Renewable and Sustainable Energy Reviews 19 (2013) 173–190

Contents lists available at SciVerse ScienceDirect

Renewable and Sustainable Energy Reviews

journal homepage: www.elsevier.com/locate/rser

Recent advances in the solar water heating systems: A review

Ruchi Shukla, K. Sumathy*, Phillip Erickson, Jiawei Gong

Department of Mechanical Engineering, North Dakota State University, PO Box 6050, Fargo, ND 58108, USA

ARTICLE INFO

Article history:
Received 1 August 2012
Received in revised form
29 October 2012
Accepted 30 October 2012
Available online 11 December 2012

Keywords:
Solar water heating
Collector
Heat pump water heater
Thermal stratification
Refrigerant

ABSTRACT

Solar water heating (SWH) systems have a widespread usage and applications in both domestic and industrial sectors. According to Renewable Energy Policy Network data (2010), 70 million houses worldwide were reported to be using SWH systems. Solar water heating is not only environmentally friendly but requires minimal maintenance and operation cost compared to other solar energy applications. SWH systems are cost effective with an attractive payback period of 2–4 years depending on the type and size of the system. Extensive research has been performed to further improve the thermal efficiency of solar water heating. This paper presents a detailed review exclusively on the design aspects of SWH systems. The first part of the paper provides a consolidated summary on the development of various system components that includes the collector, storage tank and heat exchanger. The later part of this paper covers the alternative refrigerant technology and technological advancements in improving the performance as well as the cost effectiveness of the SWH system.

Contents

1. Introduction

Solar radiation is harnessed as an alternative energy source for numerous industrial and domestic applications. Currently, apart from space heating, air-conditioning and lightning, water heating accounts for 20% of all household energy use in the United States [1]. Domestic water heating systems have shown to offer savings of about 70–90% of total water heating costs and hence are one of the best candidates to greatly reduce household energy consumption. Over the past 30 years, solar water heating has become more sustainable, efficient, and economically feasible.

For instance, solar water heating (SWH) system usage grew from a worldwide capacity of 160 GW_{th} at the start of 2010 to 185 GW_{th} by beginning of 2011. Though China leads in SWH market (118 GW_{th}), significant expansion has also occurred in the European Union, Japan, India, and Brazil. In the United States, solar water heating growth is relatively low in comparison to its water heating needs [2]. Many possibilities for significant growth

* Corresponding author. Tel.: +1 701 231 7139; fax: +1 701 231 8913.
E-mail address: sumathy.krishnan@ndsu.edu (K. Sumathy).

1364-0321/$ - see front matter © 2012 Published by Elsevier Ltd.
http://dx.doi.org/10.1016/j.rser.2012.10.048

Nomenclature

PV	photovoltaic
F'	efficiency factor
F_R	heat removal factor
U_L	heat loss coefficient (W/m^2 K)
T_a	ambient temperature (K)
I	solar radiation
K	thermal conductivity (W/m K)
H	heat transfer coefficient (W/m^2 K)
G_r	solar irradiance (W/m^2)
Nu	nusselt number

Greek symbols

η	efficiency
τ	transparency
ε	exergy
δ	plate thickness
θ_c	acceptance angle
φ	deviation angle

Abbreviations

COP	coefficient of performance
SWH	solar water heating/heater
DWH	domestic water heating
ICSSWH	integrated collector storage solar waterheating
ICS	integrated collector storage
FDM	finite difference method
DX-SAHP	direct expansion solar assisted heat pump
SAHPWH	solar assisted heat pump water heater
ISAHP	integral-type solar assisted heat pump
HPSAHP	heat pipe solar assisted heat pump
SAS-HPWH	solar-air source heat pump water heater
DSWH	domestic solar water heating/heater
CEOHP	closed-end oscillating heat pipe
SSF	solar saving factor
HTF	heat-transfer fluid
RT	recto-trapezoidal
EPDM	ethylene propylene diene monomer
TPNR	thermoplastic natural rubber
FPC	flat-plate collector
ETC	evacuated tube collector
CPC	compound parabolic concentrators
HX	heat exchanger
PIV	particle image velocimetry
CFD	computational fluid dynamics
GWP	global warming potential
ODP	ozone depletion potential
CFC	chloroflurocarbon
HCFC	hydochloroflurocarbon

exist worldwide in the domestic water heating market, and even more for industrial purposes. As of 2010, there were only 100 such industrial projects in operation [2].

Several SWH systems have been designed and developed to meet the requirement of different applications and local climatic conditions. This paper will discuss the history of design implementations and changes in SWH systems over the past 30 years. Since the 1960s, SWH technology has matured and many different designs have attained commercialization; yet there exists opportunities for further improvements in efficiency and reliability [3]. Most of these enhancements have focused on the problems associated with low ambient temperatures and the transitions in solar radiation ratios with respect to the season. Other design problems include expensive components, overheating, corrosion, and insufficient protection against freezing. Several investigations have reported improvements to address these issues, such as employing different collectors, storage tanks, and working fluids to suit regional/specific geographical conditions. From '80s to the current date, there has been an increasing interest to enhance thermal performance of SWH systems by means of improving the absorber plate characteristics, enhancing the thermal stratification of the storage tank, optimizing the design parameters, and extending the heat-transfer area [4].

2. Solar water heating systems

Solar water heating systems harness solar and, in some cases, are supplemented by ambient energy to heat water (Fig. 1). The first commercial SWH, named "Climax," was patented in the US by Kemp [5]. In the early 1900s, several researchers focused their attention to improving the design of the SWH systems to make them durable and efficient. SWH systems were commercialized on a wider scale in the early 1960s. In the sections below, a variety of SWH systems are reviewed and classified in terms of circulation methods and applications, with a discussion on the designs and modifications in recent years. These systems can be broadly categorized as passive solar water heating systems and active solar water heating systems.

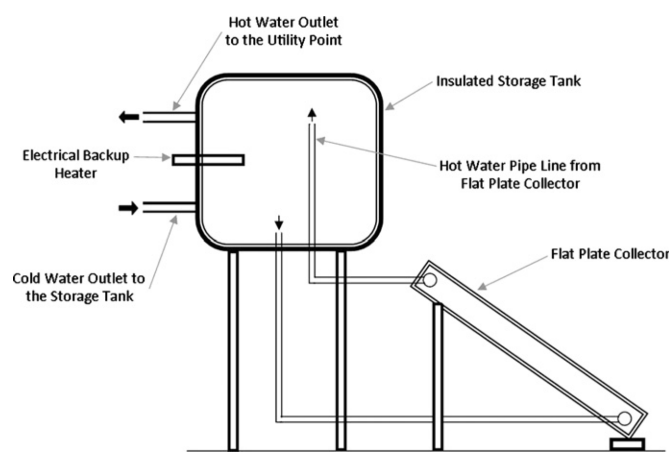

Fig. 1. Line diagram of a typical thermosyphon SWH [5].

2.1. Passive solar water heating systems

Passive solar water heating systems depend on heat driven convection to circulate water or heating fluid in the system. These passive systems can be divided into the two main categories, the integrated collector storage and the thermosyphon SWH systems.

Integrated collector storage solar water heaters (ICSSWH) use a tank that acts as both a storage and solar collector apparatus. This system is also popularly known as a batch SWH system. One of the simplest designs of ICSSWH system is in which a simple tank is enclosed with a glass cover such that it performs as a collector as well. However, one of the main drawbacks of this design is the heat loss, which is more pronounced at night. To reduce heat losses, effective measures such as selective absorber surface coatings, insulating materials, and additional glazed glass covers can be appropriately utilized.

The first detailed study on closed and exposed single tank systems was carried out by Brooks at the University of California Agricultural Experimental station in the US in 1936 [6]. While the investigations showed promising potential, further research was eclipsed by the increased use of natural gas and oil in the US, but the research in solar water heating gained momentum in Japan.

The Japanese commercialized a design in early 1950s and marketed a 'Closed-pipe' ICSSWH system [7]. This concept was further improved by introducing cylindrical vessels (the collector and storage tank combined), which are still used in many commercial designs today. However, it was observed that rectangular vessels could also function and perform similar to cylindrical vessels [8–10]. Other than cylindrical geometry, triangular-designed ICSSWH systems were also introduced, and were found to improve heat transfer rates because of increased natural convection currents [11,12]. A novel ICS vessel design based on a pyramid shape was presented by Abdel-Rehin [13], which performed successfully in the hot Egyptian climate. Along with the attempts on various geometrical designs, numerous studies have been conducted to enhance the thermal performance of the ICSSWH systems by incorporating transparent insulation material. Fibre glass, organic-based transparent foams, inorganic glass foams, and honey comb structures have been used as insulating materials in several designs [13–15].

To further enhance the collector's efficiency, researchers introduced reflectors in order to maximize the solar radiation incident on the absorber surface [16–18]. Reflecting concentrator designs can be either flat or curved, line axis, symmetrical or asymmetrical depending on the desired concentrations. Davis et al. [16] developed a symmetric cusp reflector ICSSWH system designed particularly to suit Colorado's cold weather conditions, and found that the collector efficiency can reach as high as 72%. Stickney and Nagy [17] designed an inverted ICSSWH system which consisted of a slender glass lined tank enclosed in insulation with a double glazed aperture facing downward to collect the reflected solar radiation from the parabolic reflector. This accomplished increased heat retention during night and affected the increased daytime temperatures. Yet another design focusing on insulation was introduced by Schmidt et al. [18], comprising of a pressure-resistant single tube absorber which was integrated to an involute reflector that enclosed a transparent insulation material. This type of ICS system could achieve an overall annual efficiency of 28% and an annual solar saving factor (SSF) of 58%.

To further improve the system efficiency, baffles, which help in guiding the direction of the flowing fluid, were utilized in the storage tank [19]. Taylor et al. [20] implemented a baffle arrangement to their rectangular-shaped SWH system. The baffles were provided in such a way that they separated the vessel into an inner storage volume and outer collecting volume. The first numerical and experimental studies on baffle plates included in triangular and rectangular ICSSWH systems were carried out by Sokolov and Vaxman [21]. Similar to this work, Kaushik et al. [22] utilized a baffle plate within a triangular ICS system (Fig. 2). The inclusion of the plate had a significant effect on the system's performance, especially during non-collection periods. The results showed that the thickness and material of the baffle had little impact on the system performance. Based on the above listed developments, Mohamad et al. [23] introduced a simple thermal diode (Fig. 3) in the ICS system, to avoid the reverse circulation during night period.

In the late 19th century, batch heaters were displaced due to their pertinent heat loss issues by thermosyphon systems. This design is not only popular in developing countries that experience power issues, but is also utilized in other countries (e.g.,: Cyprus) which have conducive climatic conditions. A simple open-loop thermosyphon system relies on the principle of density differences to effect circulation [24]. That is, the system uses natural convection flow to transport the heat transfer fluid or water from the collector to the storage tank.

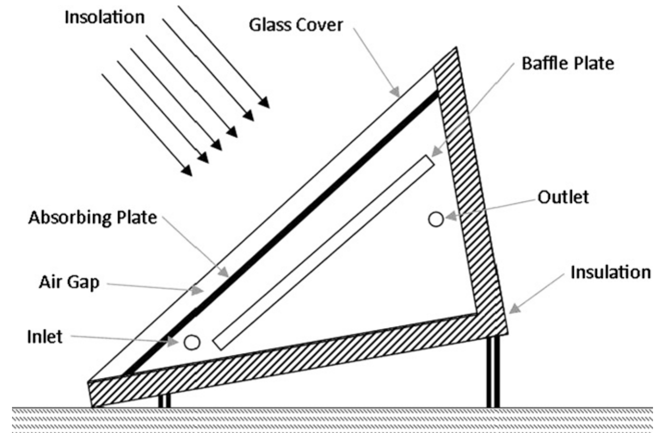

Fig. 2. Triangular built-in-storage water heater [22].

