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Theoretical modelling approaches of heat pipe solar collectors in solar systems: A comprehensive review

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Abstract

The invention of heat pipe solar collectors (HPSCs) is considered as an immense step forward towards solving the challenges of conventional solar thermal systems. Their unique qualities have acted as a great motivation for researchers to focus their studies on HPSCs and their applications. A considerable share of these studies has been allocated to theoretical studies due to several technical and economic reasons. However, to the authors' knowledge and despite many valuable efforts in this field, there is no review paper available to summarise the relevant proposed and developed theoretical models to date and identifies the research gaps in this field. Therefore, in this review paper, the latest theoretical studies in the field of HPSCs along with their advantages, disadvantages, and contribution have been categorized, reviewed, and discussed. First, the operational principles and structure of HPSCs have been explained to create a background for readers. This is followed by a short section dedicated to the simulation of solar radiation as the most important input for all solar mathematical models. In addition, various mathematical approaches including steady state models (i.e. one-dimensional energy balance and thermal resistance network methods), dynamic models, and models for novel configurations and applications of HPSCs have been reviewed. Moreover, mathematical models to determine the exergy efficiency of HPSCs, which is an effective tool to evaluate the solar systems from a thermodynamic point of view, have been presented. Finally, the challenges, research gaps, and recommendations for future research directions have been provided.

Keywords: Solar collector; Mathematical model; Heat pipe; Theoretical study

Contents

Nomenclature

a	Ambient	Q	Thermal energy (W)
ab	Absorber	Q_{ab}	Absorbed thermal energy (W)
amb	Ambient	Q_{en}	Solar energy passed through the glass tube (W)
A_c	Gross area of solar collector (m^2)	Q_{loss}	Thermal energy dissipated back into the ambient (W)
A_g	Glass tube surface area (m^2)	R	Thermal resistance (K/W)
c	Collector	r	Radiation
F_R	Heat removal factor of the solar collector	R_b	Incident radiation coefficient
f_1	Circumsolar brilliance coefficients	T	Temperature (K)
f_2	Horizon brilliance coefficients	T_a	Ambient temperature (°C)
G	Total solar irradiance intensity (W/m^2)	U_L	Solar collector heat transfer loss coefficient ($W/(m^2\ °C)$
h	Convective heat transfer coefficient (W/m^2K)	X_1	Auxiliary angle
hp	Heat pipe	X_2	Auxiliary angle
I_b	Beam radiation on FPSC	X_3	Auxiliary angle
I_{bT}	Beam radiation on HPSC	X_p	Passive shading factor
I_d	Diffused radiation on FPSC	**Greek letters**	
I_{dT}	Diffused radiation on HPSC	$\tau\alpha$	Transmittance-absorption product
I_g	Ground reflected radiation on HPSC	γ_c	Solar collector azimuth angle
N	Number	γ_s	Solar azimuth angle
n	Number	β	Collector slope
gi	Inner glass	α	Absorptivity
go	Outer glass	σ	Stefan-Boltzmann constant ($5.67 \times 10^{-8}\ W/m^2K$)
p	Absorber/plate	ρ_g	Glass diffuse reflectance

1. Introduction

The fact that fossil fuel resources are running out along with their consequent environmental side effects have motivated researchers to look for alternative energy resources. Among all the available energy resources, solar energy has got much more attention due to its accessibility, renewability, less environmental impacts, and great potential (Rassamakin et al., 2013). Nevertheless, solar energy is accessible only during daytime which creates intractable challenges regarding its domestic and industrial applications. This emphasises the importance of the efficient collection and effective storage of solar energy during its availability and explains why solar collector is the key component of all solar-driven systems (Kalogirou, 2014).

Solar collectors are fundamentally categorised into three main groups which are flat plate, evacuated tube, and heat pipe solar collectors. Flat Plate Solar Collectors (FPSC) are cheap, manufactured easily, and flexible to be applied in various forms. However, they have noticeable drawbacks affecting their technical and economic feasibility especially for cold climatic conditions and in high-temperature applications. The thermal efficiency of FPSCs drops below 40% in non-ideal climatic conditions with low ambient temperature (Rassamakin et al., 2013).

Being vulnerable to moisture (Kalogirou, S.A., 2014), having high hydraulic resistances (Burch, 2007), and dependency on sun trackers for better efficiency which adds to maintenance and operational costs (Sabiha, M.A . et al., 2015) are some of the main disadvantages of FPSCs. In addition, FPSCs are not suitable for high-temperature applications as their thermal losses increase leading to a significant decrease in their thermal efficiency. To overcome this issue, using anti-reflective glasses, better insulation, and high-quality absorbing surfaces have been proposed, however, these methods result in extra manufacturing and maintenance costs which affect the economic feasibility of FPSCs (Gani and Symons, 1979).

The second group of solar collectors are Evacuated Tube Solar Collectors (ETSC) which were proposed to resolve the challenges of FPSCs. In order to make the collector suitable for high-temperature applications, the air between two glass layers of the tubes is extracted to create a vacuum space acting as thermal insulation to reduce the thermal losses (Window and Hardin, 1984). This strategy along with the usage of selective absorbing materials and transmitters enhance the thermal efficiency of the solar collector even in cold climatic conditions with low solar radiation (Brunold et al., 2007; Speyer, 1965). Low maintenance cost is another important advantage of ETSCs (Lee, 2010) which has led to a significant increase in their market share (Bejan and Kraus, 2003; Lee, 2011; Yulong et al., 2011).

Despite the benefits mentioned above, ETSCs still have several critical disadvantages such as the possibility of overheating. The temperature of these collectors is likely to pass the specified limit for domestic applications which in return requires the continuous availability of working fluid to absorb the extra thermal energy. The absence of working fluid results in critical damages such as vacuum loss or material problems.(Rassamakin et al., 2013). ETSCs have relatively high initial costs (Xue-Fei and Zhen-Hua, 2011) and need special care while they are handled as they are made of a special fragile glass (Rassamakin et al., 2013).

Heat pipe solar collectors (HPSCs) were introduced to overcome the limitations of the first two groups of solar collectors by taking advantage of both heat pipe and ETSC technologies. Low thermal resistance, high heat removal from absorbing surface, and low hydraulic resistances are some of the advantages of this type of solar collectors (Azad, 2008). The application of heat pipes results in higher heat transfer capability and lower heat transfer weight and area as the heat transfer process of heat pipes is based on phase change which reduces the temperature drop significantly (Bejan and Kraus, 2003). Additionally, heat pipes are highly powerful in heat transfer as the heat transfer coefficient of the phase change process is high (Çağlar and Yamalı, 2012). The heat pipes operate independently from any mechanical devices (Bejan and

Kraus, 2003) and regulate the operating temperature to avoid overheating which is a common challenge in solar systems (Bienert and Wolf, 1976; DeVriers et al., 1980). Low possibility of freezing and corrosion, high operating life (Deng et al., 2015), simple fitting in buildings, simple design and light weight (Fischer, 2012) are other advantages of HPSCs.

Due to the advantages mentioned above, the researchers have been attracted to HPSCs especially in the last decade. A great share of these efforts has been allocated to theoretical studies due to recent advancements in computing power and processes. The improvement in computing tools has made analysis and performance optimization of HPSCs much easier. In addition, conducting experiments is costly and time-consuming especially when large-scale systems and multiple variables are involved. Moreover, the experimental analysis usually needs prior theoretical studies either regarding their design and sizing or finding the optimum operating parameters of the system. Furthermore, theoretical modelling provides the opportunity of going deep into the physics of the study and does not include the experimental uncertainties (Yılmaz and Mwesigye, 2018). However, the role of the experimental study should not be underestimated as all the mathematical models need experimental verification.

Despite all the valuable studies in the field of HPSCs, to the authors' knowledge, there is no review paper available to summarise all the proposed and developed theoretical models to date. Therefore, in this review paper, the latest theoretical studies in the field of HPSCs along with their advantages, disadvantages, and contribution have been categorised, reviewed, and discussed. Besides, the challenges in this field, research gaps, and recommendations for future research directions have been provided. This review paper complements the authors' two previous review papers in this field regarding the concepts and applications (Shafieian et al., 2018) and thermal efficiency improvement strategies (Shafieian et al., 2019a) of HPSCs. The authors hope that the information provided in this review paper is a valuable reference for both new and existing researchers aiming to do research or are interested in the field of solar thermal engineering and specifically HPSCs.

2. Structure and principles of operation

Figure 1 shows the overall structure of a conventional HPSC along with all its components. Two major components of HPSCs are heat pipes and evacuated glass tubes (Fig. 2). Heat pipes are basically a sealed tube made of copper which hold a specific amount of working fluid (e.g., methanol, water, and ethanol) and a wick structure (Lee, 2010). The principal heat transfer process of heat pipes is a continuous phase change of the working fluid in a way that working fluid in the evaporator section receives thermal energy and evaporates. The vapour moves upwards and reaches the condenser section in which it transfers its latent heat and turns into liquid. The working fluid in liquid state returns into the evaporator through wick structure, and the cycle goes on (Tiari et al., 2017).

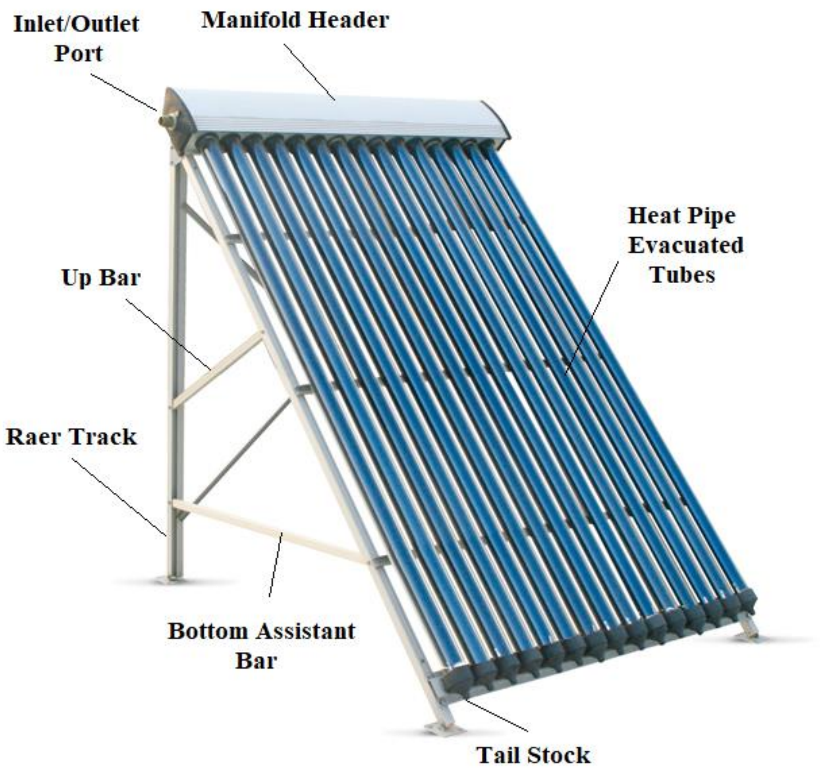

Fig. 1. Components of a typical HPSC (Shafieian et al., 2018).

