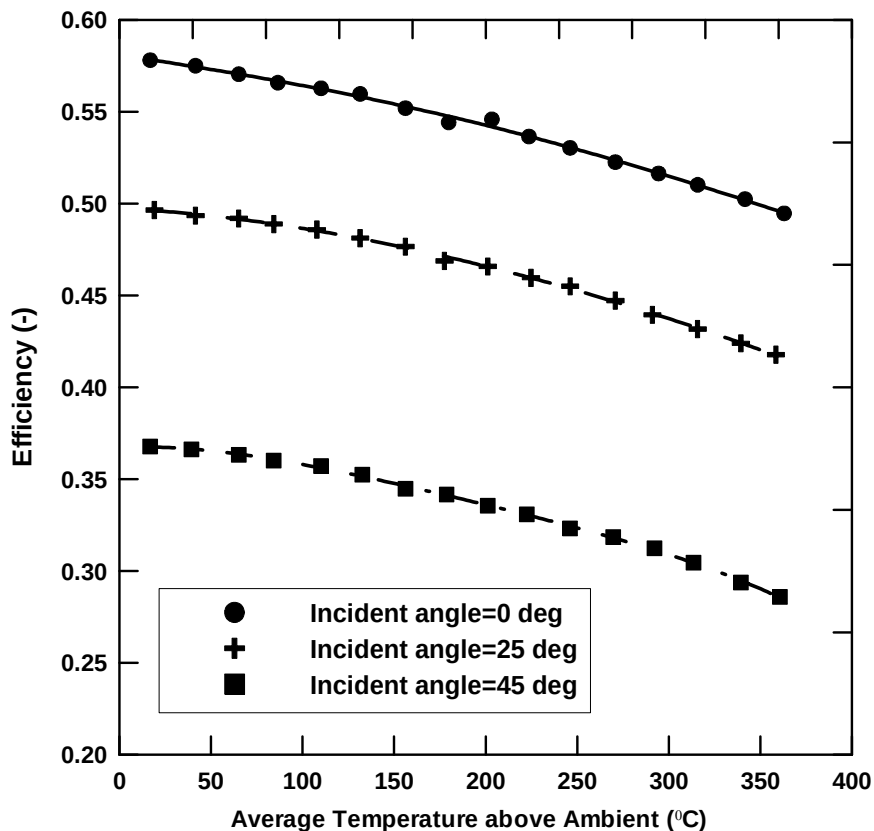


Fig. 3. Heat losses of an absorber tube

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.



462 **Fig. 4. Variation of parabolic trough collector efficiency with solar radiation**
 463 The (incident angle) modifier is a very significant factor impacting on the solar efficiency. The
 464 (incident angle modifier) η_i is defined as the (ratio of) quotient between the transmittance-
 465 absorptance product for the angle of the incidence of radiation and (for) the normal incident
 466 radiation. Not legible, rewrite. It is approximately equal to the cosine of the angle of
 467 incidence (incident angle). The (incident angle) angle of incidence modifier changes
 468 depending depends on time of day, date (the day of the year), the location and orientation of
 469 the aperture, and whether the collector (it) is stationary or tracks the sun movement about
 470 one or two axes. Collector efficiency reaches a maximum value only at zero angle of
 471 incidence (when the incident angle is zero). The efficiency of a PTSC decreases when the
 472 solar beam angle of incidence (incident angle) increases, as shown in Figure 5. The final
 473 (end) effect of the angle of incidence (incident angle) is to reduce radiation arriving at the
 474 absorber tube. Therefore, at noon time PTSC has the smallest angle of incidence (incident
 475 angle) and the highest efficiency.
 476



477
 478
 479 **Fig. 5. Variation of collector efficiency with incident angle**
 480

481 When (a vacuum is present in) the annulus between the receiver surface and the glass
 482 envelope is in vacuum state, (the) conduction and (the) convection across the annulus are
 483 effectively eliminated. Once air is introduced into the vacuum space, measured losses
 484 increase significantly since conduction and convection begin to transfer heat to the glass
 485 envelope as shown from Figure 6. Radiation loss from the heated receiver's metal surface to
 486 the glass envelope does not change significantly (is not changed very much by the presence
 487 of) when air is allowed into the annulus.
 488