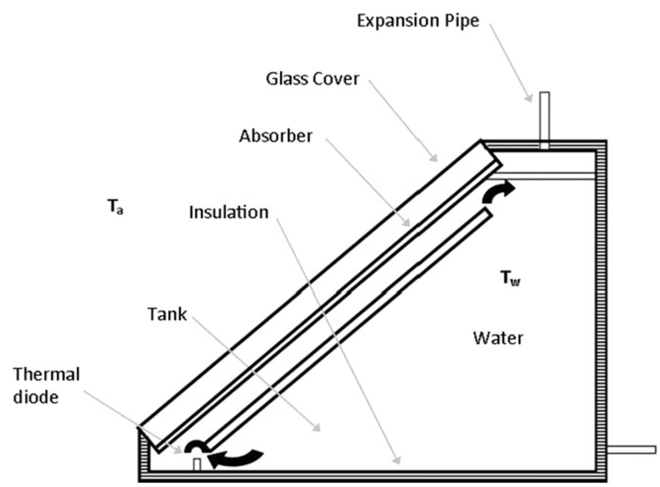

Fig. 3. Integrated solar collector/Storage tank [23].

Numerous analytical and experimental studies have been intensively carried out on the thermosyphon SWH system to analyze its performance [25–35]. Fundamental models to estimate the thermal performance of the natural circulation for SWH systems were developed to predict the heat gain in the thermosyphon-driven SWH, when subjected to no drain-off conditions. The models were validated with experimental data [25]. Yet another mathematical approach utilizing "finite-difference method" (FDM) was carried out by Ong [26] to evaluate the thermal performance of a natural-recirculation SWH system. However, the measured experimental data were contradictory to the theoretical predictions. Hence, the model was modified by taking into account the experimental conditions and used the FDM to predict the gain in the temperature at a given time step. Similarly, Sodha and Tiwari [27] used explicit expression to analyze the performance of a SWH system with natural thermosyphon circulation between the storage tank and the collector. The results confirmed that the SWH system's performance can be predicted accurately by using simple explicit equations.

To further analyze the effect of flow circulation in a thermosyphon system, Grossman et al. [28] carried out a study on a parallel plate absorber type SWH system which was subjected to forced or natural circulation of water between the storage tank and the absorber plate. This design offered the maximum area of contact between the fluid and the collecting surface exposed to the Sun and may therefore serve as a reference for comparison with other less efficient geometries. Huang [29] developed a more general theory for a parallel plate absorber type system with natural circulation of water between the plate and the tank by representing solar radiation as a sine function of time.

Other than investigations on conventional solar flat-plate collector, studies have been carried out on the use of heat-pipes in the evacuated tube collector to affect SWH systems. Ng et al. [30] developed and experimentally confirmed a theoretical model to predict the performance of an evacuated-tube heat-pipe solar collector. Performance parameters such as the collector efficiency and useful energy gain were analyzed. For a range of expected coolant temperatures, the heat loss coefficient and efficiency of the collector could be sufficiently represented by a simple linear expression. Redpath et al. [31] carried out an experimental study on a heat-pipe evacuated tube SWH system. A two dimensional particle imaging velocimetry (PIV) was used to visualize the thermosyphon fluid regime. It was concluded from the study that the overall cost of the heat-pipe evacuated tube SWH system can be reduced by using thermosyphonic flow instead of forced circulation.

One of the widely-used forms of evacuated tube collector for thermosyphon systems is the water-in-glass tube design because of its high thermal efficiency. Budihardjo and Morrison [32] carried out thorough simulation studies on the said design to analyze the optical and heat loss characteristics of a single-ended tube. The study concluded that the system's performance was less dependent on tank size compared to the commonly used flat-plate collector system; however, the system has a limitation as it can be used only in low ambient conditions due to its limited tolerance for high pressure.

A flat-plate collector integrated with a mantle heat exchanger was attempted by Huang et al. [33], to show its applicability of inherent freeze protection. It was observed that the mean daily efficiency of the SWH system using a mantle heat exchanger was up to 50%. While this is lower than the efficiency of a thermosyphon flat-plate system without a heat exchanger, it is higher than the all-glass evacuated tube SWH system.

To further improve the collector design, Jaisanker et al. [34] in 2009 used a collector with twisted tape inserts to effect higher heat transfer rates. Both the heat transfer and the friction factor characteristics were analyzed based on the experimental data to study the effect of a spacer being incorporated along with the helical insert. It was found that the inclusion of a spacer decreased the heat transfer (Nusselt number) and the frictional factor by about 11% and 19%, respectively [35].

2.2. Active solar water heating systems

Unlike passive systems, active systems use one or more pumps to circulate the working fluid in the system. Active systems can be categorized into direct circulation and indirect water heating systems. In the direct or open-loop systems, water from the storage tank is directly circulated to the collector to be heated by solar energy, whereas in the indirect active system the heat transfer fluid is circulated through the collector and rejects heat through a heat exchanger to the water in the storage tank [3].

Direct systems are simple in operation, but they are sensitive to freezing conditions and could provide hot water of moderate temperature (~50–60 °C) [24]. To overcome the freezing issues, some design modifications have been adopted. One such modification is to operate the direct circulation system in drain-back mode, in which a differential controller-integrated pump is used to circulate water from the storage tank to the solar collectors [3]. While there are numerous research papers related to the direct mode of SWH systems, some of the more recent investigations are discussed here.

More recently, vacuum tube collectors have been used for domestic water heating purposes and it has been observed that the performance of vacuum tube collectors are much higher than flat-plate collector because of low convection heat losses from the absorber. A heat transfer model to evaluate the performance of all-glass vacuum tube collectors incorporated in a direct circulation system was developed by Li et al. [36]. This simplified model takes into account of natural circulation in single glass tube as well as forced flow circulation in the manifold header. Flow equations were obtained by analyzing the friction losses and buoyancy forces inside the tube. A good agreement was observed between the predicted and computed collector outlet temperatures, and the deviation was within 5%.

Apart from the above discussed collector types, a relatively new V-trough SWH system is also now commercially available. Chong et al. [37] researched a cost-effective new V-trough SWH system that employed direct circulation. It was observed that the thermal performance of SWH could substantially improve by integrating solar absorber with a V-shaped trough reflector. The system was tested with and without glazing and different insulating materials. It was reported that the prototype could achieve an optical efficiency of 71% with a maximum outlet water temperature of 82 °C and 67 °C with and without insulation, respectively.

In general, direct heating systems are commonly employed in regions with ample sunshine and experience moderate ambient conditions. For regions with less sunshine hours and low ambient conditions, indirect water heating systems are employed. These systems are reliable in operation and ensure effective freezing protection [24]. Heat transfer fluid (HTF), such as ethylene glycol and other refrigerants is circulated between the collector and the storage tank through the heat exchanger. Indirect SWH systems operate on a heat pumps (HP) mode, to supplement the solar energy gain in the collector. These HP based SWH systems have shown several design improvements during last 20 years. The hot water is generated utilizing waste heat or other low temperature sources, in which an exclusive working fluid is circulated in the collector and the heat gain is rejected through a heat exchanger to the storage-water. The solar assisted heat pump is a commonly used type; however one of the challenges is that its performance is very low when the ambient temperature is low. There exists numerous studies [38–43] related to the indirect mode of SWH systems to improve its performance, and some of the recent research is discussed.

A variable capacity direct expansion solar-assisted HP system (DX-SAHPS) for water heating purposes was tested by Chaturvedi et al. [38]. A bare solar collector acting as an evaporator was used for the heat pump system. The system was tested for the widely varying ambient conditions, and accordingly, the compressor speed was varied through a variable frequency drive. The observational results showed that the coefficient of performance (COP) of the system can be enhanced extensively by lowering the speed of the compressor when the ambient temperatures were higher. Hence, such systems perform better in summer compared to winter.

Yet another design of DX-SAHPS was proposed by Kuang et al. [39], in which a 2 m^2 bare flat-plate collector acted as a source as well as an evaporator for the refrigerant (Fig. 4). From the simulation model it was concluded that the monthly average COP varied between 4 and 6, and the collector efficiency was about 40–60% [39]. Similar to this work, Li et al. [40] presented another experimental study on the DX-SAHP. The results showed

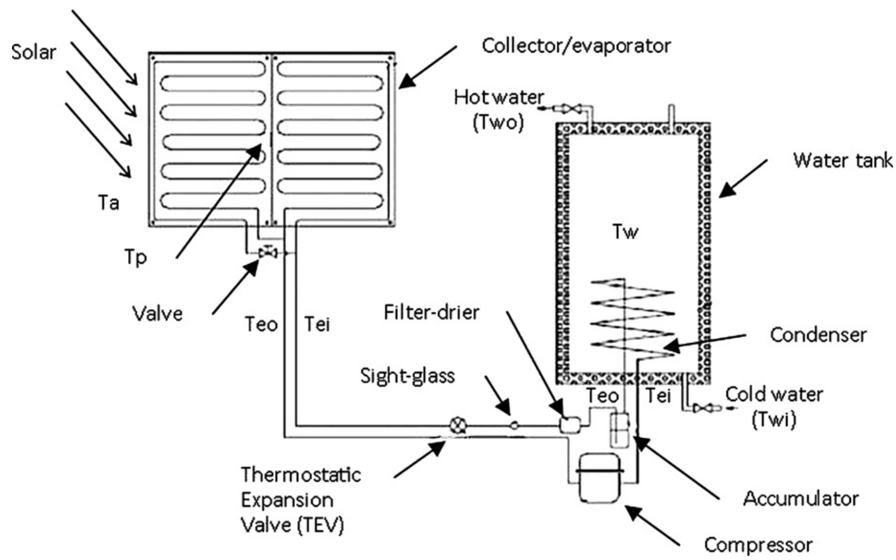

Fig. 4. DX-SAHP water heater [39].

that the proposed system can attain a COP of 6.61 on a clear sunny day (T_a=17.1 °C, I=955 W/m^2). Even during cloudy and rainy nights, the system could perform moderately with a COP of 3.11.

An integral-type solar assisted heat pump water heating system (ISAHP) was studied by Chyng et al. [41]. Experiments were conducted for over a year to record the annual performance. Simulations were performed based on the assumption that all the components are at steady-state except for the storage tank. The model agreed well with the experimental data and the predicted results were within 10% of the measured data. It was observed that the daily total COP for the system was about 1.7–2.5.

Apart from heat pumps, heat pipes were also introduced to further enhance the system's performance. A heat-pipe water heater was fabricated and tested/modeled by Huang et al. [42]. The performance of the combined solar heat pipe collector and conventional HP were examined to calculate the overall COP of the system. When solar radiation was low, the system operated in HP mode. However, during clear sunny days, the heat-pipe mode operated independently of electrical energy input, for higher thermal efficiency. The results showed that the COP of the hybrid-mode of operation could attain as high as 3.32, and as such its performance was higher by about 28.7% compared to the HP mode of operation.

Guoying et al. [43], worked on an air-source assisted solar HP system to effect water heating. This system employed a specially designed flat-plate collector provided with spiral-finned tubes (which also act as an evaporator), to harness energy from ambient air as well as solar radiation. From the simulated results it was observed that a SAS-HPWH could efficiently heat the water to 55 °C, under varying weather conditions.

3. Component designs

The main components of SWH system, such as the collector, heat exchanger, and storage tank, have been improved over the years. Also, different working fluids have been attempted by several researchers to improve the system efficiency as well as make the system functional under different operating conditions. Significant studies on the design modifications are reported below in the respective subsections.

3.1. Solar collectors

A solar collector is a heat exchanger that converts solar energy to useful heat which is transferred to the fluid flowing through the collector. It was first developed by Hottle and Whiller [44]. The design parameters such as efficiency factor (F') and heat removal factor (F_R) were developed, which significantly reduced the empiricism associated in the design of a solar collector [45]. The efficiency of a SWH system depends mainly on the effectiveness of the flat-plate collector, and thus most research has been concentrated in improving the performance of the collector; specifically the absorber plate design and glazing material.