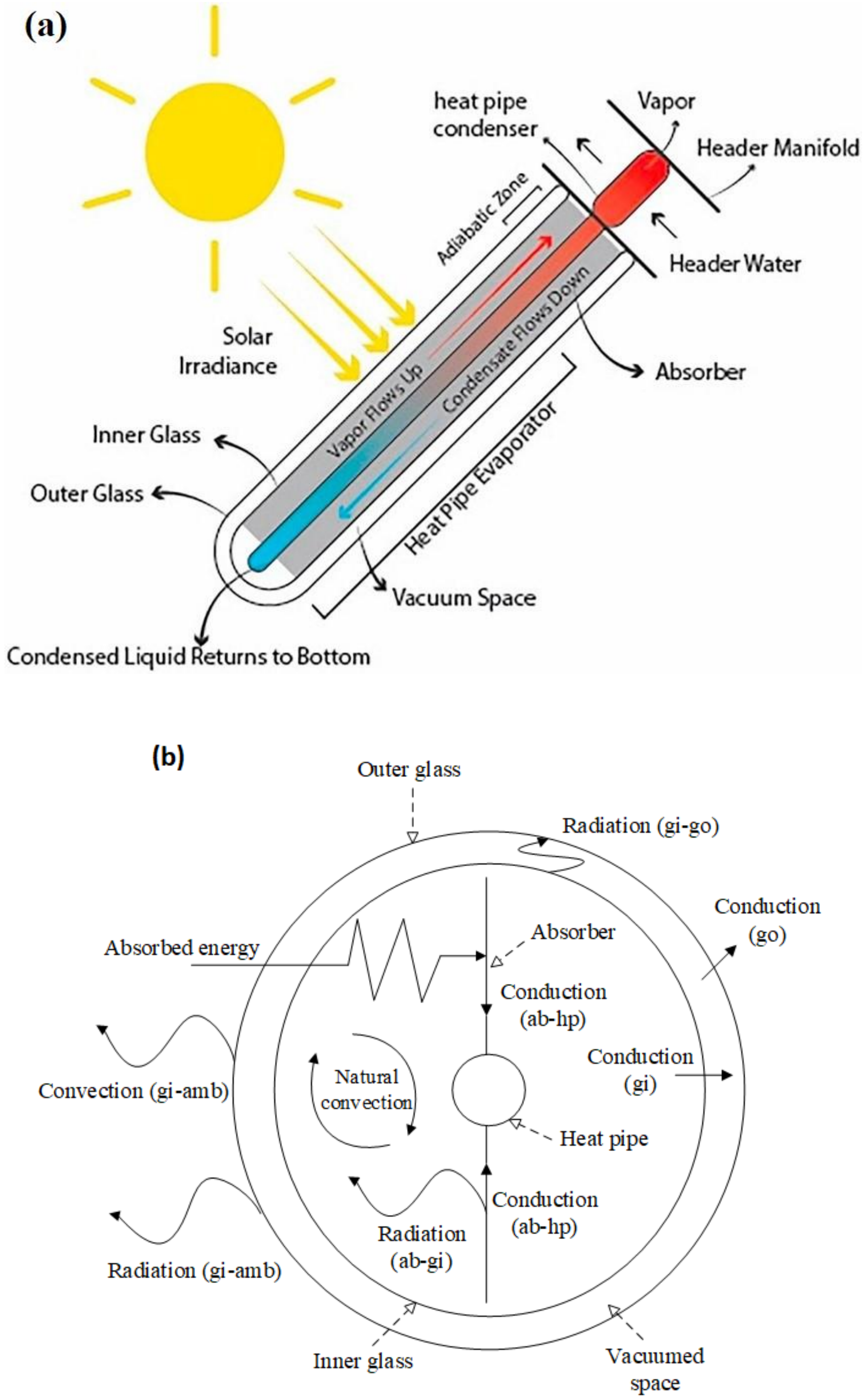

Fig. 2. Schematic of (a) structure and (b) thermal processes inside HPSCs (Elsheniti et al., 2019).

Heat pipes are one of the most efficient heat transfer devices having much higher effective thermal conductivity compared to copper itself. Choosing the proper working fluid affects the performance of heat pipes significantly and depends greatly on fluid properties, heat pipe material, operating temperature, and compatibility with the wick structure (Wallin, 2012). The thermal conductivity of the working fluid plays a key role in the performance of heat pipes, and a great share of researches has been allocated to improve this parameter by using nanofluids inside the heat pipes (Chougule et al., 2013 ; Iranmanesh et al., 2017; Mahbubul et al., 2018; Mahendran et al., 2012; Moorthy et al., 2012 ; Park and Kim, 2014; Pise et al., 2016 ; Sabiha, M.A. et al., 2015; Saravanan and Karunakaran, 2014 ; Sharafeldin and Grof, 2018; Zhao et al., 2018). Evacuated tubes of HPSCs include two layers of a special type of glass having a very small light reflection. The air between two layers of glass is extracted to create a vacuum area acting as the thermal insulation. The inner tube acts as the absorber while the outer one facilitates the passage of solar radiation (Zubriski and Dick, 2012).

The working principles of a HPSC are as follows: A portion of the stroked solar radiation is absorbed by the inner surface of vacuum tubes and used to vaporise the working fluid inside the heat pipe while another portion is dissipated back into the environment. The working fluid inside the heat pipe in the form of vapour moves upwards and reaches the condenser section of heat pipes which are located inside a manifold. The manifold acts as a heat exchanger and thermal energy is transferred from the heat pipe condensers to the solar working fluid flowing inside the manifold. The heat pipe working fluid turns into liquid state by exchanging thermal energy and returns to the evaporator section. At the same time, the temperature of the solar working fluid flowing inside the manifold increases as it moves along the manifold and over heat pipe condensers.

As the main scope of this review paper is HPSCs rather than heat pipes and evacuated tubes, more information regarding the heat pipe structure, limitations, design, characteristics, and various applications can be found in the literature (Faghri, 1995; Peterson, 1994; Reay et al., 2014; Silverstein, 1992; Smirnov, 2010). In addition, to get more information regarding the characteristics of evacuated tubes and relevant studies on ETSCs, the readers are referred to

(Bracamonte, 2017; Essa and Mostafa, 2017; Gao et al., 2013; Ghaderian and Sidik, 2017; Kumar et al., 2013; Lamnatou et al., 2012; Liu et al., 2017; Martinez et al., 2017). It is also assumed that the readers are familiar with the basic concepts and fundamentals of solar thermal engineering and heat transfer. To acquire further knowledge and information about these fields, the readers are referred to (Bergman et al., 2011; Duffie and Beckman, 2013).

3. Solar radiation

The first major step in the simulation of solar thermal processes is the determination of solar radiation characteristics. It can be limited to solar radiation intensity or expanded to specifying more detailed information including solar geometry, the direction of radiation, and the possibility of overshadowing. The traditional collector theory used for FPSCs (Daghigh et al., 2011; Duffie and Beckman, 2013) cannot be applied directly for HPSCs due to their tubular structure. Figure 3 indicates the critical angles of a HPSC (i.e. X_1, X_2, and X_3) which act as auxiliary angles to investigate the exposure angle. When the collector plane azimuth angle (γ_c) is zero and the solar azimuth angle (γ_s) varies between 0 and π/2, the following four cases of tubes shading may happen:

- No tubes shading if $\gamma_s < X_1$
- Partly tubes shading if $X_1 < \gamma_s < X_3$
- Full tubes shading if $\gamma_s > X_3$
- Half tubes shading if $\gamma_s = X_2$

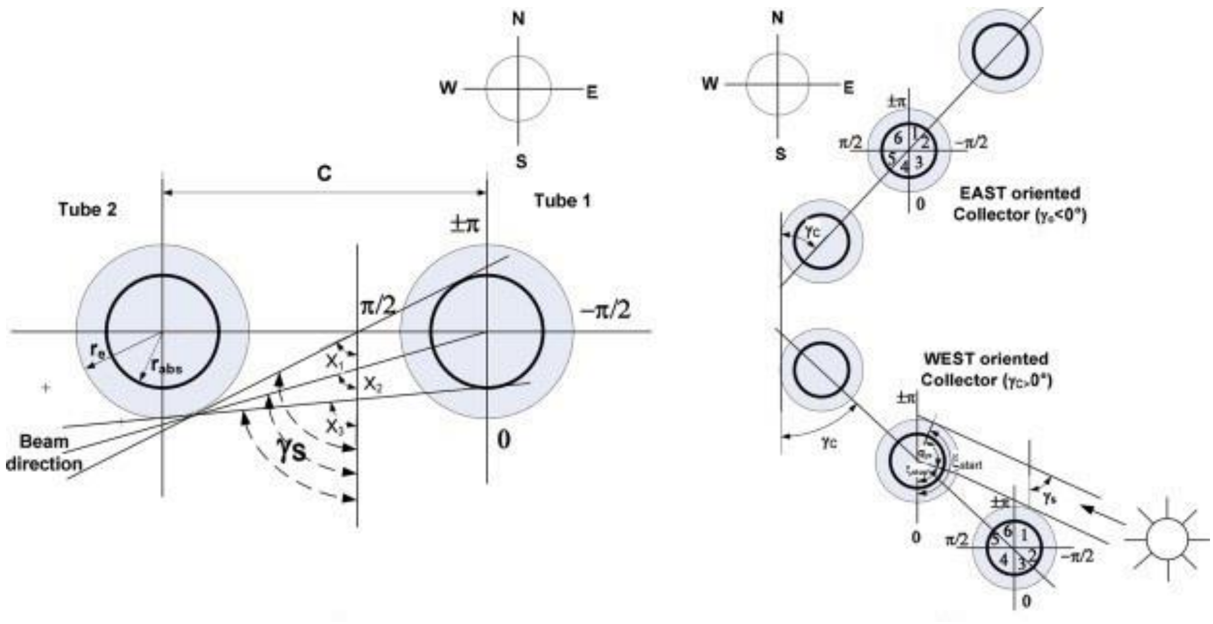

Fig. 3. Critical angles and vacuum glass sections at different orientations of a heat pipe solar collector (Fiaschi and Manfrida, 2013)

In order to determine the angle between solar radiation and the absorber surface of HPSCs (exposure angle), the following steps should be followed:

- The critical angles of X_1, X_2, and X_3 should be calculated using the solar collector azimuth angle (γ_c) and solar beam radiation direction. Depending on the orientation of the HPSC (i.e. $\gamma_c < 0$ for east and $\gamma_c > 0$ for west oriented collectors), the critical angles should be determined for all sections of all glass tubes. In Fig. 3 for example, each glass tube has been divided into 6 sections.
- The solar azimuth angle (γ_s) for each section of glass tubes should be determined.
- The initial (ξ_{start}) and final (ξ_{stop}) angles which show the exposure angle of the absorber to the beam solar radiation should be calculated.

To access all the equations required to follow the abovementioned steps along with calculation samples, the readers are referred to (Shah and Furbo, 2004).

The collected energy per square meter of the HPSC (G) can be calculated by:

$$G = (\tau\alpha)_b I_{bT} + (\tau\alpha)_d I_{dT} + (\tau\alpha)_g I_g \quad (1)$$

where, I_{bT}, I_{dT}, and I_g represent the beam, diffused, and ground reflected solar radiation, respectively.

The beam radiation intercepted by HPSCs is exactly equal to those intercepted by a flat plate solar collector utilised with a sun tracker throughout the day if the surface areas and tilt angles are the same for both collectors. Therefore, the beam radiation on the flat plate collector (I_b) can be calculated and used to determine the beam radiation on HPSCs (I_{bT}) as:

$$I_{bT} = R_b I_b \quad (2)$$

where R_b is the incident radiation coefficient and depends on the incident angle of beam radiation on HPSC and equivalent flat plate surface. The equations to find the mentioned parameters can be found in (Fiaschi and Manfrida, 2013).

The diffused solar radiation on the HPSC (I_{dT}) can be calculated by (Perez et al., 1995):

$$I_{dT} = I_d \left[(1 - f_1) \frac{1 + \cos\beta}{2} + f_1 \frac{a}{b} + f_2 \sin\beta \right] \quad (3)$$

where f_1 and f_2 represent circumsolar brilliance and horizon brilliance coefficients, respectively. More details about other parameters including brightness coefficient can be found in (Perez et al., 1995).

Assuming the isentropic ground, the ground reflected radiation on HPSC (I_g) can be calculated by (Fiaschi and Manfrida, 2013):

$$I_g = I\rho_g \left(\frac{1 - \cos\beta}{2} \right) \quad (4)$$

Another important parameter in modelling the solar radiation is the mutual passive shading factor between the evacuated glass tubes of a HPSC (X_p) which can be determined by:

$$X_p = 1 - Max\left[0, \left(D_{ab}\cos\theta_{vgt} - C\cos\theta_1\right)/D_{ab}\cos\theta_{vgt}\,(n-1)/n\right] \quad (5)$$

In the abovementioned equation, n represent the number of heat evacuated glass tubes. The details about other parameters can be found in (Fiaschi and Manfrida, 2013).

4. Steady state models

The first group of mathematical models to simulate the performance of HPSCs are steady state models. They are simple, have relatively low computing times, and are easily connected to an installation model. On the other hand, these models are not flexible and can be used under specific conditions and also overestimate or underestimate the solar potential of the system. The main assumption in steady state models is that either the parameters of the system are independent of time or the time constant of the parameters responses to the changes of climatic conditions is relatively high (Daghigh and Shafieian, 2016a). Each part of this section starts with a concise description of the steady-state mathematical modelling strategies and is followed by the studies which have implemented these models.