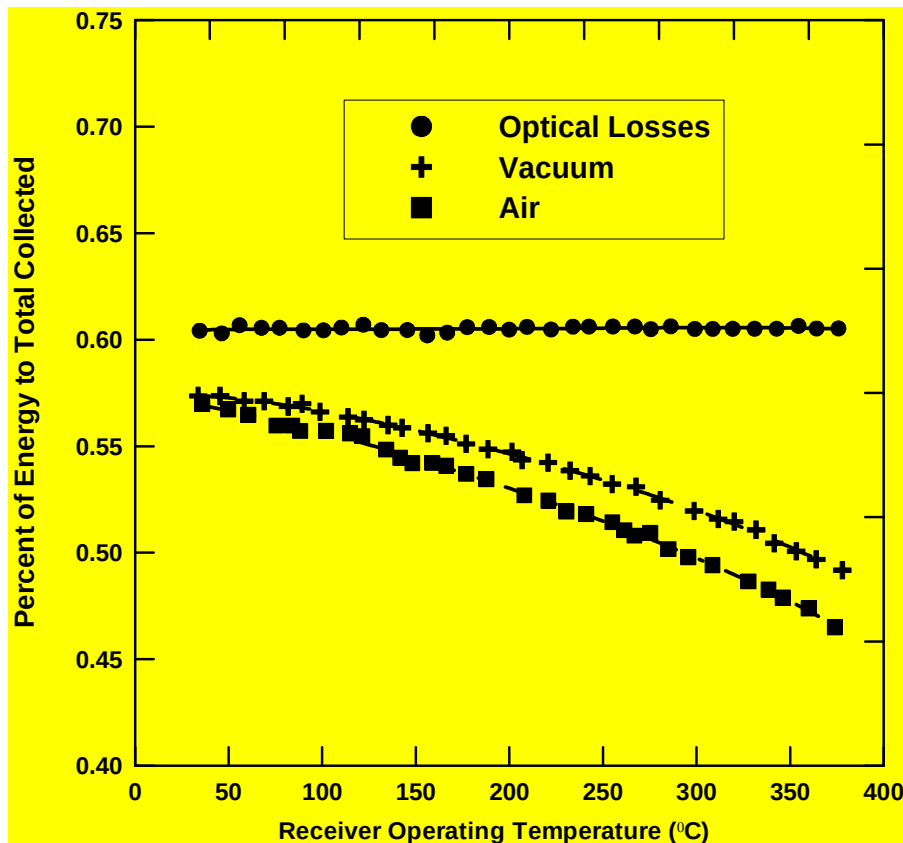


Fig. 6. Collector efficiency with air in annulus

5. CASE STUDY

A case study is carried out to investigate the performance of solar heating and cooling systems 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 order (an attempt) to convert it to nearly zero net energy building. Buildings are typically responsible for a significant part of the total primary energy consumption. (So, it is the aim goal of) Thus, researchers and building designers thrive to develop energy efficient buildings (that can use the energy produced) from renewable energy sources. Recently, Many new (and existing) buildings incorporate solar energy technology (ies) to attain environmental goals (benefits in comparison) as opposed to (the) conventional energy in terms of reducing global warming and greenhouse and other gases emission(s), mainly CO₂ (, and preventing toxic gas emissions) [36-41] (include authors). 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, the storage unit 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.