3.1.1. Flat-plate collectors

A flat-plate collector (FPC) is the heart of a SWH system and it is commonly used for harvesting solar thermal energy at low ambient temperatures. It consists of: a selectively coated a flat-plate absorber plate, a transparent cover to reduce top heat-losses from the absorber plate, heat-transport fluid (HTF) to remove heat from the absorber plate, tubes/passages for the flow of the HTF, a heat insulating support to reduce heat loss from the collector, and a protective casing to ensure the components are free from dust and moisture [44].

Several researchers have worked on the design and development of FPC. Matrawy and Farkas reported that the configuration of the collector plays an important role in dictating its thermal performance [46]. The parallel-tube collector design is one of the most commonly used configurations, in which tubes (risers) are integrated to the absorber plate, forming an integral part of the plate structure. Hottle and Whiller [45] were the first to analyze the thermal performance of parallel tube collectors. The thermal analysis comprised of a simple two-step procedure where the heat conduction from the plate to the flow duct was evaluated based on the fin efficiency factor and the transportation of heat to the fluid in the duct was predicted based on the heat removal factor [47,48]. Some of the disadvantages of this design were: non-uniform temperature distribution over the absorbed plate surface, unequal distribution of the working fluid through the collector risers, and the increase in the collector's heat loss caused by the higher temperature of the absorber plate in low flow rate conditions. To overcome these issues a serpentine tube collector was introduced. This design was typically used to compensate low flow rate conditions; the design enables the total mass flow rate to pass through the tube, increasing the heat transfer coefficient [46]. In an attempt to further improve the thermal performance of a parallel-tube collector, two parallel tube collectors were connected in parallel and this system was compared to a serpentine tube collector having an equivalent exposed area [46]. It was found that the efficiency was about 6% more than a serpentine tube collector and 10% more than a single parallel tube collector, under similar ambient conditions.

Other geometric designs [49,50] include a fin-and-tube collector, in which fluid is circulated through a corrugated channel which is in direct contact with the absorbing surface. It was found that the serpentine geometry showed better thermal performance than the parallel one.

In 2006, yet another new design [51] was proposed by Rittidech and Wannapakne (Fig. 5). A flat-plate solar collector operating in conjugation with a closed-end oscillating heat pipe (CEOHP) was investigated. The collector system using a CEOHP offers a reasonably efficient and cost-effective alternative method compared to the conventional solar collector system that utilizes heat pipe. This cost effective collector system could attain a 62% efficiency which is comparable to the performance of a heat pipe.

The core component of the flat-plate collector is the absorber plate, and its thermal performance depends on the material properties as well as on the design parameters. Numerous absorbing plate designs have been proposed in recent years [52–57]. Among different absorber plate profiles, parabolic shape is one of the most efficient, since heat output per unit volume is higher than other geometries [52]. Holland and Stedman [53] introduced a rectangular profile with a step-change in local thickness. A thermal analysis was carried out to optimize the absorber plate fin, and reported that to maximize energy transfer the fin area should be increased for the given volume of the fin (m^2/m^3). Instead of a rectangular fin design, Kundu [54] introduced a new recto-trapezoidal (RT) shaped fin. Simulation results confirmed that the RT profile absorber plate fin was superior to other profiles such as rectangular, triangular, or trapezoidal, with respect to heat transfer rate per unit volume.

Metal absorbers are more prone to corrosion; hence, to maintain performance, an absorber must be resistant to both internal and external corrosion. The use of polymer-based absorbers has increased because they are non-corrosive, light weight, cheaper, and easier to fabricate. Recent advances in polymer technology have resulted in the development of suitable materials which can withstand long exposure to sunlight, such as ethylene propylene diene monomer (EPDM) [55]. Sopian et al. [56] experimentally studied the thermal performance of flat-plate solar collector with a thermoplastic natural rubber tubing (TPNR) for the absorber plate. A commercial blend of TPNR was used for the parallel-type absorber plate. The experimental results

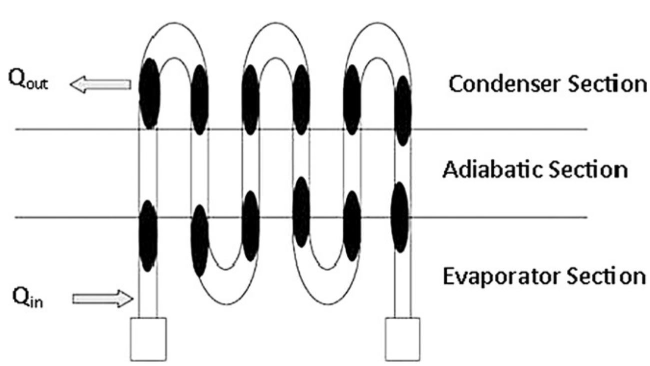

Fig. 5. Closed-end oscillating heat pipe [51].

confirmed that this natural rubber can be used as an alternative to other materials.

In Europe, new polymer flat-plate collectors have been introduced to replace metal collectors. Polymer collectors are flexible and tolerant to expansion during freezing; therefore they polymer tubes tend to contain plain water without the need for antifreeze. As such these collectors can be directly connected to the storage tank, instead of using heat exchangers that lower the thermal efficiency of the system. Use of polymeric components has also reduced the fabrication cost, when compared to metal components [44]. Liu et al. [57] presented that the cost of a conventionally-designed nylon solar absorber was roughly 80% of the cost of a similar copper unit.

Though polymer collectors are corrosion resistant, they do have several limitations such as low thermal conductivity and can be used only for moderate temperature applications. To overcome these limitations, to a certain extent, an extruded parallel plate absorber design has been proposed [58]. It is one of the most promising designs, which comprises of a pair of parallel plates, with water flowing between the two plates. The gap between the plates can be varied from a few millimeters to a few centimeters, according to the design parameters. Tsilingiris [58] evaluated the performance of a similar parallel polymer plate collector. This design facilitates the absorption of radiation at the back polymer plate in the flowing water stream, which helps to eliminate the adverse effects of the top absorbing plate conductance and results in improved collector efficiency. Results had indicated that the collector efficiency factor for a back-absorbing plate absorber was 20% higher and the collector loss coefficient was reduced by about 15% compared to a top-absorbing plate collector.

Collectors' efficiency can be further improved by incorporating appropriate transparent insulation materials. Glazing is generally used to reduce the top heat losses from the panel to the environment by convection and radiation and to ensure higher thermal performance [59]. Whereas unglazed flat-plate collectors are not as common as solar collectors with glazing [48]. One of the widely-used glazing materials in solar collectors is glass, which is low in cost and has a solar transmittance up to 90% [24]. By applying an anti-reflected coating, the solar transmittance of glass can be increased, resulting in improved efficiency [60]. Use of multiple glass layers can also help in reducing heat loss, creating air gaps that act as insulation [61]. Another type of glazing material is plastic, available as thin films or sheets, having shortwave and long wave transmittance as high as 0.40 [24]. Literature shows that in recent years, polymeric materials are also used as upper glazing in solar collectors [62].

Common designs of transparent insulation materials include the rectangular, circular, and hexagonal geometry, which consist of a honeycomb structure that causes a reduction in free convective losses [63]. Abdullah et al. [63] investigated the solar collector performance when equipped with different honeycomb-pattern arrangements. Results had confirmed that the top heat losses from the collector were considerably reduced, though the optical efficiency was marginally reduced. The heat removal factor and heat loss coefficient (U_L) were reduced to 56% for the double honeycomb unit, compared to a conventional collector (without a honeycomb structure).

The material properties of the absorber, such as absorptance and emmittance, also play an important role in dictating the heat output of the SWH system. Hellstrom et al. [64] investigated a combination of a Teflon honeycomb collector being treated with anti-reflection coating. A combined increase in absorptance from 0.95 to 0.97 and a decrease in emittance from 0.10 to 0.05 resulted in an increase in the annual performance of 6.7% at 50 °C operating temperature. It was found that when a Teflon cover was provided along with the regular glazing, it resulted in an increase in efficiency of about 6%. A further increase in efficiency (12.1%) was observed when a honeycomb structure was incorporated. A combination of absorber improvements together with a Teflon honeycomb and an anti-reflection-treated glazing doubled the increase to about 24%. Also, the study recommends a plane glass instead of a structured glass, since the structured glass has lower transmittance which affects the optical efficiency of the system.

Tripanagnostopoulos et al. [48] investigated three flat-plate collectors that used blue, black and red-brown absorbers with and without glazing. It was found that unglazed collectors with colored absorbers usually achieved low efficiencies. Both experimental and theoretical results confirmed that collectors with colored absorbers had efficiencies close to the collectors with a black absorber if dark tone color paints are used. Also it was observed that the solar collectors with or without glazing (using a colored absorber) when combined with booster reflectors could affect an increase in the thermal energy output. Similar to the above mentioned study, Wazwaz et al. [65] conducted an experimental work to determine the solar thermal performance of a nickel-pigmented aluminum-oxide selective-coated absorber. Results had indicated that the maximum collective flux of the selective absorbers was about 590–699 W/m^2. It was also suggested that commercial polyethylene can be used to protect and further insulate a glazed flat-plate solar collector.

Different types of reflectors were also incorporated in the flat-plate collector design to improve the collector's efficiency. Tripanagnostopoulos et al. [48] experimentally investigated the effect of flat booster reflectors on the thermal performance of the collector. It was found that reflectors improved the thermal energy output of the collector. Another improved design was introduced by RÖNnelid and Karlsson [66]. Instead of flat booster reflectors, corrugated booster reflectors were used for increasing the annual irradiation onto the collector plane.

Tanaka [67] theoretically analyzed the solar thermal collector incorporated with a flat-plate top reflector. The results predicted that the ratio of the reflector and collector length had a marginal effect on the absorption of radiation by the collector. Compared to the conventional solar thermal collector, by using the flat-plate reflector, daily solar radiation absorbed on the absorber plate increased to about 33% for a reflector to collector length ratio of 2.0. Recently (2011), Kostić and Pavlović [68] conducted studies on thermal collectors with and without flat-plate solar radiation reflectors. The objective was to determine the reflector plate's optimal position for a given collector's position. Experimental results had concluded that in December (winter) the optimal angle position of the bottom reflector was the lowest (5°) and in June (summer) the highest (38°). It was also observed that the thermal collectors with reflectors could affect average energy gain by about 40% in the summer period.

3.1.2. Evacuated tube collectors

Evacuated tube collectors (ETC) have been commercially available for more than 20 years. Though they have better performance in producing high temperatures when compared to flat-plate collectors, they are not competitive because of high initial costs. ETC consists of: evacuated tubes (glass-glass seal) to minimize heat losses, copper heat pipes for rapid heat transfer, and aluminum casing to provide durability and structural integrity to the system [69]. ETC minimizes the heat losses due to convection and radiation [70].

At present, glass ETC have become a key component in solar thermal utilization, as they can have high heat extraction efficiencies and considerably less heat loss issues compared to flat-plate solar collectors. One of the important design factors for the glass ETC is the shape of absorber tube [71]. For instance, Perez et al. [72] confirmed that the glass ETC with a semi-cylindrical shaped absorber tube could absorb about 16% more energy than ETC with a flat plate shaped absorber tube.

Recently, Kim and Seo [71] introduced a few efficient designs of the absorber tube and investigated the performance of the four different shapes of absorber tubes. The absorber tubes investigated were: finned tube, an U-tube welded inside a circular fin, an U-tube welded on a copper plate, and an U-tube welded inside a rectangular duct. The results indicated that U-tube welded inside a circular fin gave the best performance among the four different collectors.