4.1. One-dimensional energy balance method

In this method, the transferred energy to the solar working fluid (Q_u) is considered as the absorbed solar energy minus the heat losses from the absorber surface:

$$Q_u = A_c F_R[G(\tau\alpha) - U_L(T_m - T_a)] \quad (6)$$

where T_m represents the average values of the solar working fluid inlet and outlet temperatures. The thermal efficiency of the HPSC is then defined as:

$$\eta_c = \frac{Q_u}{GA_c} = F_R\left[\tau\alpha - \frac{U_L(T_m - T_a)}{G}\right] \quad (7)$$

Under real operational conditions, heat losses of the HPSC are functions of solar working fluid and ambient temperatures which can be expressed as (Duffie and Beckman, 2013):

$$F_R U_L = C_1 + C_2\,(T_m - T_a) \tag{8}$$

By the combination of Eq. (7) and (8), the thermal efficiency of HPSCs can be re-written as:

$$\eta_c = F_R\tau\alpha - C_1\,\frac{(T_m - T_a)}{G} - C_2\,\frac{(T_m - T_a)^2}{G} \tag{9}$$

In order to find the constants C_1 and C_2, the experimental data including the climatic and operational conditions as well as thermal efficiency of the HPSC should be used. Du et al. (Du et al., 2013) designed and manufactured an experimental platform to investigate the thermal performance of HPSCs. The experimental analysis showed that the values of C_1 and C_2 constants were 0.325 and 0.0238 under climatic conditions of Nanjing, China.

Xiao et al. (Xiao et al., 2012) proposed a new procedure for theoretical performance evaluation of flat plate HPSCs with cross flow heat exchanger. The HPSC was divided into different sections (Fig. 4) named B_0–A_1, B_1–A_2, …, B_N–A_0. The energy balance method was used and the equations were solved for each section of HPSC using Fourier's law of heat conduction:

$$\frac{d^2T}{dx^2} + \frac{G - U_l(T - T_a)}{k\delta} = 0 \tag{10}$$

To keep this review paper as concise as possible, the final solution for the first section (B_0–A_1) is presented below, and the readers are referred to (Xiao et al., 2012) for further information regarding the solution procedure and equations of other sections.

$$T = C_1 exp\left(\sqrt{\frac{U_l}{k\delta}}x\right) + C_2 exp\left(-\sqrt{\frac{U_l}{k\delta}}x\right) + \frac{G}{U_l}T_a \tag{11}$$

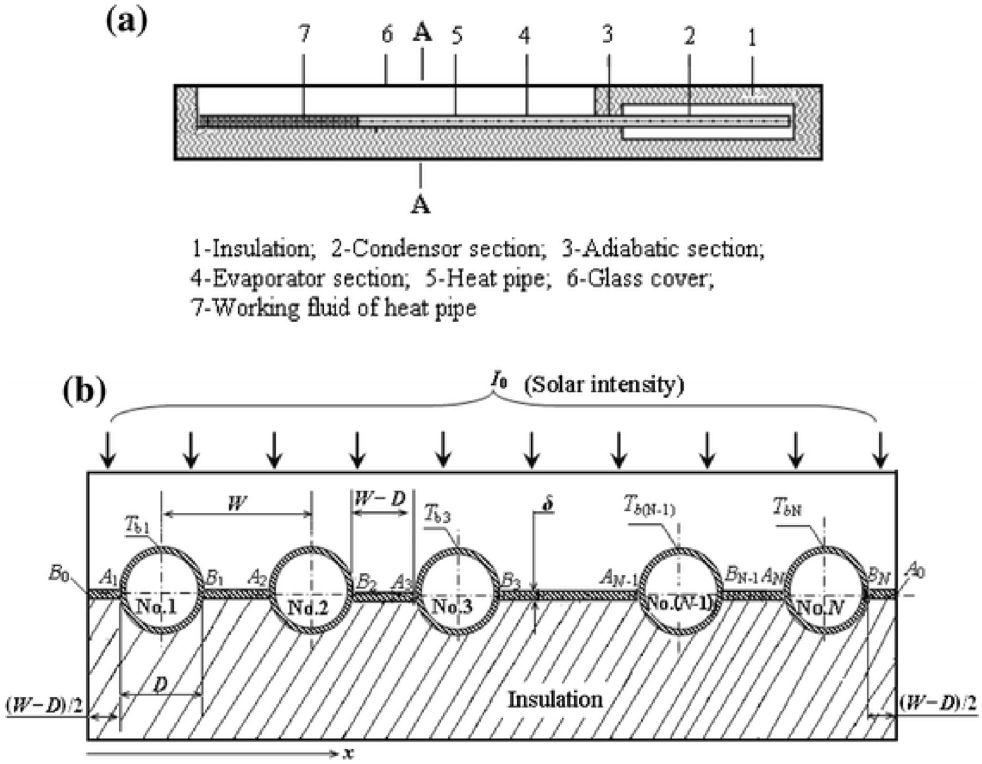

Fig. 4. (a) Heat pipe solar collector and (b) A-A cross view (Xiao et al., 2012)

Another technique of using this method is writing the energy balance for all components of the HPSC and solving all the equations together. Huang et al. (Huang et al., 2019a) used this technique to study the effect of using heat shields on the thermal performance of HPSCs. For instance, the energy balance for a glass tube can be written as (Huang et al., 2019a):

$$\alpha_g A_g G + Q_{r,p-g} = A_g \varepsilon_g \sigma\left({T_g}^4 - {T_{eq}}^4\right) + A_g h_g\left(T_g - T_a\right) \tag{12}$$

This equation states that the amount of absorbed solar energy by glass tube plus the energy received from the absorber through radiation equals to radiation energy dissipated back to the environment plus convection thermal losses. The same technique can be applied to other components by writing the input and output energies to that specific control volume. For further information regarding the equations for other components and calculation processes, the readers are referred to (Huang et al., 2019a). Moreover, Table 1 summarises the significant previous studies in which the one-dimensional energy balance method has been applied to develop a mathematical model.

Table 1. Summary of recent studies in which the one-dimensional energy balance method has been applied

Description	**Remarks and key findings**	**Reference**
A theoretical model was developed to study the influence of six various solar working fluids on the performance of HPSCs under climatic conditions of Uşak, Turkey.	• The results of the simple developed model were in good agreement with experimental data. • The maximum relative error of 10% for energy and exergy efficiency predictions.	(Ersöz, 2016)
A theoretical model was developed to evaluate the efficiency of a new configuration of HPSCs using thin membrane heat pipes	• The best equation to describe the efficiency of the HPSC based on operational and climatic conditions was η_c = -460.4 $\frac{(T_m-T_a)}{G}$ + 70.094.	(Riffat et al., 2005)
A theoretical model was proposed to investigate the thermal performance of a new configuration of HPSCs equipped with microchannel heat pipe	• The developed theoretical model was used to evaluate the effect of climatic conditions (e.g., solar radiation), structural parameters (e.g., glazing covers, and operational inputs (e.g., solar working fluid mass flow rate) on the energy performance of HPSCs.	(Diallo et al., 2019)

The traditional collector theory (Duffie and Beckman, 2013) used for flat plate solar collectors was modified to be applicable for HPSCs.	• The theoretical and experimental results almost match with the maximum relative error of 8.6%.	(Ismail and Abogderah, 1998)
The performance of a new HPSC including wick-assisted heat pipes and a novel manifold was evaluated both theoretically and experimentally.	• Effect of different parameters such as heat pipe temperature, useful heat transfer rate, and collector outlet temperature on the thermal efficiency of the new configuration was studied theoretically. • The maximum difference between the theoretical and experimental collector outlet temperature was 4 °C.	(Azad, 2009)
A theoretical model was developed to analyze the performance of wickless HPSCs in a solar water heating system.	• The performance of the system depends greatly on the design phase, and therefore, the optimization of physical and operational parameters plays a crucial role in the performance enhancement of the system. • The comparison between theoretical and experimental amounts absorbed energy by the solar working fluid showed the maximum difference of 200 W.	(Mathioulakis and Belessiotis, 2002)

A mathematical model was used to compare heat pipe and evacuated tube solar water heating systems technically and economically in Hong Kong	• The daily and annual thermal performance of heat pipe solar system was slightly better than the evacuated tube system. • The payback periods of the two systems were almost the same.	(Chow et al., 2011)

4.2. Thermal resistance network method

The role of theoretical simulations in heat and mass transfer fields has become much more significant by the development of computer technology. However, the direct simulation of HPSCs is a complex procedure due to the continuous phase change processes which occur inside the heat pipes. The thermal resistance network method was proposed as an alternative approach to make the process of numerical simulations more convenient. In this section, the thermal resistance network method which has been widely used to theoretically model the thermal processes inside the components of HPSCs (i.e., collector absorber, heat pipe, and manifold) is explained in details. In addition, the previous studies applied this method in their investigations along with their findings are summarized.

The theoretical model involves three processes which are (i) the conversion of solar energy into thermal energy using absorber surface, (ii) transportation of the absorbed energy from evaporator section to condenser section of heat pipes, and (iii) thermal energy exchange between heat pipes and solar working fluid through the manifold section (Wang et al., 2012), which are explained below in the separate subheading. The major assumptions considered in this mathematical model are as follows (Azad, 2008; Daghigh and Shafieian, 2016a; Shafieian et al., 2019b):

- The heat loss from the manifold section to the ambient area is negligible.
- There is no temperature gradient in the heat pipes' longitudinal direction. The only existing temperature gradient is radial.
- There is no contact resistance between absorber surface, heat pipe's wall, and wick structure.
- There are no nuclear or chemical reactions.
- The indirect effects of kinetic and potential energies are not considered.

- Energy transfer towards the system has a positive sign.
- The solar working fluid has a constant specific heat capacity.

4.2.1. Absorber

The heat transfer processes occur inside the evacuated tubes of HPSCs are shown schematically in Fig. 5. A part of the solar energy which is radiated on the collector surface is absorbed while the other part is dissipated back to the ambient area. The absorbed part of solar radiation will then be transferred using heat pipes. There are several thermal resistances between the absorber surface and the environment which are discussed in this section.

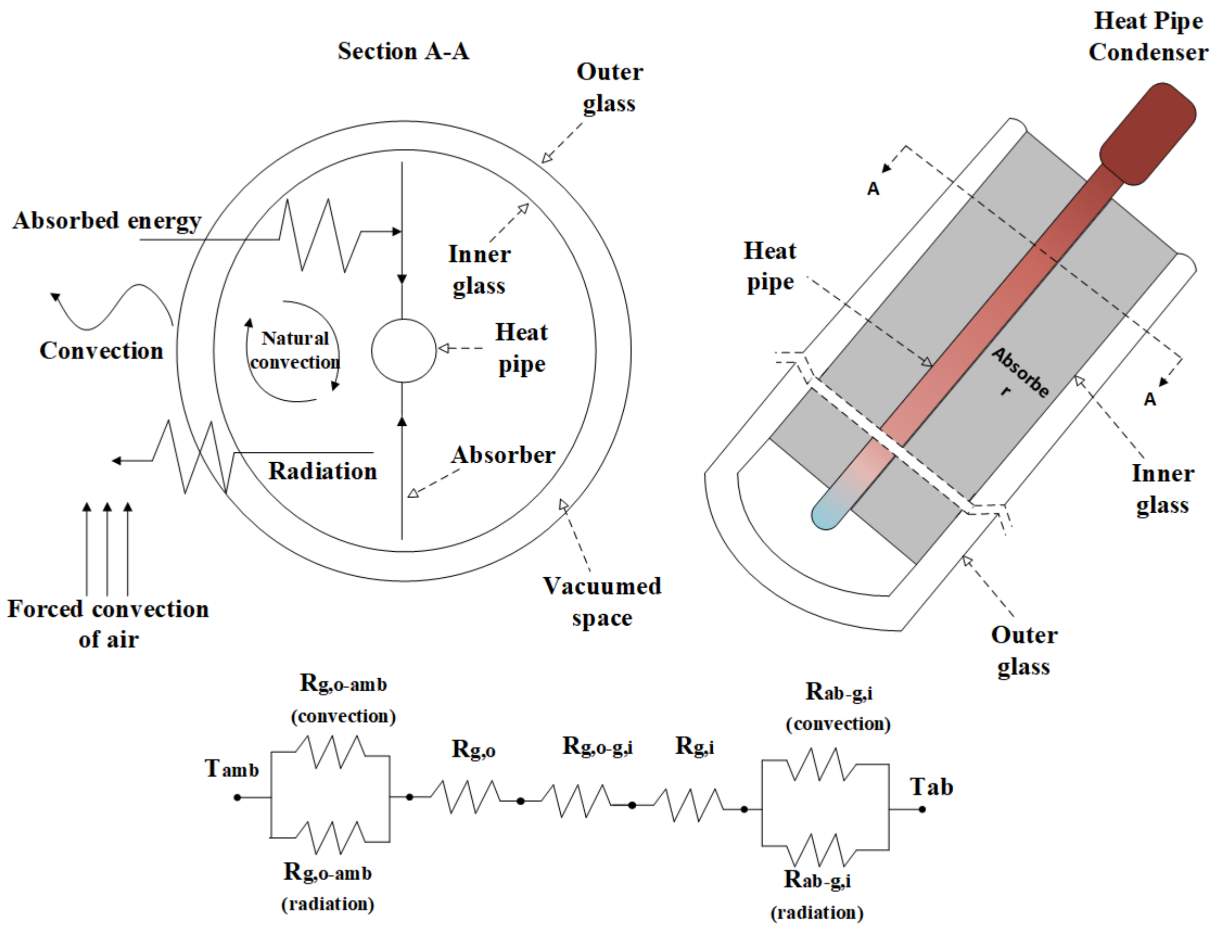

Fig. 5. Heat transfer processes along with their thermal resistances in the vacuum tubes of HPSCs (Shafieian et al., 2019b).