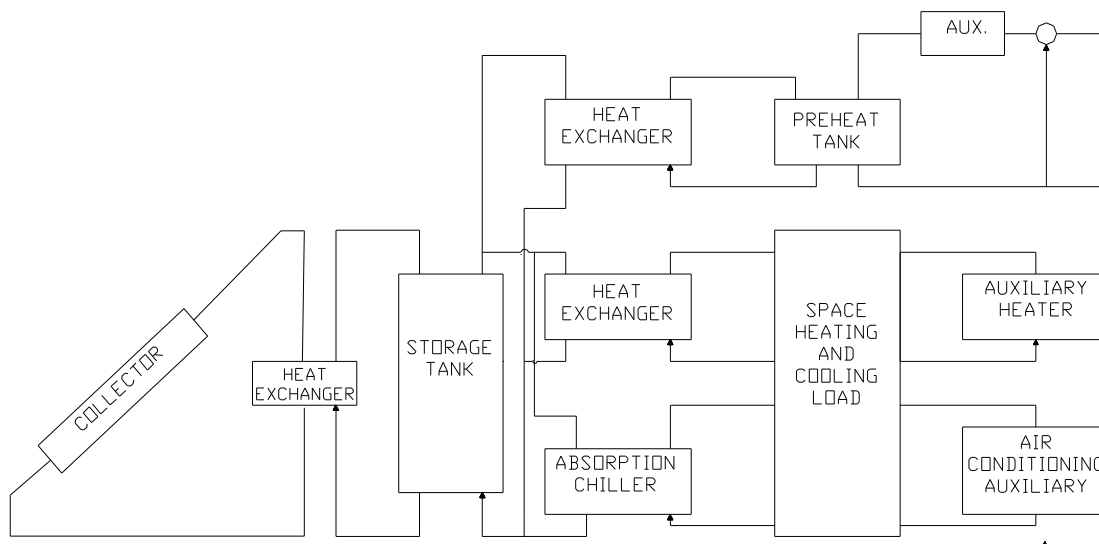


Fig. 7. Schematic diagram of a solar heating and cooling system

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 temperatures ranging between 87°C and 93°C, a temperature of 87°C (minimum), 93°C (maximum) and leaves, leaving this unit 10°C cooler than the supply and (returns) ending into (to) the storage, or to the auxiliary heater if storage temperature 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 input energy he puts. In addition, a constant COP model of a vapor compression air conditioner is included as a secondary cooling auxiliary unit, 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 model dwelling under study (building studied) is a (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 an input file. 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 up to a load peak is of about 21 kW (6 tons of refrigeration), which occurs in August.

Different collector parameters and operating conditions were essayed (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] (authors) is employed. This component models the thermal behavior of a building with (having) multiple thermal zones. The building description is ready by (??? not legible) 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.////Full Stop. From the simulation done with the TRNbuild (Type 56) of TRNSYS to determine the building's demand for air conditioning in the house under study, we obtained results that indicate that this demand starts spans from April to October, with critical periods of maximum load in (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 flow inlet and two flow outlet (flows), (type 60), is adapted. Several data files of single-effect absorption chillers, (of single-effect) employing LiBr-H₂O solution as working fluid, (type 107) according to Yazaki chiller WFC SH10 are used. Other types used were: Type 15, TMY2, Type 3b for pump 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 the 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 are completely satisfied for (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 under (in) all hot climatic conditions in (of) Kuwait, with a (where the) maximum cooling load of approximately (is within) 21 KW (6 ton refrigeration).