Shah and Furbo [73] introduced yet another absorber tube design. A parallel-connected evacuated double glass tube collector design using a tubular absorber was found to absorb incident radiation from all directions. The experimental and numerical studies on the tubular collector design confirmed that the design is very promising especially in higher latitudes. However, one of the major problems of all-glass ETC designs is the heat extraction through the long, narrow, single-ended absorber tube. Over the years, different designs of ETC have been reported in several research papers [74,75]. Two basic ETC designs are the simple fluid-in-glass and fluid-in-metal designs. The fluid-in-metal design can be further classified into heat pipe ETC and U-tube ETC. Heat pipe ETC and U-tube glass ETC are the two most widely-used solar collectors in domestic water heating. Research on the different designs of ETC has focused on the enhancement of the solar collector's performance. Details on some of the recent studies on heat pipe ETC are discussed below.

In 2009, Rittidech et al. [76] constructed a simple closed-loop, oscillating, heat-pipe ETC to overcome issues concerning corrosion and icing (during winter). The system attained a comparable efficiency of about 76%. For a heat-pipe, a vacuum environment is essential to attain high thermal frequency [75]. However, there exist several issues in providing perfect vacuum conditions since the non-condensable gases, released during the operation, remain in the heat-pipe and affect its performance [76]. Hence, there exists a limitation for the usage of heat-pipe ETC systems.

Apart from ETC heat-pipes, U-tube glass ETC are also commonly used collectors. In contrast to the all-glass evacuated tube and the heat-pipe ETC, the U-tube ETC has a simple structure and is also tolerant to high pressure. Diaz [77] introduced a novel design of a mini-channel-based ETC which was found to improve thermal efficiency at the expense of power consumption. The collector design consisted of a U-shaped flat-plate selectively coated absorber provided with fins to enhance heat transfer area between tube walls and the working fluid. The study showed that the mini-channel tube collector attained better efficiency compared to the standard U-tube collector. Similar to Diaz's work, Ma et al. [78] investigated the thermal performance of a double-layered glass ETC. The heat efficiency factor and heat loss coefficient were examined using a one-dimensional analytical solution. The study reported the influence of an air layer on the heat efficiency and an increase of 10% was noticed when the air layer thickness was increased such that its resistivity was at 40 W/m K. It was also noticed that when the synthetical conductance was increased from 5 to 40 W/m K, solar collector efficiency increased by 10% and the fluid outlet temperature increased by 16%. Ma et al. [75] extended their work to study the effectiveness of filled-type ETC of a U-tube design (Fig. 6). Different filler materials like air and graphite were attempted. Results had shown that the thermal performance of the filled-type ETC were similar, if the thermal conductivity of the filled layer was around 10 W/m K, whereas the performance increased by 12% if the filler material (graphite) thermal conductivity was around 100 W/m K.

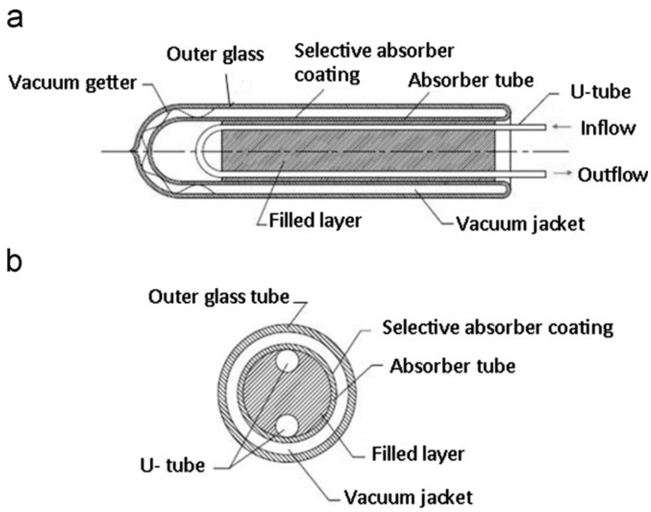

Fig. 6. The filled-type evacuated tube solar collector [75].

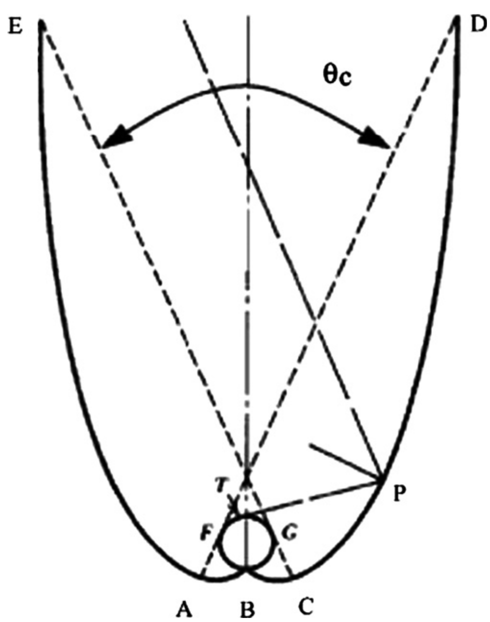

Fig. 7. Schematic diagram of a CPC collector [24].

3.1.3. Compound parabolic concentrators

Flat-plate collectors and evacuated tube collectors are widely used in solar thermal applications, mainly to provide low to intermediate temperatures (20 °C to 120 °C). However, to attain temperatures above this range, concentrators/reflectors have to be employed to maximize the incident radiation to affect high temperatures. Compound parabolic concentrators (CPC) solar collectors are non-imaging concentrators having the ability to reflect most of the incident radiation to the absorber. The schematic diagram of the most common type, Winston's CPC, is shown in Fig. 7, in which the lower sections of the reflector are circular (BA and BC) and the upper sections (AE and CD) are parabolic [24].

There are other types of non-imaging concentrators in recent years. Kaiyan et al. [79] introduced a novel type of CPC which consists of multiple curved surfaces that could provide forward reflective and imaging light beam. Fig. 8 shows the schematic of this new system which is composed of a parabolic and a flat contour. It can also be referred to as an imaging CPC. Its notable characteristic is that its focus has been moved to back side of the concentrator. The reflected light is transmitted forward from entrance aperture to exit aperture instead of backward, as in a traditional parabolic concentrator. There exist several other designs of CPC collectors that have been developed over the years which can efficiently operate at high temperature range [80–88].

One of them is a cost effective non-evacuated CPC collector incorporated with a flat or cylindrical absorber; however, they are not commonly used because of their inherent heat losses [80]. To improve the thermal efficiency of CPC collectors and to overcome heat loss issues, different designs of CPC reflector profiles were introduced. In conventional CPC reflector profile, the reflector is in contact with the absorber tube at the cusp region, which results in conductive heat losses [81]. To avoid the conductive heat loss from the absorber, it is necessary to create a gap between the tubular absorber and the reflector. However, this gap between the absorber and the envelope might marginally reduce the incoming radiation on the absorber tube. Winston et al. [82] developed an alternative design that preserved the ideal flux concentration on the absorber at the expense of slightly oversizing the reflector. This design helps in preventing gap losses but does not totally eliminate them. Another attempt was made by Oommen and Jayaraman [81,83] to overcome gap loss issues. A CPC reflector with a V-groove at the bottom of the reflector was designed and fabricated with a half acceptance angle of 23.5° for a

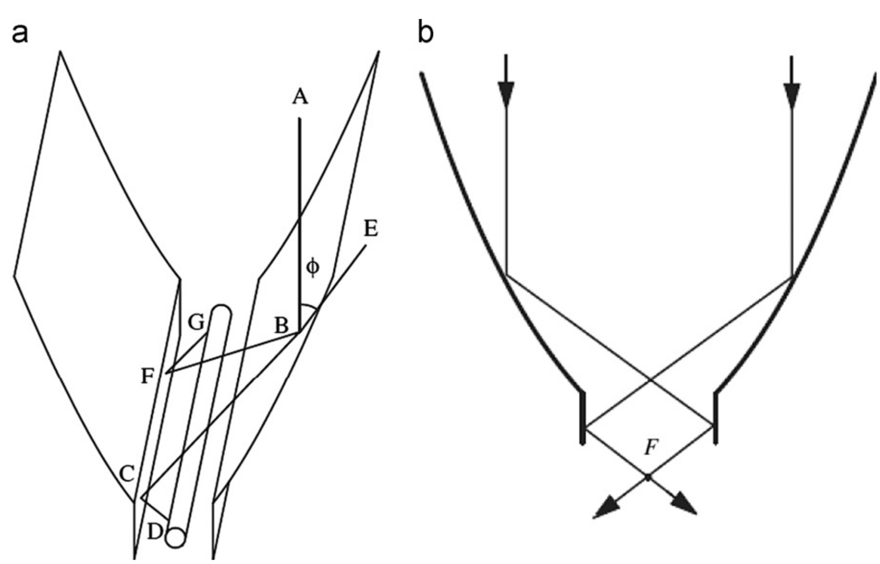

Fig. 8. Schematic of the multiple curved surface compound collector. (a) light transmission pattern; (b) concentric model and the passed light [79].

tubular absorber with an outer diameter of 30 mm. The study showed that the instantaneous efficiency of the CPC module was higher than the flat plate collectors, even at higher operating temperatures.

Apart from the above discussed CPC collector design, a new V-trough collector model compromising of five parallel reflector troughs aligned east-west was tested by Rönnelid [84]. V-troughs and CPC have the same geometry; therefore the results, in general, are valid for low-concentrating CPCs. Results indicated that in low-concentrating solar collectors the use of transparent insulation, like Teflon films, can significantly reduce the heat losses.

To further reduce the absorber thermal losses of the CPC collectors, three main design directions have been attempted. Rabl [85] introduced the first design direction which was based on an asymmetric mirror and inverted absorber configurations; this had helped to suppress the thermal losses due to convection and to achieve higher absorber temperatures [85]. Another study on a stationary asymmetric mirror collector was performed by Kienzlen et al. [86] and confirmed that to attain a reasonable level of optical efficiency and low thermal losses, mirrors of high reflectance and well insulated selective absorber were essential. The second design direction was the interposition of a transparent material between the absorber and the cover. One of the methods to achieve this type is to place thin Teflon film under the cover of a CPC collector with one face illuminating the flat absorber. Rönnelid et al. [84,87] constructed a CPC-collector with low U value for use in large-scale applications. Low emitting reflectors and a Teflon-film between the reflectors and the cover were introduced. Experimental data had confirmed that the heat losses from the absorber to the cover can be considerably reduced when compared to the CPC collector provided with only one cover and high emitting reflectors. Another method to achieve low heat losses is to place transparent baffles within the CPC cavity. Introduction of the baffle into the collector cavity reduces internal convection thereby reducing heat losses. Eames and Norton [88] studied the performance of a CPC collector by introducing the baffle into the cavity and confirmed that the collector efficiency was increased.

Introduction of advanced materials, such as absorber coatings with high absorptance and low emittance properties, high reflectance mirrors, and glazing with high transmittance, was the third direction to enhance the efficiencies of the CPC collectors. For this proposed direction, Tripanagnostopoulos et al. [86] designed and tested a non-evacuated stationary CPC solar collector with flat bifacial absorbers. Two experimental models were proposed to evaluate the performance of the proposed CPC type collector: the CPC-S model (single unit CPC), with one mirror-absorber unit, and the CPC-T model (three units CPC), with three smaller mirror-absorber units incorporated in one device. The CPC-S model was tested for different types of front/inverted absorbers, with respect to their emittance and reflectance properties. It was observed that among all the tested mirror-absorber types, the aluminized myler reflector with selective/black and selective/selective flat bifacial absorber showed the best results. Experimental results also confirmed that the CPC collectors based on the proposed mirror-absorber geometry could reduce thermal losses. Among the two models, CPC-T type was the most efficient.