The solar energy absorption and the process of heat loss are described by the following thermal energy balance (Wang et al., 2012):

$$Q_{ab} = Q_{en} - Q_{loss} \tag{12}$$

Parameters Q_{en} (W), and Q_{loss} (W) represent the stroked solar energy on the collector surface and heat loss to the environment, respectively, which can be calculated by (Lienhard IV and Lienhard, 2008; Riffat et al., 2005):

$$Q_{en} = \tau_{go}\tau_{gi}\alpha_c A_{ab} N_{hp} G \tag{13}$$

$$Q_{loss} = \frac{T_{ab} - T_{amb}}{R_t} \tag{14}$$

Parameter R_t (K/W) in the abovementioned equations represents the summation of all the thermal resistances exist between the ambient and absorber surface (Fig. 5). These thermal resistances include: (i) $R_{ab\text{-}gi}$ which is the combination of radiation and natural convection between the absorber surface and the inner glass; (ii) R_{gi} which represents the conductive thermal resistance of the inner glass; (iii) $R_{gi\text{-}go}$ which is the radiative thermal resistance exists between the inner and outer glass; (v) R_{go} which represents the conductive thermal resistance of the outer glass; and (iv) $R_{go\text{-}amb}$ which is the combination of radiative and forced convective thermal resistances exist between the outer glass and environment. More details regarding the formulas to calculate the mentioned thermal resistances can be found in (Shafieian et al., 2019b).

4.2.2. Heat pipe

The total thermal resistance of a heat pipe (R_{hp}) is defined as the summation of all the thermal resistances exist between the evaporator and condenser sections (Fig. 6).

$$R_{hp} = R_h + R_{f,h} + R_{w,h} + R_{wi,e} + R_{i,e} + R_v + R_{i,c} + R_{w,c} + R_{f,c} + R_c \tag{15}$$

In the abovementioned equation, R_h and R_c represent the convective thermal resistance of evaporator and condenser sections, respectively. The residuals which are formed on the outer wall of the evaporator and condenser sections are represented by $R_{f,h}$ and $R_{f,c}$, respectively. Parameters $R_{w,h}$ and $R_{w,c}$ are the conductive thermal resistances of evaporator and condenser sections, respectively. The conductive thermal resistance of wick structure is represented by $R_{wi,e}$ while $R_{i,e}$ and $R_{i,c}$ are evaporation and condensation thermal resistances, respectively. For more information regarding the mentioned thermal resistances and their calculation procedure, the readers are referred to (Azad, 2008; Dunn and Reay, 2012; Jahanbakhsh et al., 2015; Lee, 2010; Reay et al., 2013).

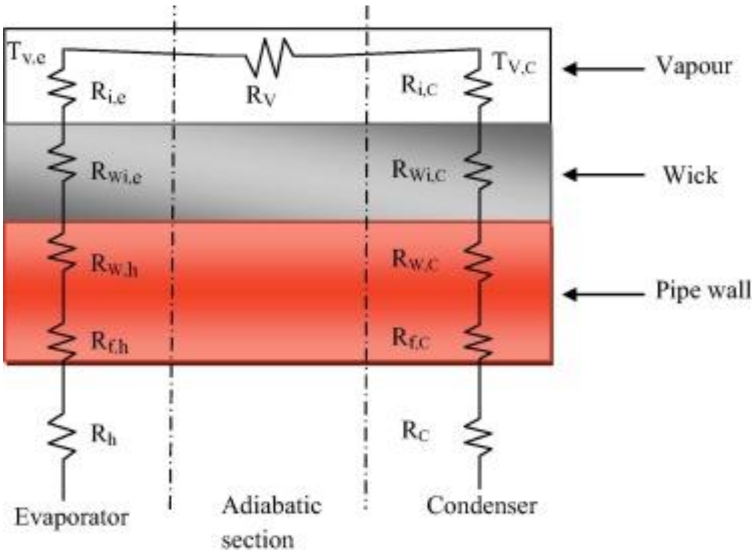

Fig. 6. Thermal resistance network of a heat pipe (Jahanbakhsh et al., 2015).

4.2.3. Manifold section

The solar working fluid entering the manifold section of a HPSC reaches the first condenser at its lowest temperature. Due to heat transfer between solar working fluid and heat pipe condensers, the temperature of solar working fluid increases as it moves toward the next condenser. Therefore, the outlet temperature of one consider is considered as the inlet temperature of the next one in the row (Fig. 7). The heat pipe working fluid condensation process occurs at almost a constant temperature; therefore, the effectiveness-NTU (Number of Transfer Units) (Bergman et al., 2011) method is applied to model the heat transfer inside the manifold:

$$T_{o,n} = T_{i,n} + \varepsilon_n (T_{c,n} - T_{i,n}) \quad (16)$$

where $T_{i,n}$ and $T_{c,n}$ are the solar working fluid inlet and condensor temperatures, respectively, while ε_n represents the effectiveness of the n[th] heat pipe.

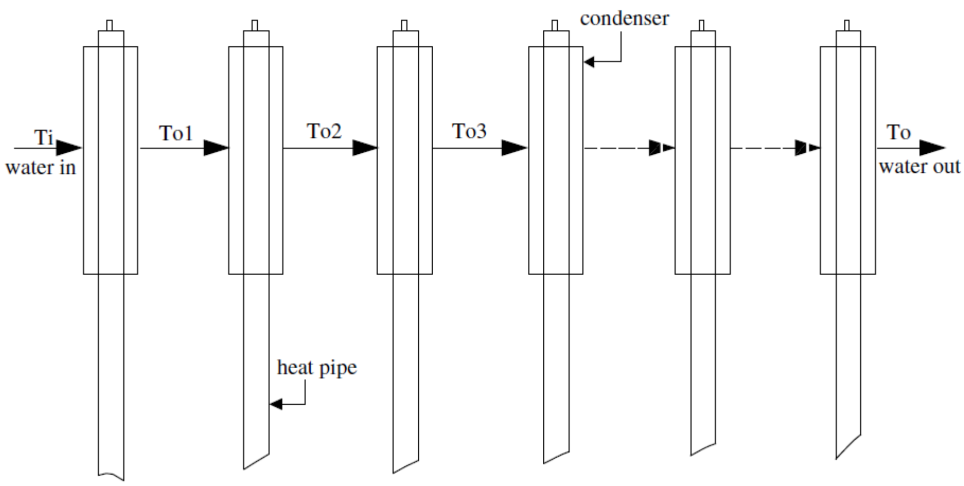

Fig. 7. Schematic diagram of condensers arrangement inside the manifold of a HPSC (Azad, 2008)

4.2.4. Applications of thermal resistance network method

The thermal resistance network method has been widely used in the literature due to its simplicity and satisfactory accuracy. One of the first studies using this method was carried out by Azad who investigated the performance of a HPSC in a water heating system (Azad, 2008). The theoretical results were compared to experimental data and indicated a good agreement with the maximum relative error of 5%. In addition, the evaporator to condenser length ratio was studied theoretically, and its optimum value was proved to be 8.25. Shafieian et al. used this method to find the optimum surface area of a HPSC in a solar water heating system aimed to meet the hot water demand in cold seasons (Shafieian et al., 2019b). According to the theoretical results, the optimum number of evacuated tubes was obtained to be 25 for climatic conditions of Perth in Western Australia. Figure 8 shows the computational process flowchart of the developed mathematical model based on the thermal resistance network method.

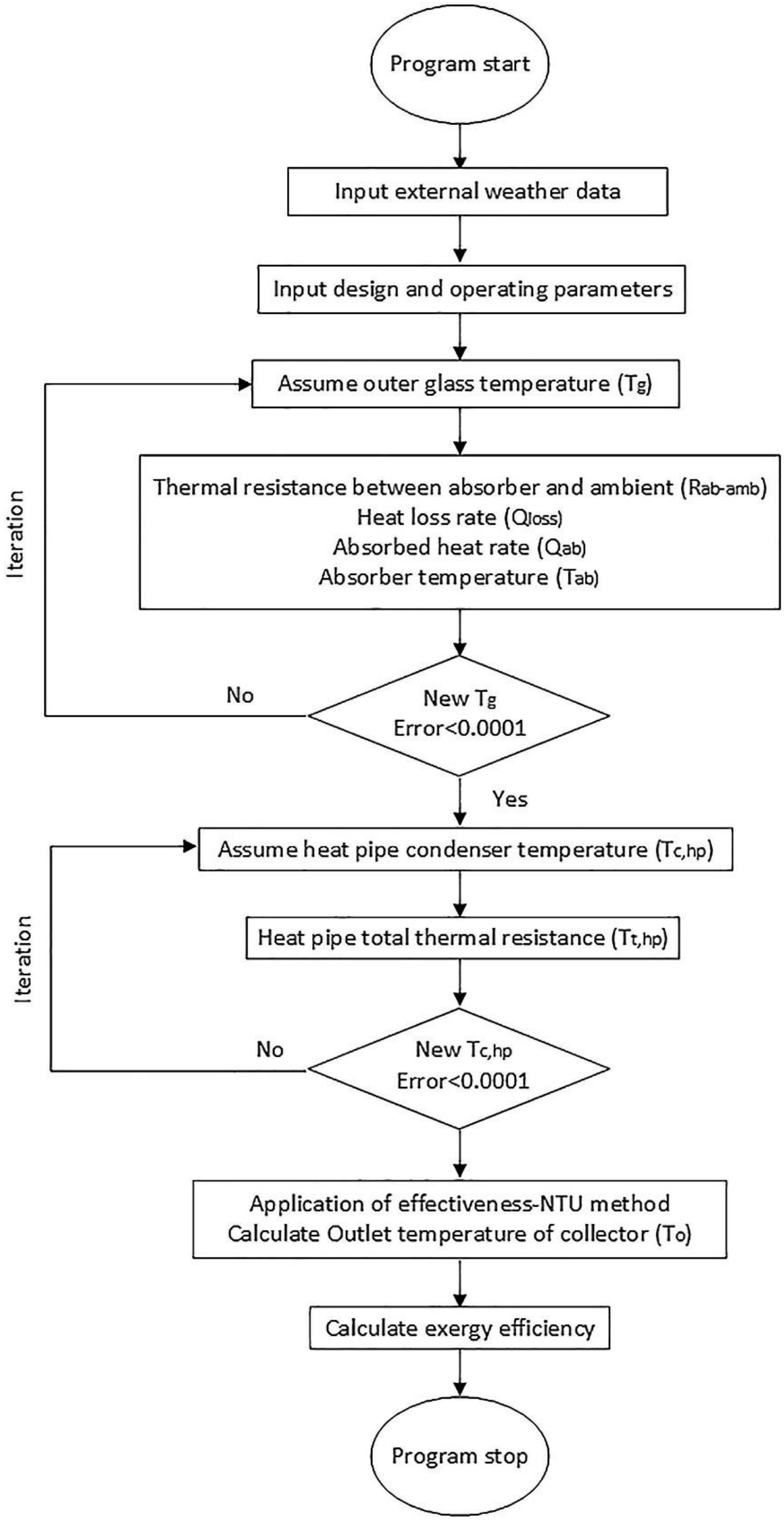

Fig. 8. Computational process flowchart of the thermal resistance network method (Shafieian et al., 2019b)