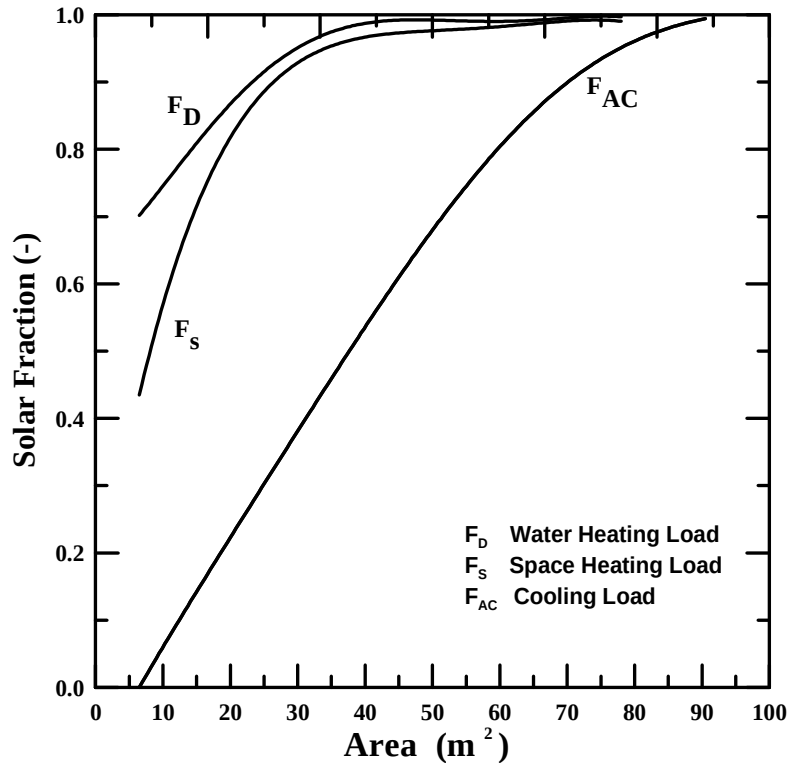


Fig. 8. Solar fraction variation with collector area

The coefficient of performance (COP) of the absorption chiller is approximately 0.66 which is within the accepted practical values of the conventional lithium bromide system.

A program is written to determine the yearly reduced (avoided) CO₂ emission as a result of integrating parabolic trough collectors on the house roof. The annual variation of (avoided) CO₂ decreased emission (with) vs. collector area is presented in Figure 9. (As shown from the) In this figure at (the required) collector's area values (to) satisfying the house cooling loads, (the avoided) CO₂ reduction in emission is approximately (equals to) 12.3 tonne/year. In addition, there are costs due to the application of the Kyoto Protocol, which penalizes the emission(s) of green house effect gases(i). They should be added to the costs of conventional energy resources. Considering that the (an) application of this protocol will enhance the economic and environmental aspects of solar systems in Kuwait. (much more). The present results should encourage governments for wide installation of solar heating and cooling systems in different applications which will significantly reduce the (energy) consumption of conventional fuel as well as reducing environmental pollution to keep our community healthy and clean.

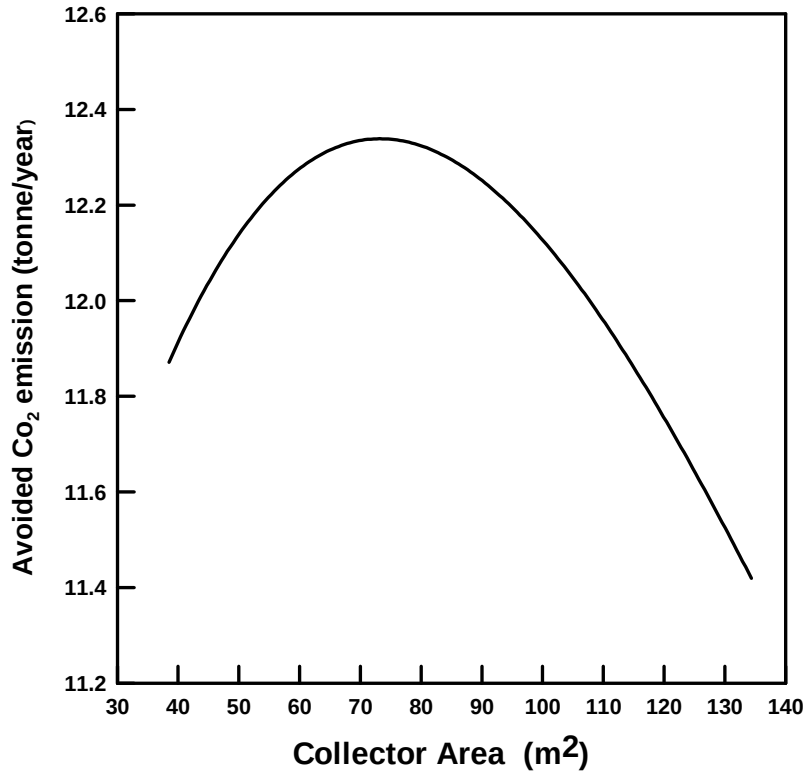


Fig. 9. Avoided CO₂ emission variation with collector area (azimuth angle=0)

6. CONCLUSIONS

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

- The performance of parabolic trough collectors can be significantly enhanced by optimizing its parameters as well as operating conditions.
- Convection loss from the absorber tube to the supporting structures is the largest among the other losses (conduction and radiation).
- The incident angle modifier is an (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 is in vacuo, (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 under (in) all (hot) climatic conditions (of) in Kuwait.
- Total (avoided) reduction in CO₂ emission may reach to some (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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