3.2. *Storage tank*

The storage tank is an important component of the SWH system which plays a major role in dictating the system performance. A storage tank is commonly used to store the collected solar thermal energy providing the required hot water at the preferred temperature to the end uses [89]. Storage tanks are generally constructed using steel, concrete, plastic, fiber glass or other suitable materials used to store hot water. Among all the types mentioned, steel tanks are most commonly employed since they are easy to install [70].

One of the main problems experienced with the storage tank is heat losses due to the mixing of hot and cold water. To minimize the mixing and aid thermal stratification, several designs of storage tanks have been proposed. Stratification is essential to minimize the mixing and thereby affects the harvesting of maximum energy from the collector [90]. To evaluate the performance of stratified storage tanks, several experimental and theoretical studies were conducted for different geometrical designs and operational conditions. Geometrical factors include different orientations such as horizontal or vertical tanks, thickness of the wall, tank height to diameter ratio, and material of the tank. Operational factors include the temperature difference between inlet and outlet fluid stored in the tank, climatic conditions and the flow rate at the inlet and the outlet of the tank [91]. In 1979, Koppen et al. and Holland and Lightstone found that the increase in the thermal stratification in the storage tank helped in improving the thermal performance [92].

Better thermal stratification can be attained by using inlet stratifiers combined with low flow operation in the solar collector loop. Flexible fabric piping is generally used as an inlet pipe to affect stratification because they can expand or contract, equalizing the pressure in the pipe. Andersen et al. [92] investigated a number of different fabric pipes, made up of Nylon, filament polyester, spun polyester and acrylic, and compared them to the conventional non-flexible inlet stratifiers. The study compared the performance of pipes with one and two fabric layers. Results showed that enhanced performance was possible by using two fabric layers with a distance of about 10 mm between each fabric layer instead of using one thick layer. Yet another design was attempted by Shah et al. [93]. A CFD analysis was carried out on a rigid stratifier and was experimentally validated with PIV and temperature measurements. The stratifier consisted of a main tube with three circular opening and was mounted inside the storage tank. The stratifier was observed to be most efficient for flow rates between 5l/min and 81/min.

Recently, a storage tank design employing a double chimney device that acts as a thermal diode was examined with an objective to maximize temperature stratification. This design was experimentally tested by Rhee et al. [90] to measure the temperature stratification in a solar water storage tank. Compared to the baseline (fully mixed tank), the newly designed double chimney express-elevator design showed a three times improvement in temperature stratification. To further improve the thermal stratification, instead of conventionally used one outlet, the tank can be provided with two draw-off pipes, one at the very top and one in the middle of the tank. Other than minimizing heat losses, this method provides an additional advantage. That is, part of the hot water can be tapped through the lower draw-off pipe which is at a lower temperature. This will enhance the stratification in the tank as well as facilitate additional cold water to enter the tank, thereby increasing the collector efficiency as well as the overall thermal performance of SWH system. Furbo et al. [94] also confirmed that the thermal performance of the SWH system can be increased by using two draw-off levels from the solar tanks instead of one draw-off level at a fixed position. The study reported that the best position of the second draw-off level can either be fixed in the middle or just above the mid-point of the tank. With a realistic draw off of hot water at 45 °C, the thermal efficiency by the second draw off-level was 6%.

To enhance stratification, different inlet designs of the storage tank have been proposed. Recently, Altuntop et al. [89] investigated the effect of 12 different obstacles on the thermal

Table 1
Recent significant studies on solar water heating systems.

SWH system components	References	Salient design details	Temperature T (°C)	Efficiency η (%)	Advantages	Disadvantages
	Tripanagnostopoulos et al. [48]	– Absorber coated red, blue and reddish brown – Flat booster reflector – Polyurethane insulation – Water as working fluid	T_{outlet}=27 °C	$\eta_{collector=}$80%	– Cost effective – Thermal energy output is high	– Higher convection thermal losses – Low optical efficiency
	Rittidech and Wannapakne [51]	– Integrated closed-end oscillating heat pipe collector (CEOHP) – Black coated absorber plate – 50cm thick glass wool insulation – R134a as working fluid	T_{tank}=45 °C	$\eta_{collector}$=62%	– Cost effective – Simple construction – Corrosion-free operation – Elimination of winter icing problems	– Limited to low temperature applications – (∼40–50 °C)
Flat-plate collector	Sopian et al. [56]	– Thermoplastic natural rubber tubing as absorber plate – 50 mm foam insulation – Water as working fluid	T_{tank}=65 °C	$\eta_{collector}$=72%	– Non-corrosive – Easy to fabricate – Low manufacturing cost – High performance collector – Durable	– Low thermal conductivity and low useful-gain – Limited to low temperature applications – (∼50–60 °C)
	Abdullah et al. [63]	– Black-coated copper absorber plate – 15 mm glass fibre and 25 mm polystyrene insulation – Single and double honeycomb insulation – Water as working fluid	T_{outlet}=60 °C	$\eta_{collector}$=68%	– Reduce top heat losses	– Optimum gap thickness needs to be adjusted in order to enhance the thermal performance of solar collector
	Wazwaz et al. [65]	– Nickel-pigmented aluminum oxide selective absorber – Polystyrene transparent cover – Water as working fluid	T_{outlet}=40 °C	$\eta_{collector}$=78%	– Absorptivity is in the range of 0.91–0.97% – Emissivity in the range of 11.0–22.5%	– Requires frequent replacement of the transparent cover – Requires higher prototype volume for low flux conditions
	Kostic and Pavlovic [68]	– Selectively-coated aluminum sheet (absorber) with nickel sulphate solution – Two aluminum sheet reflector – Mineral wool insulation – Water as working fluid	T_{outlet}=37 °C	$\eta_{collector}$=52%	– Payback time is low – 35–44% energy gain	– Limited to low temperature applications
Evacuated-tube collector	Kim and Seo [71]	– Two-layered glass tube – Circular fin U-tube absorber – Air as working fluid	T_{outlet}=43.1 °C	$\eta_{collector}$=55.8%	– No shadow effect – Performance is not dependent on incident angle	– Low efficiency
	Shah and Furbo [73]	– Double-glass tubes – Tubular absorber – Water as working fluid	T_{outlet}=32 °C	$\eta_{collector}$= ∼60–70%	– Suitable for high latitudes – Higher heat transfer coefficient	– Shadow of the tubes lower the performance

	Morrison et al. [74]	– Aluminum sheet reflector – Water-in-glass etc – Water as working fluid	$T_{outlet}=26.5\ ^{\circ}C$	$\eta_{collector}=79\%$	– Simple in design – Low manufacturing cost – Can be used in different climatic zones	– Higher convective heat losses – Less optical efficiency
	Rittidech et al. [76]	– Closed-loop oscillating pipes – R-134a as working fluid	$T_{tank}=37\ ^{\circ}C$	$\eta_{collector}=76\%$	– Non-corrosive – No freezing issues	– Release of non-condensable gases – Difficult to maintain vacuum environment
	Ma et al. [78]	– Two-layered glass etc – U-shaped absorber tube – Water as working fluid	$T_{outlet}=38\ ^{\circ}C$	$\eta_{collector}=59\%$	– Reduced heat losses – High heat extraction – Collector efficiency increases within synthetic conductance	– Difficult to maintain – vacuum environment
	Kaiyan et al. [79]	– Multiple curved-surface compound concentrator – Water as working fluid	$T_{outlet}=80\ ^{\circ}C$	$\eta_{collector}=71\%$	– High intercept factor – Receiver and its support rack could be installed at the bottom	– Requires tracking – Requires larger surface area than the parabolic dish – High reflective losses
Compound parabolic concentrator	Oommen and Jayaraman [81,83]	– V-groove CPC – Selectively-coated copper tubes – Aluminum polyester foil reflector – Glass wool insulation – Water as working fluid	$T_{outlet}=66\ ^{\circ}C$	$\eta_{collector}=60\%$	– Reduced gap losses – Low thermal losses – Can be used as pressure steam generators	– Requires tracking – Limited to medium temperature applications
	Rönnelid et al. [84,87]	– V-Trough model – Flat absorber with 5 parallel-reflector trough – Aluminum reflector foil – Teflon film insulation – Water as working fluid	$T_{outlet}=80\ ^{\circ}C$	$\eta_{collector}=\sim 65–75\%$	– Easier to manufacture – High thermal stability	– Aluminum laminate shrinks at 150–160 °C – High emittance results in higher heat losses
	Altuntop et al. [89]	– Thermally-stratified cylindrical tank – Cylindrical \baffle plate	$T_{tank}=47\ ^{\circ}C$	$\eta_{tank=}\sim 50–60\%$	– Direct contact area between cold and hot water in the tank is reduced – High thermal stratification	– Installation of baffle plates is laborious – High material cost
	Rhee et al. [90]	– Thermal express-elevator partition device serves as stratifier – 25 mm thick polyurethane partitions	$T_{tank}=28\ ^{\circ}C$	$\eta_{express\ elevator}=52\%$	– Express elevator design showed 3 times improvement in thermal stratification compared to the fully mixed tank	– Design optimization is required to identify the optimum positioning of the partition device
Storage tank	Shah et al. [93]	– Rectangular glass tank – A pipe with opening serves as the stratifier with flaps – Flaps serves as non-return valves	$T_{stratifier}=51\ ^{\circ}C$	$\eta_{stratifier}=92\%$	– Flaps reduce unwanted flow into the stratifier – Most efficient for flow rates of 5–8 L/min	– Acts more as a mixing device than as a stratifying device because mixed fluid enters the tank through the top opening – Fluid is sucked into the stratifier through the lowest hole
	Furbo et al.[94]	– Standard mantle tanks – Two draw off levels – Two cross-linked polyethylene pipes acts as the stratifier for draw off at	$T_{tank}=45–50\ ^{\circ}C$	Not available	– Better thermal performance – Increased solar collector efficiency	– Suitable only for two flow systems

Table 1 (*continued*)

SWH system components	References	Salient design details	Temperature *T* (°C)	Efficiency η (%)	Advantages	Disadvantages
		the top and middle of the tank				
	Shah and Furbo. [96]	– Transparent tank with 5 mm plexiglass walls – Variable inlet design – Unstructured meshes – Small hemispherical baffle plate	T_{tank}=42 °C	η_{exergy}=70%	– Less mixing – Exergy is higher for an ideal draw-off conditions	– Energy quality is reduced with poor inlet design
	Alizadeh [99]	– Thermally stratified horizontal cylindrical tank – Divergent conical tube as the inlet nozzle	T_{tank}=42 °C	η_{tank}=~55–65%	– Cost effective in both the construction and operational phases	– Susceptible to degradation of thermal stratification – Limited to integrated collector storage tank water heating systems
	Spur et al. [103]	– Novel, partitioned stratified tank with lower half being preheated – Horizontal divider plate with a hole in the center – Helical copper pipe heat exchanger – Counter flow – 4 cm thick glass wool insulation	T_{hxout}=75 °C	15% higher	– High heat extraction rate	– Limited to only medium temperature conditions (60–70 °C)
Heat exchanger	Morrison et al. [108]	– Horizontal annular mantle heat exchanger – Cross flow	T_{tank}=27–50 °C	Not available	– Simple design – Large heat exchange surface area – Good overall heat transfer	– Does not provide an effective connection between the collector and the tank – Thermal stratification degrades, but could be improved by repositioning the inlet port at a higher position
	Wang et al. [113]	– Shell and tube heat exchanger with spiral groove tube bundle – Cross flow – Baffle plates – Fibre glass insulation	T_{tank}=75–80 °C	η_{system}=~70–80%	– Superior shell-side heat transfer per unit length – Light-weight – Lower manufacturing cost – Pressure drop comparable to the smooth tube bundle – Considerable increase in flow rate	– Requires auxiliary heating during cloudy days

stratification in a cylindrical storage tank by predicting the temperature distribution within the tank. Fig. 9 shows the twelve different obstacle geometries analyzed in the study. Results indicated that placing an obstacle in the tank provided better thermal stratification compared to the case with no obstacle. Also, the obstacles numbered 7, 8 and 11 which had a gap in the center, resulted in better thermal stratification than those that had a gap near the tank wall. The study had concluded that, in terms of hot water supply, the obstacle 11 provided the best thermal stratification in the tank.