Elsheniti and Elsamni witnessed the high relative error between the theoretical modelling using this method and experimental data when the inlet temperatures of the HPSC was high (Elsheniti et al., 2019). To resolve this issue, a new term for the collector thermal mass was added to the conventional mathematical model. This change improved the accuracy of the model and the maximum relative error was reduced from 12.5% to 4.4%. Figure 9 shows the proposed modified thermal resistance network for modeling HPSCs. In this figure, R2-R9 are the heat pipe thermal resistances and further information regarding them can be found in (Elsheniti et al., 2019)

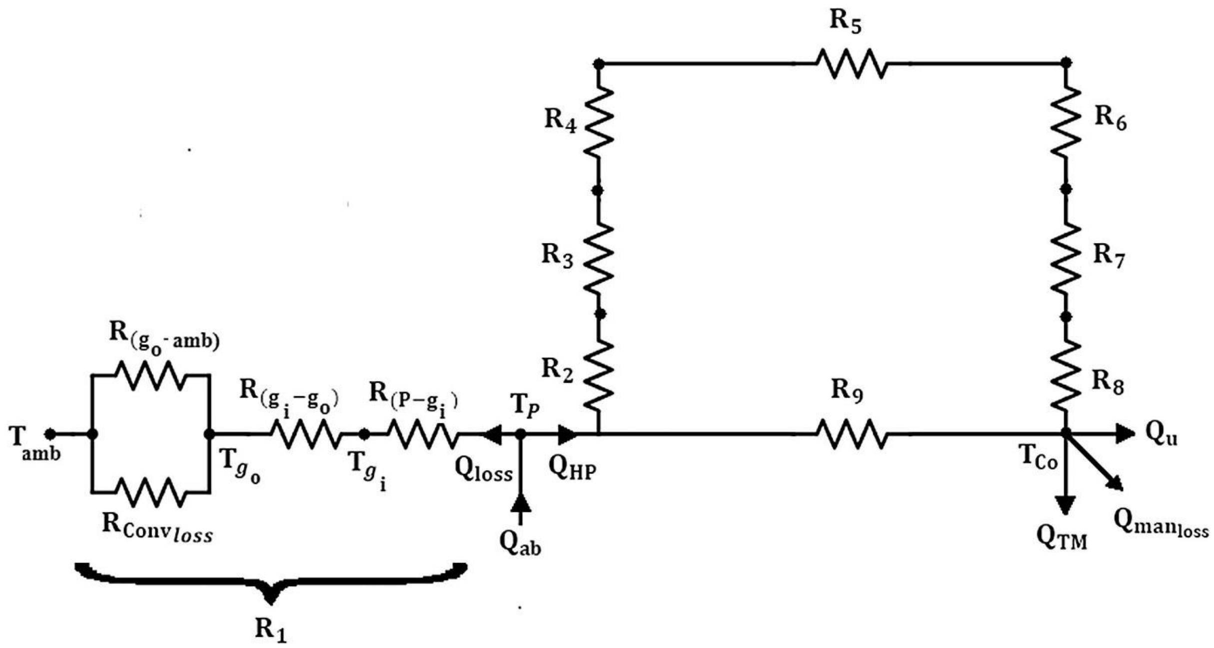

Fig. 9. Modified thermal resistance network for modelling HPSCs (Elsheniti et al., 2019)

Jahanbakhsh et al. used the thermal resistance network method to investigate the application of water-ethanol solution as the heat pipe working fluid in a HPSC (Jahanbakhsh et al., 2015). The slight effect of using a wick structure on the performance improvement of HPSCs was one of the major findings of this study. In a theoretical and experimental study, the performance of a HPSC in a solar air heating system was investigated (Daghigh and Shafieian, 2016b). The developed model was used to find the optimum surface area of the collector under climatic conditions of Sannandaj, Iran. The optimum area of the collector in this system was concluded to be equal to the surface area of a collector with 30 evacuated tubes.

A theoretical model based on the thermal resistance network method was developed to study the performance of HPSCs in summer (Daghigh and Shafieian, 2016d). The results obtained from the developed model was validated with experimental data and indicated the maximum relative error of 8.4%. The best equation to describe the efficiency of the HPSC was reported to be y = 9404.2x^3 - 160.65x^2 - 4.8676x + 0.3435, where x and y represent $\frac{(T_m - T_a)}{G}$ and thermal efficiency, respectively. In a similar study, the thermal resistance network method was used to simulate the thermal processes of a HPSC in a multipurpose solar water heating-drying system (Daghigh and Shafieian, 2016a). The maximum divergence of the theoretical results from the experimental data was reported to be 6.4% in this study.

Brahim et al. proposed the combination of energy balance equations and thermal resistance network method to simulate the operational processes of a flat plate HPSC (Taoufik et al., 2013). Their study was mainly focused on the effect of collector physical parameters such as spacing gap between the absorber plate and the glass cover, tube spacing, and absorber plate emissivity on the thermal efficiency of the collector. In addition, the performance of the system using different working fluids was studied theoretically using the proposed model.

Zhang et al. combined the thermal resistance network method with mass, momentum, and energy conservation equations to develop a mathematical model for performance evaluation of flat plate HPSCs (Zhang et al., 2017). The expanded equations were discretized using the finite volume method and second-order upwind difference scheme was applied to solve the discrete equations. A hexahedral structured grid was created (Fig. 9) and its independency was investigated and confirmed. The proposed model was validated using several verification approaches showing the maximum deviation of 2.5% for thermal efficiency.

Sivakumar used the thermal resistance network method to find the optimum ratio of the heat pipe evaporator to condenser in HPSCs and also investigated the performance of the solar system by changing the operational parameters such as solar working fluid inlet temperature (Sivakumar et al., 2012). The effects of adding heat shields to HPSCs on their thermal efficiency was investigated by Huang et al. (Huang et al., 2019b) using a one-dimensional steady state model. The theoretical model was based on the combination of energy balance equations and thermal resistance network method.

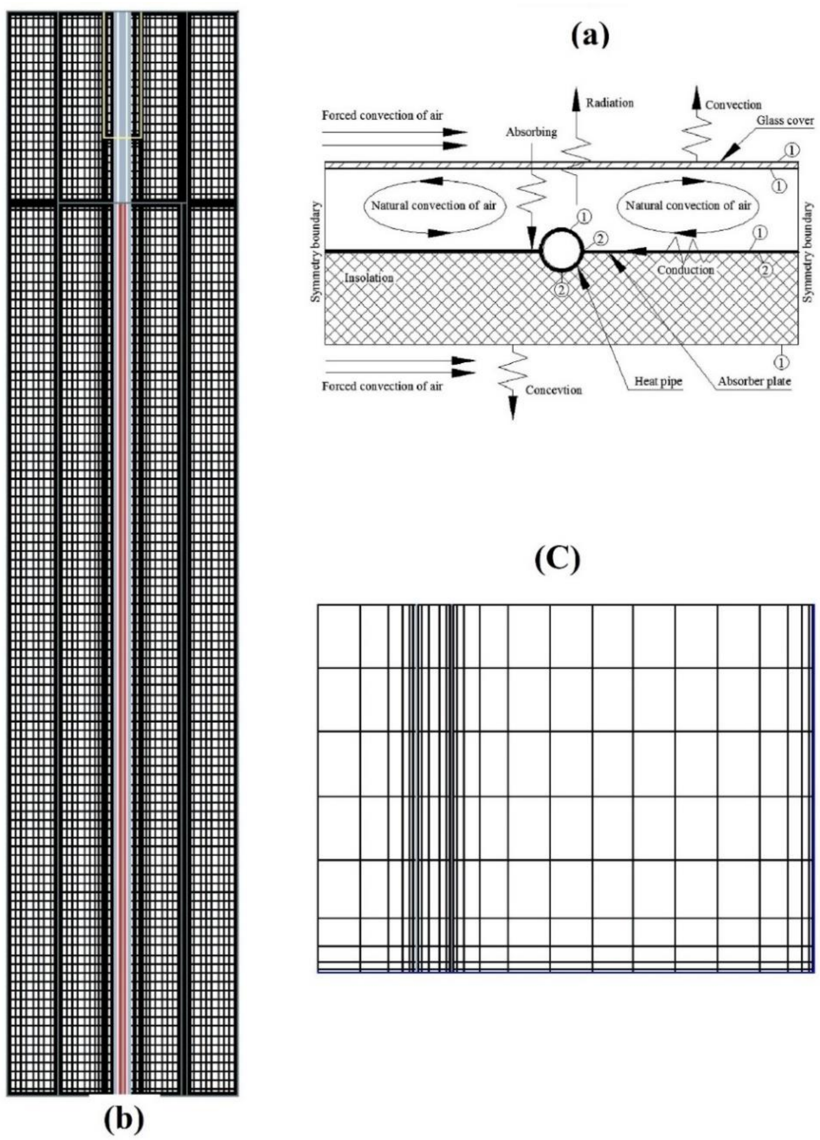

Fig. 9. (a) The thermal processes inside the HPSC, (b) Mesh distribution of HPSC in x direction; (b) Enlarged view boundary layers around heat pipe (Zhang et al., 2017).

5. Dynamic models

The second group of mathematical models to simulate the performance of HPSCs are dynamic models. Dynamic models describe the solar collector and its thermal process in more details compared to steady state models. In addition, they predict the solar potential of the system more accurately and are useful tools to determine the control strategy of the solar systems. On the other hand, these models are complex and require more computation time, cost, and effort compared to steady state models.

One of the most comprehensive dynamic models was proposed by Arab and Abbas aiming to optimise the thermal performance of HPSCs (Arab and Abbas, 2013). The proposed model was developed based on energy balance equations. The HPSC was divided into six different sections as are shown in Fig. 10 and labelled as b1-b6. The model consisted of two sub-models (i.e. solar collector and heat pipe) developed by different methods and integrated to form the overall model. In the first sub-model, the energy balance equations were written for each section separately, while in the second sub-model, the thermal resistance network was applied and a set of differential–algebraic equations were solved. Figure 11 shows the thermal resistance network of the entire HPSC. Here, only the equation for section b1 is presented (Eq. 17) and the readers are referred to (Arab and Abbas, 2013) for further information.

$$M_g C_g \frac{dT_{g,i}}{dt} = A_p\left(\alpha_g + \rho_g \tau_g\right)G + \frac{A_p \sigma}{\frac{1}{\varepsilon_p} + \frac{1-\varepsilon_g}{\beta \varepsilon_g}}\left(T_{p,i}^4 - T_{g,i}^4\right) + A_p \sigma \varepsilon_g \left(T_{s,i}^4 - T_{g,i}^4\right) + A_g h_a\left(T_a - T_{g,i}\right) \tag{17}$$

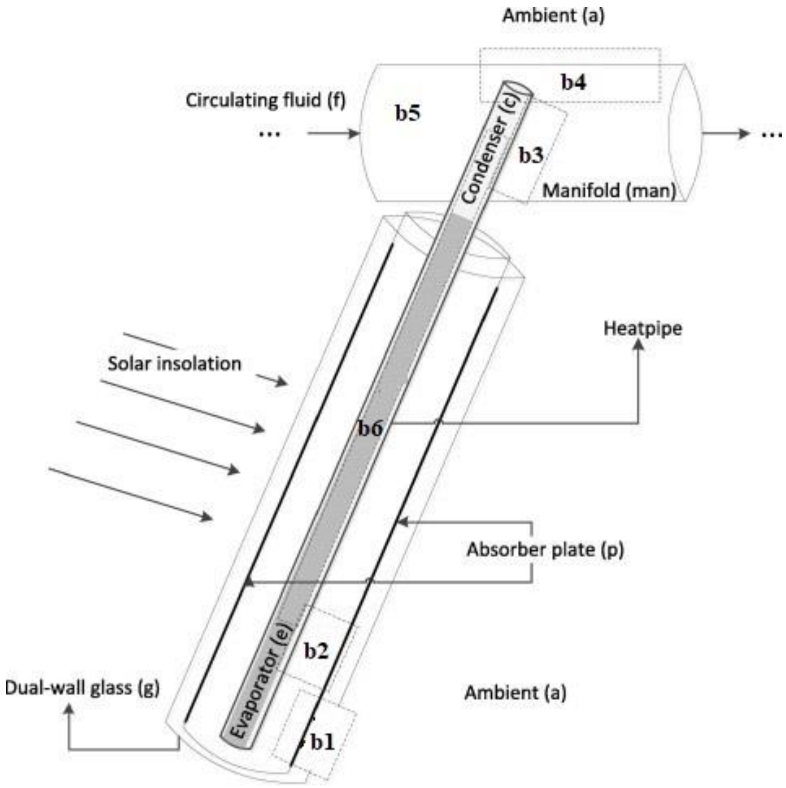

Fig. 10. The HPSC divided into different sections for theoretical analysis (Arab and Abbas, 2013)