Thermal stratification can also be improved by incorporating baffle plates within the tank. Baffle plates help in guiding the fluid flow and effectively decrease the impingement of the fluid [95]. The effect of the inlet design with diverse baffle plates on the fluid's movement in the tank was numerically as well as experimentally investigated by Shah and Furbo [96]. Diverse baffle plate designs include hemispherical baffle plates, large flat baffle plates and raw pipes, and were tested at different discharge flow rates. It was found that the conical baffle plates followed by the flat baffle plates resulted in the best performance, as they progressively increased the flow rate in the tank within the space between the tank walls.

Several configurations of the storage tanks (e.g., vertical and horizontal) had been investigated to evaluate their thermal performance. Dehghan et al. [97] experimentally investigated the thermal performance of a vertical water tank. Under realistic conditions, hourly measurements of the temperature distribution inside the tank, collector's flow rate, and its inlet and outlet temperatures were considered in evaluating the thermal characteristics of the system. Experiments were conducted mainly during summer time and it was observed that the thermal stratification was well established for at least 11 h of a day.

Transient thermal behavior of a vertical storage tank with a mantle heat exchanger in the charging mode was numerically investigated by Barzegar and Dehghan [98]. Along with the process of formation of thermal stratification in the storage tank, the influence of the Rayleigh number in the storage tank and the Reynolds number for the flow through the mantle heat exchanger were also studied. It was found that a better thermal stratification was established for higher values of Rayleigh number. Similarly, an increase in the incoming fluid velocity through the mantle heat exchanger could affect a faster formation of the thermal stratification. It was also noticed that superior thermal performance can be achieved when the mantle heat exchanger was placed at a midlevel height of the storage tank.

In continuation to the above work, Dehghan and Barzegar [91] recently (2011) investigated a similar transient thermal performance behavior of a vertical storage tank with a mantle heat exchanger in the discharge mode. The study had confirmed that for higher values of Grashof number, the pre-established thermal stratification was well maintained during the discharging mode of operation. It was reported that the size of the inlet port also plays a significant role on thermal characteristics of the tank. The study suggests that, in order to ensure the least amount of mixing inside the tank during the discharge consumption period, the inflow Reynolds number and port diameter should be kept below a certain value.

Besides the vertical storage tank design, the horizontal cylindrical storage configuration is a very common design adopted in thermosyphon SWH system. This geometry requires a more detailed thermal performance analysis as it is more susceptible to degradation of thermal stratification due to its small vertical dimension relative to its horizontal length. Alizadeh [99] studied the thermal behavior of a horizontal cylindrical storage tank. Four sets of experiments were conducted and the study reported that using a divergent conical tube as the inlet nozzle will help in obtaining better thermal stratification due to the diffusion effect of the nozzle. Also, experiments showed that the performance could be improved if the initial bottom temperature of the tank was higher than the injected cold water temperature. Yet another study on horizontal tanks was performed by Helwa et al. [100] to investigate the effect of hot water consumption rate on the temperature distribution inside a horizontal storage tank. Experimental results concluded that the thermal stratification inside the tank was highly dependent on the pre-assumed load pattern. In comparison with a vertical storage tank, for the same given input conditions, the thermal performance of a horizontal tank was inferior [101].

3.3. Heat exchanger

In an indirect type of SWH system, a heat exchanger (HX) is utilized to transfer absorbed solar heat from the working fluid to the storage tank. The HXs are generally made from conductive material such as aluminum, stainless steel, cast iron, copper, steel, and bronze. To ensure good thermal conductivity and resistance to corrosion, copper is popularly used in SWH system. Several configurations of HXs have been designed for indirect water heating storage tanks and the most common configurations are: the immersed coil-in-tank, the shell-and-tube, and the mantle heat exchanger [95]. Several configurations of HXs have been proposed to improve the overall performance of the SWH system [102–113].

The coil-in-tank heat exchanger is a coil containing several loops of tubing made up of either single or double wall immersed in the storage tank. Farrington and Bingham [102] tested and analyzed the efficiency of load side immersed HX and confirmed that a smooth coil HX outperformed the finned HX with the same outside surface area. The study had reported that the load-side

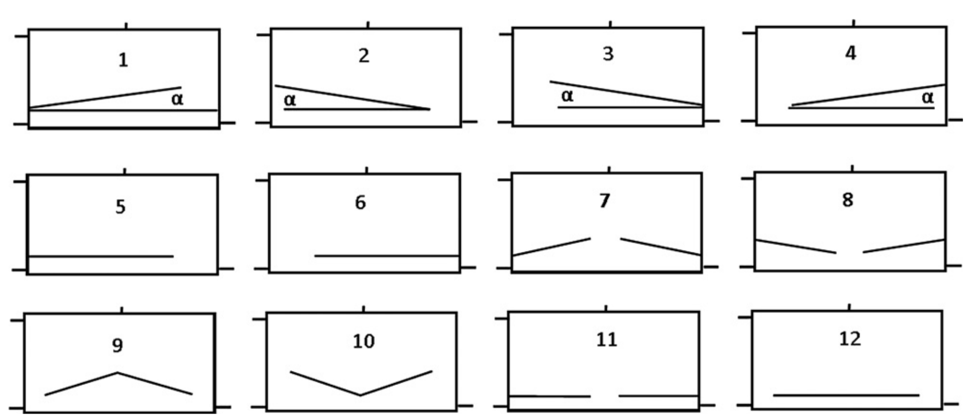

Fig. 9. Obstacles geometries and their assembly in the tank [89].

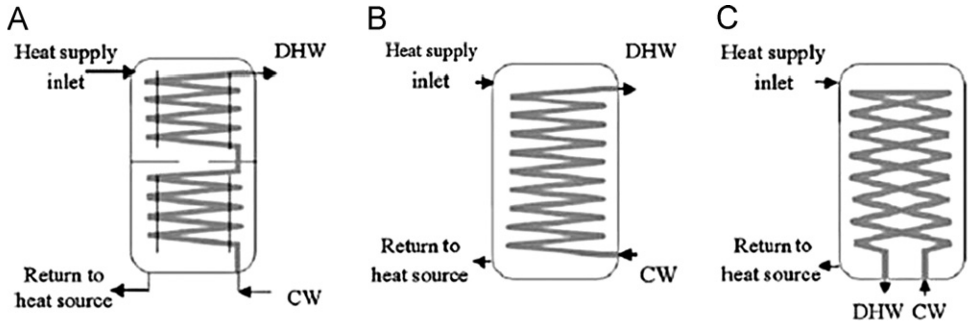

Fig. 10. Schematic design of Tanks A, B and C [103].

HX also served as an obstacle and thereby improved the stratification in the storage tanks.

Several configurations of the coil-in-tank HX were studied by Spur et al. [103]. The behavior of a novel stratified (Tank A), and two standard tanks (Tank B and C) were analyzed (Fig. 10). A novel-stratified storage tank (A) was chosen to determine the effect of an improved inner storage design and was compared with the most commonly used design B and C. Tank B had a HX which was coiled upward from the bottom to the top of the tank whereas the HX in tank C was coiled upward from the bottom to the top of the tank and then with a U-turn towards downward the bottom of the tank. The study concluded that the stratified tank was up to 32% more effective when compared to the commercially available tank. Results had confirmed that to attain maximum heat extraction, the immersed HX position should be coiled upwards and located in the upper part of the tank [103].

Another common configuration of HX is shell-and-tube exchanger which has two varieties: the first type is an external HX, where the cold water is drawn from bottom of the tank by using a second circulating pump and is allowed to flow through the secondary side of the HX in a counter-flow method, and the second type is an external HX with a bypass circuit, in which a three-way valve is provided at two different heights [95]. When compared to the internal heat exchanger, an external HX has better thermal transfer properties [104]. Parent et al. [105] analyzed the performance of the shell-and-tube HX external to the storage tank, in which fluid flow was induced by natural convection. Compared to the immersed coil type configuration this design was found to be more practical and the effectiveness of the HX could reach as high as 90%.

The mantle HX is yet another configuration which helps in providing a large heat transfer area for a given volume of collector fluid flow in the mantle. This makes the mantle HX as one of the efficient and simplest ways of facilitating high HX effectiveness in promoting thermal stratification. One of the prominent advantages of this design is that both the hot water tank and the heat exchanger are combined into one unit [95]. Numerous analytical and experimental studies had been intensively carried out on mantle heat exchangers to improve their performance [106,107].

Research on the horizontal mantle HX was performed by Morrison et al. [108]. The characteristics of the horizontal mantle HX for application in thermosyphon solar water heaters were investigated. To evaluate the flow patterns in the annular passageways and the heat transfer into the inner tank, an experimental prototype was built. Results had indicated that the mantle HX configurations used in solar water heater applications degraded the thermal stratification in the inner tank. To improve the stratification characteristics of the said type of HX, it was suggested to place the inlet port at a higher position in the annulus to direct the inlet jet parallel to the annular gap rather than perpendicular to the tank wall, and also to employ a stratification guide for the incoming jet.

The vertical mantle HX for forced circulation SWH system was experimentally investigated by Buar et al. [109]. A simple Nusselt number correlation was used to evaluate the heat transfer for the laminar flow between the plates if finite difference method was employed to solve the energy balance equations. Their experimental data showed that there was very little improvement in annual performance between the vertical mantle and external HX. Shah and Furbo [110] also researched the performance of the vertical mantle HX. A numerical model was developed to analyze the external flow over a vertical flat plate to evaluate the convective heat transfer coefficient. Simulation results had shown that the temperature stratification in the inner tank was not affected by the mantle fluid as long as the fluid flow was restricted within the mantle gap until it reached the thermal equilibrium level with the water in the inner tank.

To further improve the mantle HX design it is important to examine the heat transfer as well as the fluid motion in the tank and how it influences the thermal stratifications. Recently, Knudsen et al. [111] investigated the flow structure inside the inner tank and the heat transfer in vertical mantle HXs. A PIV system was used to measure the flow structure and velocities in the inner tank as well as within the mantle. Also, a CFD model of a vertical mantle HX was developed to evaluate the heat flux on the mantle and the tank. The heat transfer analysis predicted a mixed convection flow in the mantle near the inlet and that the heat transfer was directly proportional to the mantle inlet temperature relative to the core tank temperature.

The multi-tube type of double-tube HX was newly designed by Shi et al. [112], which consisted of a group of HX units with a tube bundle arranged inside an outer tube. The outer tube of the HX unit is packed with a tube bundle, resembling a shell-and-tube HX of several meters length. This multi-tube type of double-tube HX helps in increasing the flow rate and heat transfer area per unit length, while reducing the pressure drop on the shell side of the HX. Wang et al. [113] had extended their work and introduced yet another design namely, RODbaffle multi-tube type of double-tube HX. Experimental results showed that the RODbaffle multi-tube type HX with spiral groove tube bundle provided superior shell-side heat transfer and pressure drop characteristics compared to the plate baffle type (Table 1). It was reported that this new multi-tube type of double-tube heat exchanger has a promising future in the petrochemical industry.

3.4. Heat transfer fluid

Heat transfer fluid (HTF) plays a very significant role in a solar collector system. In a SWH system, HTF absorbs energy in the collector and transmits the energy through the heat exchanger to the water in the storage tank. The HTF properties such as boiling point, freezing point, flash point, viscosity, and thermal capacity play a role in the selection of a working fluid for a SWH system.

For example, a SWH system will require a HTF with a low freezing point and a high boiling point in cold and hot climates, respectively. Some of the common HTFs used are air, water, hydrocarbon oils, Glycol/water mixture, and refrigerants/phase-change liquids [114,115].