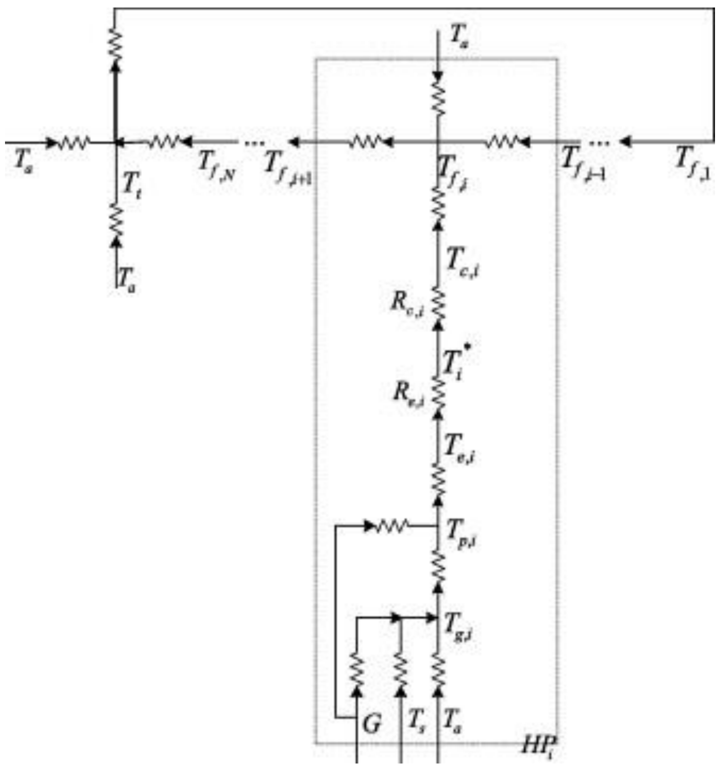

Fig. 11. The thermal resistance network of the entire HPSC (Arab and Abbas, 2013)

Brahim et al. used the combination of energy conservation and thermal resistance network method in dynamic modes to analyze the performance of HPSCs (Brahim et al., 2014). The HPSC was split into several isothermal nodes as shown in Fig. 12. The thermal energy exchanged between two interconnected nodes (i.e. named *i* and *j*) was considered as a linear function of the temperature difference between the nodes:

$$Q_{i,j} = k_{i,j}(T_j - T_i) \tag{18}$$

where $k_{i,j}$ is the heat transfer coefficient which varies by time and depends linearly on the nodes' temperature. In addition, the energy balance of i^{th} node was written as:

$$M\frac{dT_i}{dt} = \sum_j k_{i,j}(T_j - T_i) + E_i \tag{19}$$

where M_i and E_i represent the multiplication of mass and specific heat and energy gain of i^{th} node, respectively. Equation (18) was written for all nodes as a function of time resulting in a set of equations which were solved to determine the temperature of all the nodes. Moreover, the backward-difference scheme discretization method was applied to solve Eq. (19).

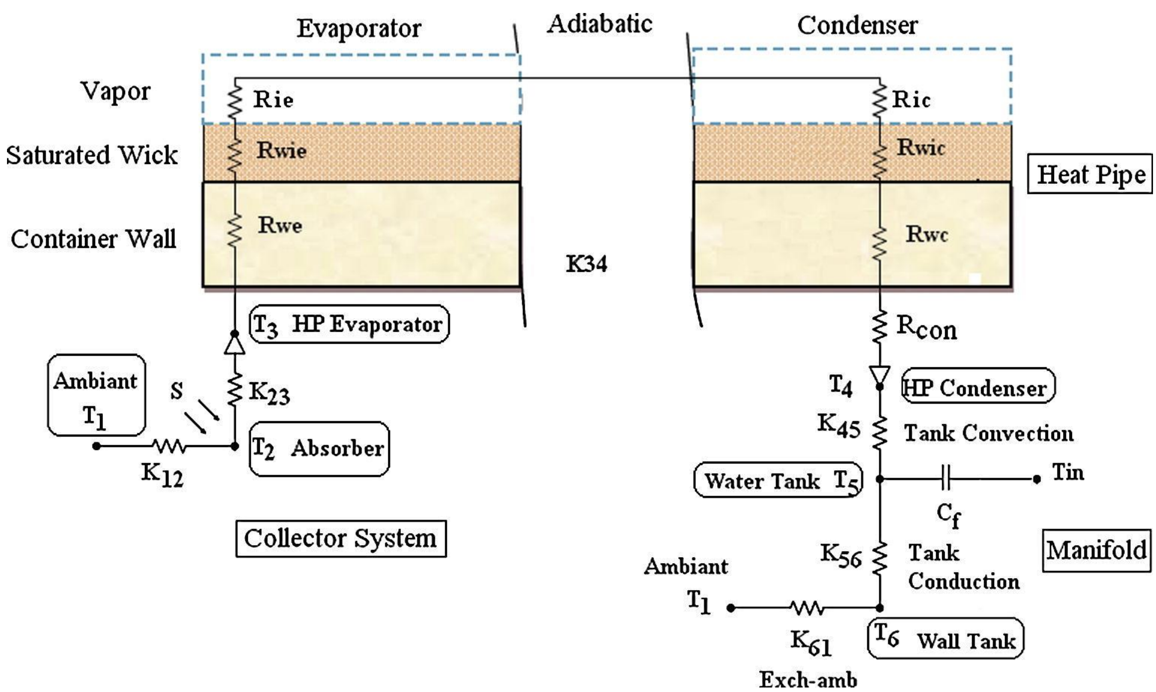

Fig. 12. The thermal resistance network of the entire HPSC (Brahim et al., 2014)

Naghavi et al. developed a mathematical model to study a novel heat pipe solar water heating system consisting of a conventional HPSC and phase change materials (Naghavi et al., 2015). The solar radiation in this study was modelled using the equations presented in Section 3. Solving the heat transfer equations, the temperature along the heat pipes of a HPSC for evaporator, adiabatic, and condenser sections were obtained as Eq. (20)-(22), respectively. Moreover, Table 2 summarises the significant previous studies in which various dynamic techniques have been employed.

$$T_{hp}(x,t) = T_{mf} + \frac{G}{2\pi L_c}\left[\left(\frac{\ln(r_{o,hp}/r_{i,hp})}{k_w} + \frac{\ln(r_{o,wick}/r_{i,wick})}{k_{eff}}\right)\left(1 + \frac{L_c}{L_e}\right) + \frac{1}{h_{c,wf} + r_{c,hp}}\right] \tag{20}$$

$$T_{hp}(x,t) = T_{mf} + \frac{G}{2\pi L_c}\left[\left(\frac{\ln(r_{o,hp}/r_{i,hp})}{k_w} + \frac{\ln(r_{o,wick}/r_{i,wick})}{k_{eff}}\right) + \frac{1}{h_{c,wf} + r_{c,hp}}\right] \tag{21}$$

$$T_{hp}(x,t) = T_{mf} + \frac{G}{2\pi L_c h_{c,wf}} \tag{22}$$

Table 2. Summary of recent studies in which dynamic techniques have been used.

Description	Remarks and key findings	Reference
A dynamic model based on energy balance equations was proposed to investigate the thermal performance of a HPSC	• The equations in this study were expanded for glass tubes, absorber, heat pipe evaporator and condenser, and solar working fluid. • The maximum error of 2% between theoretical and experimental outlet temperature of the collector showed the good accuracy of the developed model in simulating the thermal processes occurs in a HPSC	(Bourdoukan et al., 2008)
A new structure of HPSCs was proposed and studied by developing a transient theoretical model based on energy conservation equations.	• The finite difference scheme and the explicit method was applied to solve the energy balance equations. • Several recommendations were given to improve the thermal efficiency of HPSCs based on the developed model.	(Wei et al., 2013)
A transient axisymmetric numerical simulation was performed to investigate the thermal performance of a capillary driven HPSC.	• Lattice Boltzmann and Double Distribution Function methods were applied to solve the equations numerically.	(Grissa et al., 2017)

	• A comprehensive database and several guidelines were provided to optimize the performance of HPSCs.	
An automated dynamic algorithm was developed to study the performance of a HPSC in a solar water heating system to meet a real water consumption pattern over a year.	• The algorithm was able to control the HPSC, water storage tank temperature, operation of pumps, water extraction, and operation of auxiliary electric heater. • The flowchart of the developed controlling algorithm which is shown in Fig. 13 can be used in similar studies.	(Ayompe and Duffy, 2013)
The thermal efficiency of a wickless HPSC was investigated using a dynamic model based on energy balance equations.	• The theoretical results indicated that the optimum solar working fluid mass flow rate is close to the recommended value by ASHRAE standard • The results showed the importance of optimizing the number of evacuated glass tubes in HPSCs.	(Hussein, 2007)
An unsteady mathematical model was proposed to evaluate the daily and annual technical and economic performance of HPSCs in water heating systems of Hong Kong.	• The developed model was validated in both summer and winter modes and showed its accuracy and capability in both cold and hot seasons.	(Chow et al., 2011)

An unsteady state theoretical model was proposed to simulate the natural heat transfer process of thermosiphon heat pipe solar water heating systems.	• The capability of the developed model to be used as the optimization tool for the HPSWH system was verified.	(Hussein, 2002, 2003)