Air and water have been commonly used as HTFs in SWH systems. Air has certain advantages compared to water. That is, it is non-corrosive in nature and has no issues such as boiling/freezing. However, it's very low heat capacity enforces limitation to low temperature applications and is not used for domestic water heating purposes. On the other hand, water's high specific heat, such as, its low viscosity, non-toxic, and less expensive features have positioned water to be the most popular working fluid in SWH systems. However, its corrosive nature (especially at high temperatures) as well as freezing and scaling issues poses a challenge in the design of collector tubing and plumbing. To overcome relatively high freezing point of water, a glycol additive is generally used along with water to act as an antifreeze [114].

Refrigerants are generally used as HTFs in heat pumps, refrigerators, and air conditioners. The use of refrigerant-filled solar collectors in a heat pump system is one of the important developments in the heat pump and solar technology. Refrigerants are more effective than other fluids especially during cloudy conditions because of their low boiling points, and high heat capacities. Several research projects have been conducted to study the effectiveness in operating the SWH system involving 2-phase heat transfer process. The fundamental thermal performance of two thermosyphon solar collectors, charged with acetone, and petroleum-ether, was studied by Soin et al. [116,117]. It was reported that the efficiency of a two-phase collector was only about 6% to 11% which is considerably lower compared to a direct thermosyphon SWH system.

In indirect (closed-loop) SWH systems, chlorofluorocarbon (CFC) refrigerants are more commonly used as HTFs for their stability, nonflammable, and noncorrosive properties, low toxicity and low freezing point. Specific examples include: R-11, R-12, R-13, R-113, R-114, and R-115.

Schreyer [118] investigated the effectiveness of R-11 refrigerant in the thermosyphon collector for domestic applications. Experimental results proved that the peak instantaneous efficiency for a two-phase refrigerant was higher than using the hydronic fluid. Similar work was carried out by Farrington et al. [119] for a single-family residence; the SWH system when operated based on forced circulations, could realize an efficiency of about 35%. Joudi and Al-Tabbakh [120] also confirmed that the collector efficiency of two-phase system was about 20% higher than a single phase collector. The system could attain instantaneous efficiency as high as 60%, which was in the range of conventional solar water collectors [121]. To further analyze the R-11 driven SWH system, Fanney and Terlizzi [122] had developed correlations to predict the heat transfer by R-11 and were validated with experimental data.

One of the disadvantages of refrigerants being used for closed-loop systems is their low efficiency compared to the conventional open-loop systems [123]. However, it is possible to increase the efficiency of the SWH systems by charging the collector to the fullest ensuring the fluid levels in the inlet of the tubes to be nearly the same as the outlet [124]. Radhwan and Zaki [118] depicted the thermal non-equilibrium vapor generation process with fully R-11 refrigerant filled collector tubes and showed the dependency of the thermophysical properties (temperature, pressure) of the refrigerant on the circulation flow rate while in phase change.

CFCs are harmful to the global environment due to their chlorine content, chemical stability, and they have both ozone depletion potential (ODP) and Global Warming Potential (GWP). Thus, the use of CFCs was phased out and hydrochlorofluorocarbons (HCFC) were introduced [125]. These HCFC also contain chlorine but have much lower ODP and GWP than CFCs, typically 2–5% and 20%, respectively, in comparison to CFC-12. Examples of HCFC refrigerants include R-22, R-401, R-402, R-403, R-408, etc. [125].

HCFCs and hydroflurocarbon (HFC) heat transfer fluids were also tested for their effectiveness on the SWH system performance. Some of the significant studies are mentioned in this section. Payakaruk et al. [126] investigated refrigerants, such as R-22, R-123, R134a, ethanol, and water at different filling ratios of 50%, 80%, as well as 100% and reported their influence on the thermal efficiency of the system. It was reported that the heat transfer characteristics of the system depend on the latent heat of vaporization of the HTF; that is, the lower the latent heat of vaporization the higher the heat transfer rate.

Esen and Esen [121] also studied the thermal performance of a closed two-phase thermosyphon heat pipe based solar collector using the refrigerants R-134a, R-407C, and R-410A. Results showed that the performance of each of the refrigerants were similar with a maximum collection efficiency of around 50%, which was much higher compared to R-22. This may be due to the fact that, these refrigerants posses properties of high thermal conductivity, latent heat as well as low viscosity, which are conducive in effecting improved COP.

Yet another comparative study was performed by Kim [127] who tested R-22 and four other alternative fluids such as a R-134a, R-32/134a (30/70%), R-407C and R-410A. The study was conducted in an experimental breadboard water-to-water heat pump in which a water/ethylene glycol mixture was used as a heat transfer fluid. R-410A was found to have the highest COP, provided the thermal capacities were maintained the same.

HCFCs and HFCs are considered to be short-term HTFs. Recently; natural fluids have been considered as long-term HTF and have successfully replaced the HCFC and HFC refrigerants, which are still harmful to the environment. Long-term HTF are halogen-free natural working fluids. They are environmentally benign because of their very low or near zero ODP and GWP [128]. Some of the common examples of natural HTFs are propane (R-290), butane (R-600), iso-butane (R-600a), propylene (R-600), ammonia (R-717) and carbon dioxide (R-744) [125].

Natural working fluids are good substitutes for R-22 and researchers have proved this through various studies. Theoretical performance calculations for three refrigerants such as R-12, R-22, and propane for low and medium ranges of evaporating temperatures were compared by Charters et al. [129]. The three refrigerants showed that their COPs were the same at low evaporating temperatures (-20 °C to -10 °C), whereas at medium evaporating temperatures (0 °C to -10 °C), R-12 was found to have the highest COP. Further, Urma [130] conducted an experimental study and confirmed that it was feasible to use propane as a drop-in substitution for R-12 in small domestic refrigeration units. Another comparative study was conducted by Stoecker [131] where R-717 was compared with other refrigerants such as R-22, R-123, and R-134a and confirmed that R-717 is a viable candidate as a refrigerant for district cooling plant water chiller.

Another promising natural fluid is carbon dioxide (CO_2) as it is non-flammable, non-corrosive, and non-toxic to the environment. CO_2 can be used as a refrigerant in a transcritical heat pump cycle as is has a low critical point (31.1 °C at 73.7 bars) [132]. Research on CO_2 has gained momentum in recent years and the main objective is to investigate the possibility of applying CO_2 to heat pump SWH systems and also to investigate the performance of a transcritical CO_2 heat pump. Nekså et al. [133] recognized that heating tap water is one of the promising applications for a transcritical CO_2 system. A CO_2 heat pump water heater may produce hot water temperature up to 90 °C without any operational problems and the primary energy consumption can be reduced by more than 75% compared to electrical systems.

Further, the performance of a water-source transcritical CO_2 heat pump for water heating was examined by White et al. [134]. The theoretical analysis showed that it was possible to produce hot water at temperature around 120 °C with COP of about 21%. Cecchinato et al. [135], also carried out a similar study on CO_2 and reported that the transcritical cycle was efficient only if its characteristics were exploited through a suitable design.

The performance of transcritical CO_2 cycle with an internal heat exchanger was further investigated by Kim et al. [136]. The study relayed that optimum discharge pressures to affect maximum COP for a given internal HX length. However, the optimum pressure values decreased, if the length of the HX was increased. Also, it impacted the flow rate and compressor power to lower values. In addition to the both above observation, it was also noticed that with an increase in the HX length, though COP was enhanced, the heating capacity had decreased marginally due to the trade-off between the effectiveness and pressure drop in the internal HX.

Yet another simulation study on an air-source heat pump water heater using CO_2 as HTF was conducted by Laipradit et al. [137]. The effects of compressor speed, the temperature of inlet water at the gas cooler and the mass flow rate ratio of water and CO_2 on an air source heat pump were analyzed. The temperature of water was between 55 °C and 75 °C for the system investigated which had a 4 kW compressor. For, hot water with a high temperature, higher compressor speed and higher compression work were required, which led to lower COP. The study concluded that, a decrease in the inlet water temperature and an increase in the air temperature would increase the COP. It was recommended that a suitable flow rate ratio of water and refrigerant should be used to obtain high COP level at the required hot water temperature.

The potential of the supercritical CO_2-based collector in the field of solar thermal utilization was further studied by Zhang and Yamaguchi [138]. The experimental data showed that the temperature, pressure and mass flow rate of CO_2 increased with solar radiation. The study also showed that its performance, when operated in transcritical condition was much higher compared to the water-based collector.

4. Conclusion

Renewable energy research has become increasingly important since the signing of the Kyoto Protocol. Solar water heating (SWH) is one of the most effective technologies to convert solar energy into thermal energy and is considered to be a developed and commercialized technology. However, there exist opportunities to further improve the system performance to increase its reliability and efficiency. A concise review primarily on the design features and related technical advancements of the SWH systems in terms of both energy efficiency and cost effectiveness has been presented.

Several solar water heating designs have been introduced in the market and are more commonly utilized in the tropical regions of developing countries. Recent developments in heat pump based solar collector technology exhibit a promising design to utilize solar energy as a reliable heating source for water heating applications in solar adverse regions. Heat pump based solar water heating is influenced by many factors including the nature of the refrigerant. Due to the environmental concerns, the refrigerants with high global warming potential have come under scrutiny and several have already been phased out. Driven in part by these concerns, new refrigerants are being sought out and "old" refrigerants such as carbon dioxide, ammonia and propane are being investigated. Apart from the choice of working fluid, there has been a major research focus in improving the performance of various components of the SWH system. These developments would further pave a way to increase its market penetrability along with providing significant environmental and financial benefits.

Acknowledgements

This research was supported by a grant from the Pakistan-US Science and Technology Cooperation Program, US Department of State (jointly administered by the National Academics and Higher Education Commission of Pakistan).

References

[1] Energy Information Administration, Energy consumption and expenditure tables of the 2005 residential energy consumption survey, EIA, Washington, DC (2005) accessed January 2001. 〈http:www.cia.gov/emcu/recs/2005/c&e/detailed_tables2005 c&e.html〉.
[2] Renewable Energy Policy Network, Renewable 2011: Global Status Report. 〈http://www.ren21.net/gsr〉.
[3] 〈http://en.wikipedia.org/wiki/Solar_water_heating〉.
[4] Martin RH, Perez-Garcia J, Garcia A, Garcia-Soto FJ, Lopez-Galiana E. Simulation of an enhanced flat-plate solar liquid collector with wire-coil insert devices. Solar Energy 2011;85:455–69.
[5] Kemp CM. US patents no 451384, April 28, 1891. (Patent Records).
[6] Brooks FA. Solar energy and its use for heating water in California, Agricultural Bulletin 602. California Agricultural Experiment Station. Berkeley, CA, USA: University of California; 1936. (Bulletin).
[7] Tanishita I. Present situation of commercial solar water heaters in Japan, Transactions on the use of Solar Energy. Tucson, Arizona: The Scientific Basis; 1955 67-78. (Article).
[8] Burns S, Zeenni E, Guven HM. Stratification and performance of a solar bread box water heater, Proceedings of the American solar energy conference, San Diego, CA, USA 1985 p. 430–435. (Conference Paper).
[9] Savorin J. Study of solar water heating in Algeria. Proceedings of UNESCO, conference on sew sources of energy, Rome, Italy. Vol. 51961 p. 93–101. (Conference Paper).
[10] Smyth M, McGarrigle P, Eames PC, Norton B. Experimental comparison of alternative convection suppression arrangements for concentrating integral collector storage solar water heaters. Solar Energy 2005;78:223–33.
[11] Ecevit A, Al-Shariah AM, Apaydin ED. Triangular built-in-storage solar water heater. Solar Energy 1989;42:253–65.
[12] Kaushik SC, Kumar R, Garg HP, Prakash J. Transient analysis of a triangular built-in-storage solar water heater under winter conditions. Heat Recovery Systems and CHP 1994;14:337–41.
[13] Abdel-Rehim ZS. New design of solar water heater. Journal of Engineering and Applied Sciences 1997;44(3):583–97 (Article).
[14] McCracken H. How to build a passive solar water heater. CA,USA: Carlos: Alturas; 1978 (Article).
[15] Goetzberger A, Rommel M. Prospects for integrated storage collector systems in central Europe. Solar Energy 1987;39:211–9.
[16] Davis WD, The climax-cusp solar water heater, Proceedings of second national passive solar conference, Philadelphia, Pennsylvania, USA 1978. (Conference Paper).
[17] Stickney BL, Aaboe EH. Comparative performance indices for solar batch water heaters, Proceedings of the sixth national passive solar conference, Portland, Oregon, USA 1981p. 131–135. (Conference Paper).
[18] Schmidt C, Goetzberger A. Single-tube integrated collector storage systems with transparent insulation and involute reflector. Solar Energy 1990;45: 93–100.
[19] Van Straaten JP. The utilization of solar energy part lll: solar water heaters for low cost housing, Proceeding of the second south-eastern conference on application of solar energy, Baton Rouge, Louisiana, USA 1976 p. 56–66. (Article).
[20] Taylor RH US Patent no. 2388940, November 13th, 1945.
[21] Sokolov M, Vaxman M. Analysis of an integral compact solar water heater. Solar Energy 1983;30:237–46.
[22] Kaushik SC, Kumar R, Garg HP. Effect of baffle plate on the performance of a triangular built-in-storage solar water heater. Energy Conversion and Management 1995;36:337–42.
[23] Mohamad AA. Integrated solar collector–storage tank system with thermal diode. Solar Energy 1997;61:211–8.
[24] Kalogirou SA. Solar thermal collectors and applications. Progress in Energy and Combustion Science 2004;30:231–95.