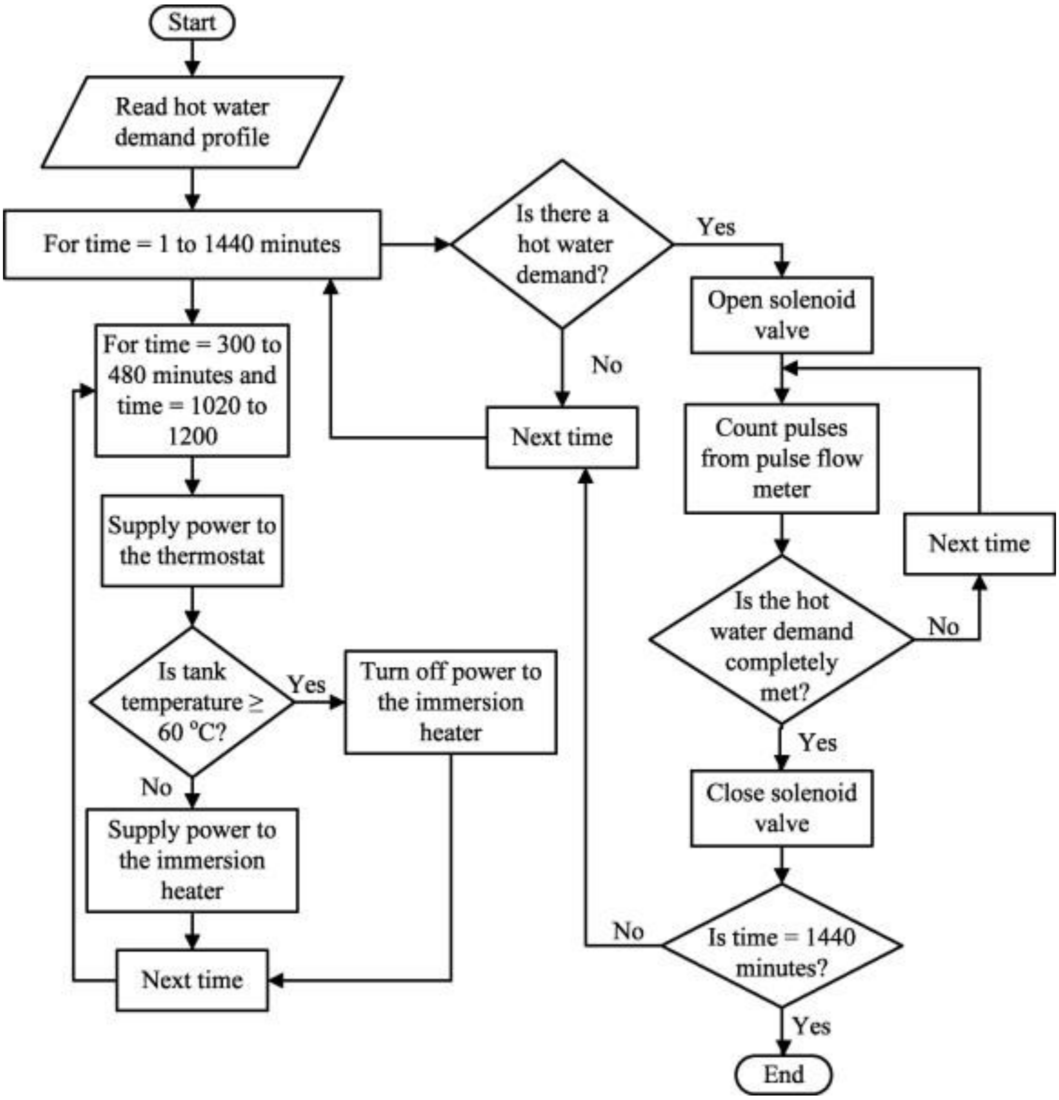

Fig. 13. Flowchart of a controlling algorithm to study the performance of a HPSC in a solar water heating system (Ayompe and Duffy, 2013)

6. Modelling novel configurations and applications of HPSCs

Despite the noticeable advantages of HPSCs, their thermal efficiency is still the major challenge of solar industry. Many researchers have tried to improve the thermal efficiency of HPSCs by proposing different methods such as modifying the structure of HPSCs, changing the manifold section geometry, and using them in multi-purpose applications (Shafieian et al., 2019a). Theoretical modelling plays an important role in technical and economic feasibility investigation of the new designs and consequently has been used widely in studies regarding the novel configurations and applications of HPSCs.

He et al. proposed a novel solar water heating system by integrating a loop HPSC with a heat pump (He et al., 2015). A mathematical model was developed aiming to simulate the operational performance of the system using heat balance equations and thermal resistance network method while momentum and energy balance equations were applied to model the heat pump. In a similar study, a loop HPSC, a heat pump, and photovoltaic panels were combined to form a solar water heating system (Zhang et al., 2013b). The heat transfer processes were simulated using the thermal resistance network method (Fig. 14). The developed model was used to determine the optimum geometrical configurations of the proposed system. Moreover, based on the theoretical results, the appropriate operational condition for various cases was recommended in the paper. To acquire further knowledge regarding the design process and equations for this type of solar heating systems, the readers are referred to (Zhang et al., 2013a).

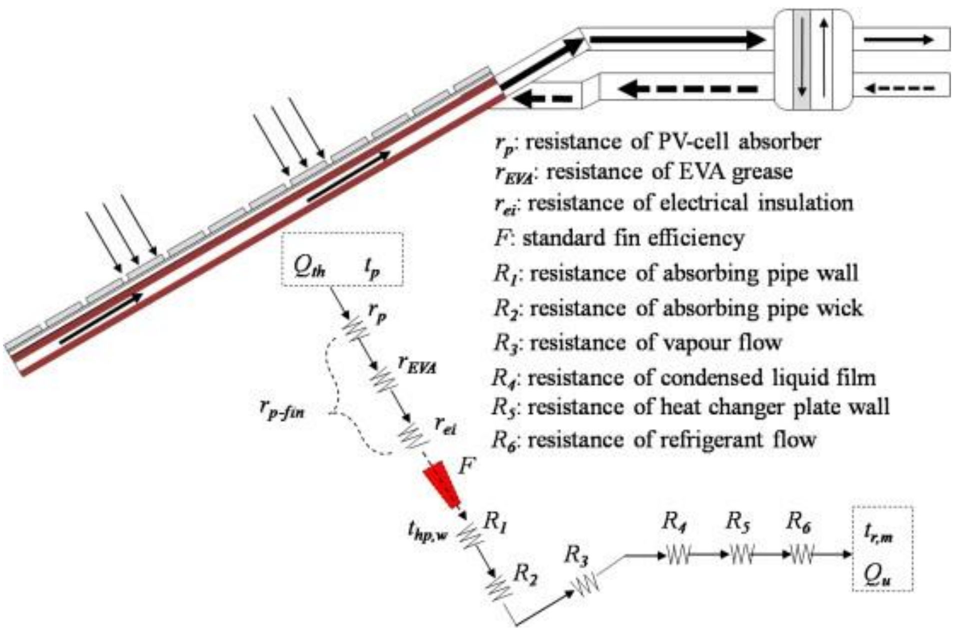

Fig. 14. Thermal resistance network used to simulate the performance of a new solar photovoltaic/heat pump system (Zhang et al., 2013b).

The technical feasibility of a novel solar water heating system consisting of a loop HPSC and a heat pump was explored theoretically by He et al. (He et al., 2014). Energy conversion and heat transfer processes of the system was modelled using the thermal resistance network method (Fig. 15). The developed model was shown to be an efficient tool to find the optimum working fluid and system configuration.

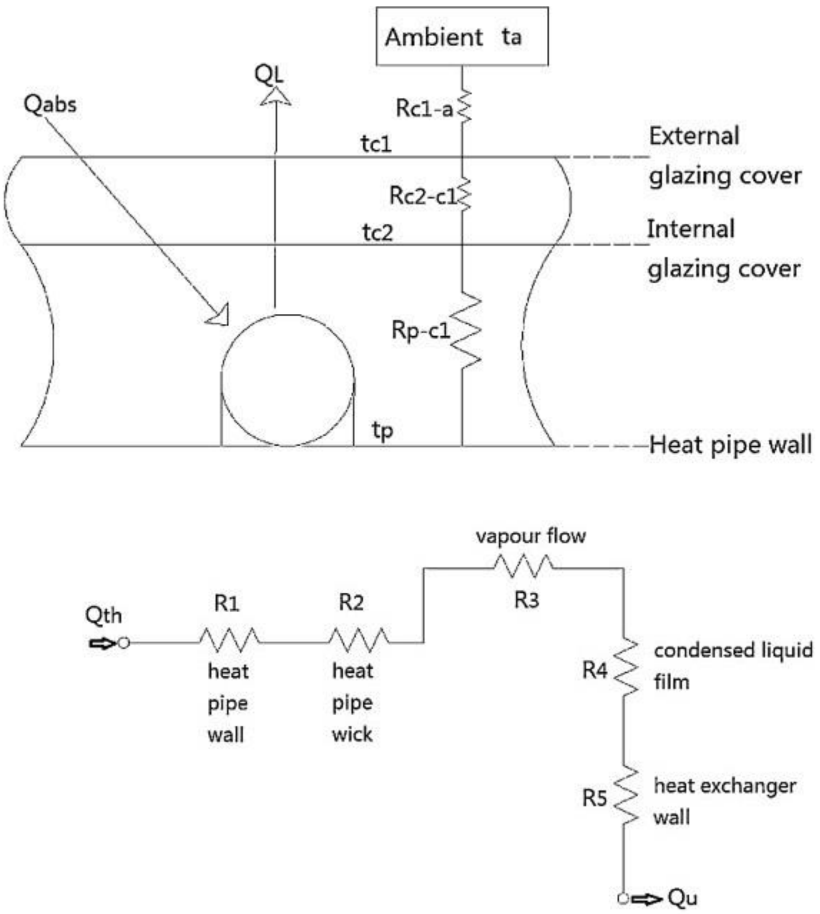

Fig. 15. Thermal resistance network of the loop HPSC (He et al., 2014).

Zhang et al. proposed a novel multipurpose solar air and water heating system by the combination of a loop HPSC, a photovoltaic panel, and a heat pump (Zhang, X. et al., 2014). The performance of the system was studied theoretically by writing the dynamic energy balance equations for all the components of the system. The equations for the solar absorber were solved by dividing the geometry into small sections (Fig. 16) and using numerical approaches while the thermal resistance network was applied to model the loop heat pipes.

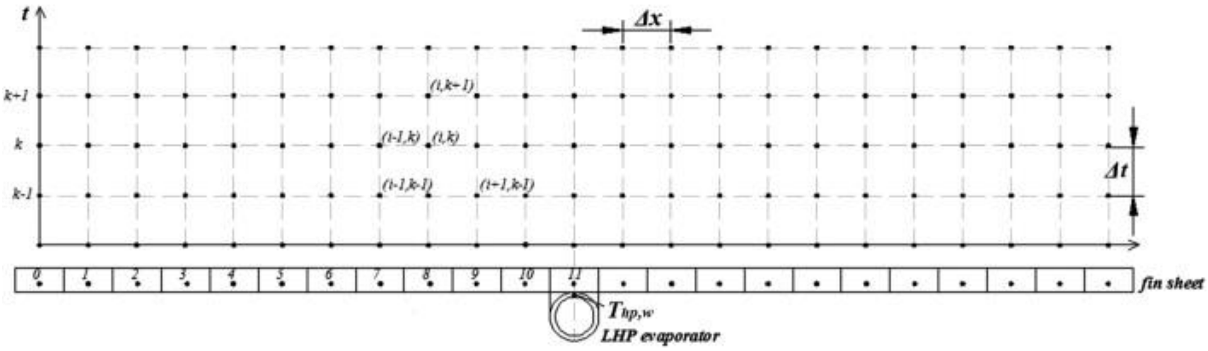

Fig. 16. Grid meshing for the absorber (Zhang, X. et al., 2014).

Albanese et al. proposed the integration of HPSCs with the walls of residential houses to form a solar wall water heating system (Albanese et al., 2012). The performance of the system was simulated for four different climates using the thermal resistance network approach (Fig. 17). The developed model was validated using data from laboratory experiments and was used to investigate the parametric sensitivity of the proposed system. Zengrui and Zhigang developed a dynamic heat transfer model to optimize the heat transfer characteristics of a heat pipe solar wall (Zengrui and Zhang, 2018). The energy saving potential of a heat pipe solar wall system was investigated theoretically during cold seasons (Tan and Zhang, 2016). In addition, several recommendations were given to improve the overall heat transfer coefficient based on the results obtained from the developed model.

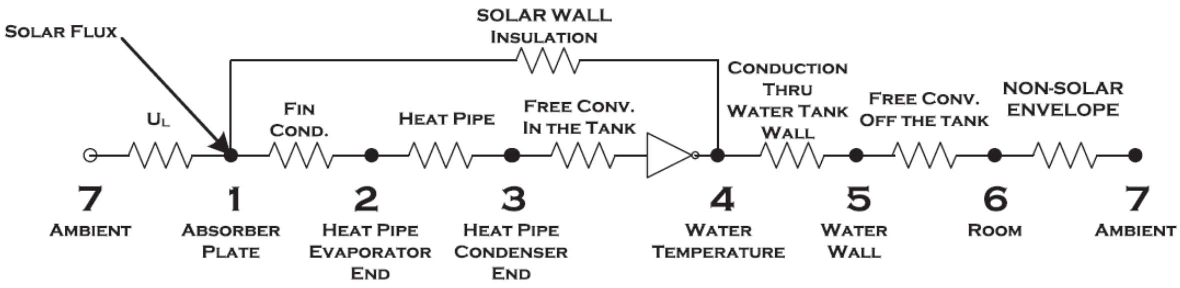

Fig. 17. Thermal resistance network of a heat pipe solar wall (Albanese et al., 2012)

Robinson and Sharp tried to reduce the unwanted thermal gains of heat pipe solar walls by modifying the design and applying new control strategies (Robinson and Sharp, 2015). Figure 18 shows the thermal resistance network which was developed for their study. In similar studies and using the same theoretical methodology, the role of a solar heat pipe wall in overall energy conservation improvement of a residential house was investigated (Robinson et al., 2013; Sun et al., 2015; Zhang, Z. et al., 2014).

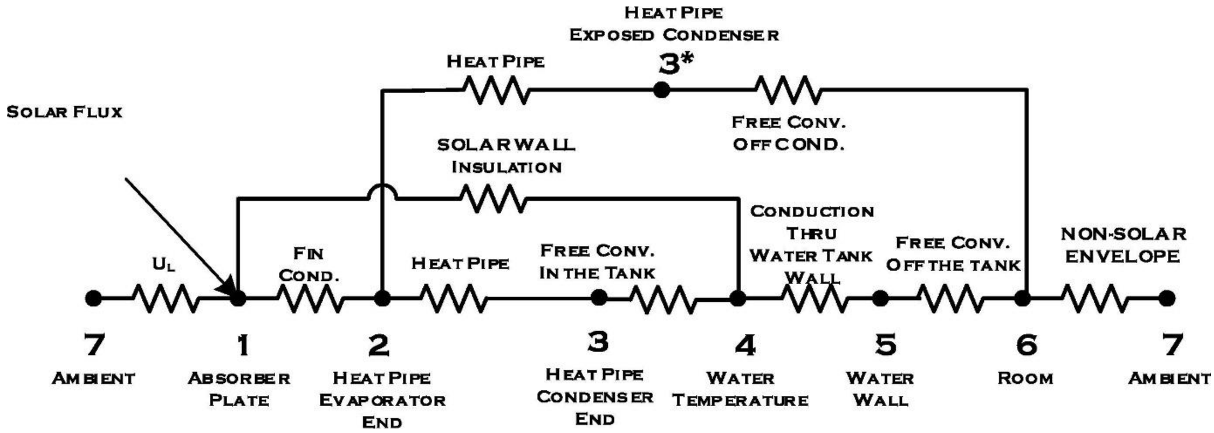

Fig. 18. Thermal resistance network of a heat pipe solar wall (Robinson and Sharp, 2015).

Chen et al. developed a dynamic model to optimize a solar heat pipe photovoltaic/thermal heat pump system (Chen et al., 2017). The combination of energy balance equations and thermal resistance network in the dynamic model were used to simulate the performance of the system. The unsteady energy equations were discretised using Newton's backward interpolation approach. In a similar study, He at al. combined HPSCs and thermoelectric modules to form a combined solar water heating and electricity generation system (He et al., 2012). The theoretical model developed based on energy balance equations aimed to relate the thermal and electrical efficiencies of the integrated system to the operational and external parameters.

In another novel configuration, Makki et al. incorporated HPSCs with photovoltaic panels and thermoelectric generators to form a multipurpose hybrid system (Fig. 19) (Makki et al., 2016). The energy balance equations in conjunction with thermal resistance network method were used to simulate the thermal processes of the HPSC. The boundary conditions and solution methods can be found in (Makki et al., 2016). Ayompe et al. investigated the capability of the Trnsys software to predict the thermal processes of a HPSC in a solar water heating system (Ayompe et al., 2011). A hot water consumption pattern was considered in this study to investigate the performance of the system under real conditions. Considering the mean absolute errors for collectors delivered heat, 7.6% difference was observed between the simulation and experimental results.

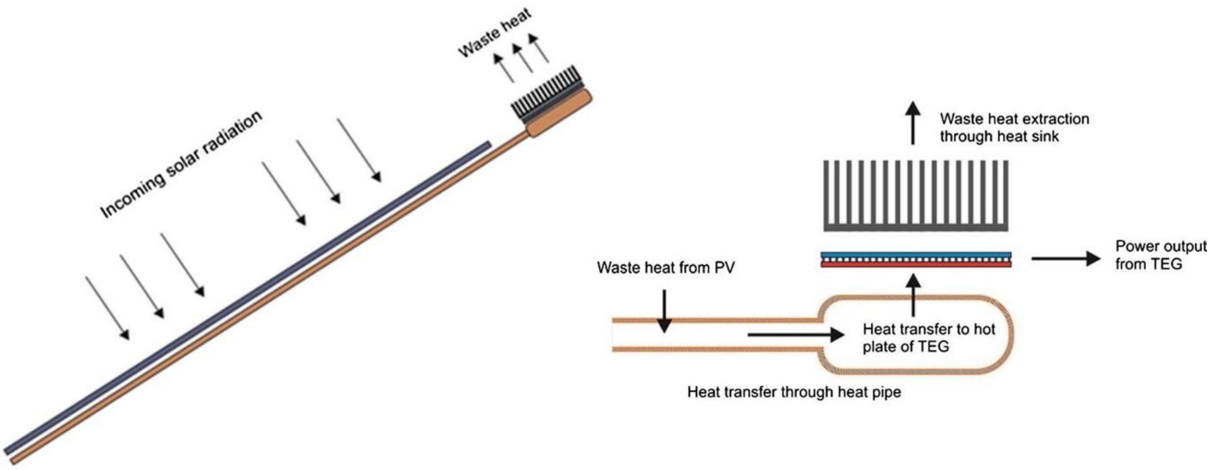

Fig. 19. HPSC in conjunction with photovoltaic panels and thermoelectric generators (Makki et al., 2016)

7. Exergy analysis

Solar systems are usually investigated by principles of energy analysis (i.e. the first law of thermodynamics) in which different energy inlets and outlets of the solar system are determined and the overall efficiency of the solar system is defined as the useful outlet energy divided by the summation of all energy inlets. In spite of its significant advantages, the weakness of energy analysis is this fact that it does not consider the degradation of the energy quality which happens in the processes (Kalogirou et al., 2016).

Exergy analysis (the second law analysis) is considered as an effective tool to evaluate the solar systems from a thermodynamic point of view. Exergy analysis can be used to determine the time and magnitude of significant energy losses helping the recognition of opportunities for thermodynamic improvement of the solar systems. Exergy analysis is useful for identification of thermodynamic imperfections resulting in efficient design of solar systems (Gunerhan and Hepbasli, 2007).

For a HPSC, the exergy balance equation can be written as (Akpinar and Koçyiğit, 2010):

$$\sum \dot{E}x_{in} - \sum \dot{E}x_{out} = \dot{E}x_{dest} \tag{23}$$

Equation (23) can be expanded to (Akpinar and Koçyiğit, 2010):

$$\sum \left(1 - \frac{T_o}{T_k}\right) Q_k - W + \sum m_{in}\varphi_{in} - \sum m_{out}\varphi_{out} = Ex_{dest} \tag{24}$$

where Q (kW) and W (kW) represent heat transfer and work rate, respectively. In abovementioned equation, φ (kJ/kg) is the physical exergy flow and can be determined by (Shafieian et al., 2019c):

$$\varphi_{in/out} = \left(h_{in/out} - h_0\right) - T_0(s_{in/out} - s_0) \tag{25}$$

where T_0 (K), s (kJ/kgK), and h (kJ/kg) are the temperature of dead state, specific entropy, and specific enthalpy, respectively.

The exergy efficiency of solar systems is defined as (Daghigh and Shafieian, 2016e):

$$\eta_{sc} = \frac{Ex_u}{Ex_{sc}} \tag{26}$$

In the abovementioned equation, Ex_u (kW) and Ex_{sc} (kW) represent useful exergy and absorbed exergy by the solar collector, respectively. These parameters can be calculated by (Gunerhan and Hepbasli, 2007):

$$Ex_u = m_w[(h - h_0) - T_0(s - s_0)] \tag{27}$$

$$Ex_u = m_w C_w[(T_o - T_i) - T_0\left(\ln\frac{T_o}{T_i}\right)] \tag{28}$$

$$Ex_{sc} = AG\left[1 + \frac{1}{3}(\frac{T_o}{T_{sr}})^4 - \frac{4}{3}(\frac{T_o}{T_{sr}})\right] \tag{29}$$

where T_{sr} is the solar radiation temperature and C_w (J/kgK) represents specific heat capacity.

For more information regarding the basic principles of exergy analysis of solar collectors, the readers are referred to (Kalogirou et al., 2016).

Despite its noticeable advantages, exergy analysis has been applied by a few researchers. Kargarsharifabad et al. performed a comprehensive theoretical study to investigate the exergy efficiency of flat plate HPSCs (Kargarsharifabad et al., 2014). In their study, the inlet exergy rate consisted of exergy rate of the solar radiation while the outlet exergy rate was the rate of exergy loss and transferred exergy. The exergy destruction was considered to be the function of the temperature difference between sun and absorber, absorber and heat pipe evaporator, heat pipe evaporator and condenser, and heat pipe condenser and solar working fluid flowing inside the manifold section of the HPSC.

Esroz used exergy analysis to theoretically investigate the influence of six various solar working fluids on the performance of HPSCs under climatic conditions of Uşak, Turkey (Ersöz, 2016). Similarly, exergy analysis was used to study the exergy destruction in HPSC used in a solar dying system (Daghigh and Shafieian, 2016c). The results showed the maximum irreversibility of 99% at the beginning of the day. In a comprehensive theoretical study, Corumlu et al. used the exergy analysis to investigate the opportunities to decrease the exergy destruction of a hybrid solar heat pipe hydrogen production system (Corumlu et al., 2018).

Shafieian et al. considered exergy efficiency as one of the main important parameters in performance analysis of HPSCs. A formula to calculate the exergy efficiency of HPSCs as a function of operational and environmental parameters was obtained from the results of this study (Shafieian et al., 2019b). In a similar study, the exergy efficiency of a HPSC was investigated and the maximum exergy efficiency of 4.5% was obtained (Daghigh and Shafieian, 2016d). The exergy efficiency of a HPSC used in a multipurpose solar water heating-drying system was also studied and the highest exergy efficiencies were obtained to be in the afternoon than in the morning (Daghigh and Shafieian, 2016a).

8. Future trends in theoretical study of HPSCs

As the application of HPSC is expanding fast in a wide range of solar systems, the theoretical studies in this field have grown noticeably. Although the previous studies and efforts have resulted in significant achievements to date, HPSCs have high potential regarding optimization and development. Moreover, there is a continuous need for expansion and improvement in the existing knowledge. Therefore, by considering the reviewed studies, the following new research directions are recommended:

- There are various methods for optimization of heat pipe solar systems including Taguchi method and genetic algorithm. These techniques are highly under-researched in this field.
- Application of software such as Trnsys for long-term performance evaluation of heat pipe solar systems is crucial and highly recommended for future research.
- Considering the improvements in computing technology and power, the application of computational fluid dynamics for simulating, analysing, and optimizing the heat pipe solar systems should be expanded.
- In spite of the effectiveness and significant role of exergy analysis in recognition of opportunities for thermodynamic improvement of HPSCs in solar systems, there is very

limited research in this field. The expansion of the exergy analysis of HPSCs in various solar systems is highly recommended.

- Most of the previous theoretical studies in the field of HPSCs have modelled these collectors as stand-alone components of the system. However, the operational conditions of the systems in which they operate affect their inlets and consequently performance. Modelling the heat pipe solar system as a whole and under different climatic conditions can be considered as an interesting research topic.
- Mathematical models are useful tools to investigate the feasibility of novel designs aiming to improve the efficiency of HPSCs. Novel designs such as application of turbulators in the manifold section of the HPSCs should be studied and expanded significantly.
- Studying the impacts of changing the structure of HPSCs on its performance is very costly. Theoretical studies can be applied to investigate the application of new physical parameters including new coating or glass materials, manifold structure, and dimensions of components.
- There is very limited research on the economic analysis of HPSCs and their economic feasibility in previous solar systems and future proposed applications.

9. Conclusions

This study comprehensively reviews the recent theoretical studies concerning HPSCs along with their advantages, disadvantages, and contribution. Section 2 explains the operational principles and structure of HPSCs. Section 3 reports the determination of solar radiation characteristics as the first major step in the simulation of solar thermal processes. Section 4 contains a comprehensive review of steady state techniques including one-dimensional energy balance and thermal resistance network methods along with their application in the literature. Section 5 covers the proposed dynamic models which describe HPSC and its thermal processes in more details compared to steady state models. Section 6 summarizes the modelling of novel configurations and applications of HPSCs aiming to improve the overall efficiency of heat pipe solar systems. Section 7 relates to the studies in which exergy analysis has been considered as a useful tool for identification of thermodynamic imperfections resulting in efficient design of solar systems. Lastly, Section 7 identifies the main challenges and existing research gaps in the field and presents recommendations for future research directions.

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