[25] Close DJ. The performance of solar water heaters with natural circulation. Solar Energy 1962;6:33–40.
[26] Ong KS. An improved computer program for the thermal performance of a solar water heater. Solar Energy 1976;18:183–91.
[27] Sodha MS, Tiwari GN. Analysis of natural circulation solar water heating systems. Energy Conversion and Management 1981;21:283–8.
[28] Grossman G, Shitzer A, Zvirin Y. Heat transfer analysis of a flat-plate solar energy collector. Solar Energy 1977;19:493–502.
[29] Huang BJ. Similarity theory of solar water heater with natural circulation. Solar Energy 1980;25:105–16.
[30] Ng KC, Yap C, Khor TH. Outdoor testing of evacuated-tube heat-pipe solar collectors, Proceedings of the Institution of Mechanical Engineers 2000; 214(part E): 23–30. (Article).
[31] Redpath DAG, Eames PC, Lo SNG, Griffiths PW. Experimental investigation of natural convection heat exchange within a physical model of the manifold chamber of a thermosyphon heat-pipe evacuated tube solar water heater. Solar Energy 2009;83:988–97.
[32] Budihardjo I, Morrison GL. Performance of water-in-glass evacuated tube solar water heaters. Solar Energy 2009;83:49–56.
[33] Huang J, Pu S, Gao W, Que Y. Experimental investigation on thermal performance of thermosyphon flat-plate solar water heater with a mantle heat exchanger. Energy 2010;35:3563–8.
[34] Jaisankar S, Radhakrishnan TK, Sheeba KN. Experimental studies on heat transfer and friction factor characteristics of forced circulation solar water heater system fitted with helical twisted tapes. Solar Energy 2009;83: 1943–52.
[35] Jaisankar S, Radhakrishnan TK, Sheeba KN, Suresh S. Experimental investigation of heat transfer and friction factor characteristics of thermosyphon solar water heater system fitted with spacer at the trailing edge of Left–Right twisted tapes. Energy Conversion and Management 2009;50:2638–49.
[36] Li Z, Chen C, Luo H, Zhang Y, Xue Y. All-glass vacuum tube collector heat transfer model used in forced-circulation solar water heating system. Solar Energy 2010;84:1413–21.
[37] Chong KK, Chay KG, Chin KH. Study of a solar water heater using stationary V-trough collector. Renewable Energy 2012;39:207–15.
[38] Chaturvedi SK, Chen DT, Kheireddine A. Thermal performance of a variable capacity direct expansion solar-assisted heat pump. Energy Conversion and Management 1998;39:181–91.
[39] Kuang YH, Sumathy K, Wang RZ. Study on a direct-expansion solar-assisted heat pump water heating system. International Journal of Energy Research 2003;27:531–48.
[40] Li YW, Wang RZ, Wu JY, Xu YX. Experimental performance analysis on a direct-expansion solar-assisted heat pump water heater. Applied Thermal Engineering 2007;27:2858–68.
[41] Chyng JP, Lee CP, Huang BJ. Performance analysis of a solar-assisted heat pump water heater. Solar Energy 2003;74:33–44.
[42] Huang BJ, Lee JP, Chyng JP. Heat-pipe enhanced solar-assisted heat pump water heater. Solar Energy 2005;78:375–81.
[43] Xu G, Zhang X, Deng S. A simulation study on the operating performance of a solar-air source heat pump water heater. Applied Thermal Engineering 2006;26:1257–65.
[44] 〈http://en.wikipedia.org/wiki/Solar_thermal_collector#Flat_plate_collectors〉.
[45] Hottle HC, Whiller A, Evaluation of flat-plate solar collector performance, vol.2. Transaction of conference on the use of Solar Energy, University of Arizona; 1958 p. 74–104. (Conference Paper).
[46] Matrawy KK, Farkas I. Comparison study for three types of solar collectors for water heating. Energy Conversion and Management 1997;38:861–9.
[47] Duffie J A, Beckman WA. Solar engineering of thermal processes. New York: Wiley & Sons; 1980 (Article).
[48] Tripanagnostopoulos Y, Souliotis M, Nousia T. Solar collectors with colored absorbers. Solar Energy. 2000;68:343–56.
[49] Rommel M, Moock W. Collector efficiency factor F' for absorbers with rectangular fluid ducts contacting the entire surface. Solar Energy 1997;60:199–207.
[50] Alvarez A, Cabeza O, Muñiz MC, Varela LM. Experimental and numerical investigation of a flat-plate solar collector. Energy 2010;35:3707–16.
[51] Rittidech S, Wannapakne S. Experimental study of the performance of a solar collector by closed-end oscillating heat pipe (CEOHP). Applied Thermal Engineering 2007;27:1978–85.
[52] Kovarik M. Optimal distribution of heat conducting material in the finned pipe solar energy collector. Solar Energy 1978;21:477–84.
[53] Hollands KGT, Stedman BA. Optimization of an absorber plate fin having a step-change in local thickness. Solar Energy 1992;49:493–5.
[54] Kundu B. Performance analysis and optimization of absorber plates of different geometry for a flat-plate solar collector: a comparative study. Applied Thermal Engineering 2002;22:999–1012.
[55] O'Brien-Bernini FC, McGowan JG. Performance modeling of non-metallic flat plate solar collectors. Solar Energy 1984;33:305–19.
[56] Sopian K, Zulkifli R, Sahari J, Othman MY. Thermal performance of thermoplastic natural rubber solar collector. Journal of Materials Processing Technology 2002;123:179–84.
[57] Liu W, Davidson J, Mantell S. Thermal analysis of polymer heat exchangers for solar water heating: a case study. Journal of Solar Energy Engineering 2000;122:84–91.
[58] Tsilingiris PT. Back absorbing parallel plate polymer absorbers in solar collector design. Energy Conversion and Management 2002;43:135–50.
[59] Dorfling C, Hornung CH. Hallmark B, Beaumont RJJ, Fovargue H, Mackley MR. The experimental response and modelling of a solar heat collector fabricated from plastic microcapillary films. Solar Energy Materials and Solar Cells 2010;94:1207–21.
[60] Furbo S, Jivan Shah L. Thermal advantages for solar heating systems with a glass cover with antireflection surfaces. Solar Energy 2003;74:513–23.
[61] Akhtar N, Mullick SC. Computation of glass-cover temperatures and top heat loss coefficient of flat-plate solar collectors with double glazing. Energy 2007;32:1067–74.
[62] Whiller A, Plastic covers for solar collectors (Book).
[63] Abdullah AH, Abou-Ziyan HZ, Ghoneim AA. Thermal performance of flat plate solar collector using various arrangements of compound honeycomb. Energy Conversion and Management 2003;44:3093–112.
[64] Hellstrom B, Adsten M, Nostell P, Karlsson B, Wackelgard E. The impact of optical and thermal properties on the performance of flat plate solar collectors. Renewable Energy 2003;28:331–44.
[65] Wazwaz A, Salmi J, Hallak H, Bes R. Solar thermal performance of a nickel-pigmented aluminium oxide selective absorber. Renewable Energy 2002;27: 277–92.
[66] RÖNnelid M, Karlsson B. The use of corrugated booster reflectros for solar collector fields. Solar Energy 1999;65:343–51.
[67] Tanaka H. Solar thermal collector augmented by flat plate booster reflector: optimum inclination of collector and reflector. Applied Energy 2011;88: 1395–404.
[68] Kostić LT, Pavlović ZT. Optimal position of flat plate reflectors of solar thermal collector. Energy and Buildings 2012;45:161–8.
[69] 〈http://www.solarpanelsplus.com/evacuated-tube-collectors/〉.
[70] Hossain MS, Saidur R, Fayaz H, Rahim NA, Islam MR, Ahamed JU, et al. Review on solar water heater collector and thermal energy performance of circulating pipe. Renewable and Sustainable Energy Reviews 2011;15: 3801–12.
[71] Kim Y, Seo T. Thermal performances comparisons of the glass evacuated tube solar collectors with shapes of absorber tube. Renewable Energy 2007;32:772–95.
[72] Perez R, Seals R, Anderson J, Menicucci D. Calculating solar radiation received by tubular collectors. Journal of Solar Energy Engineering 1995;117:341–4.
[73] Shah LJ, Furbo S. Vertical evacuated tubular-collectors utilizing solar radiation from all directions. Applied Energy 2004;78:371–95.
[74] Morrison GL, Budihardjo I, Behnia M, Water-in-glass evacuated tube solar water heaters. In: Saman W.Y. Charters W.W.S. (Eds), Proceedings of ISES Solar World Congress, Adelaide, Australia, p. 545–550. (Article).
[75] Liang R, Ma L, Zhang J, Zhao D. Theoretical and experimental investigation of the filled-type evacuated tube solar collector with U tube. Solar Energy 2011;85:1735–44.
[76] Rittidech S, Donmaung A, Kumsombut K. Experimental study of the performance of a circular tube solar collector with closed-loop oscillating heat-pipe with check valve (CLOHP/CV). Renewable Energy 2009;34:2234–8.
[77] Diaz G, 2008, Performance analysis and design optimization of a mini-channel evacuated tube solar collector, In Proceedings of ASME IMECE, October 31-November 6, 2008. Boston, Massachusetts, USA. (Article).
[78] Ma L, Lu Z, Zhang J, Liang R. Thermal performance analysis of the glass evacuated tube solar collector with U-tube. Building and Environment 2010;45:1959–67.
[79] Kaiyan H, Hongfei Z, Tao T. A novel multiple curved surfaces compound concentrator. Solar Energy 2011;85:523–9.
[80] Rabl A, O'Gallagher J, Winston R. Design and test of non-evacuated solar collectors with compound parabolic concentrators. Solar Energy 1980;25: 335–51.
[81] Oommen R, Jayaraman S. Development and performance analysis of compound parabolic solar concentrators with reduced gap losses—'V' groove reflector. Renewable Energy 2002;27:259–75.
[82] Winston R. Ideal flux concentrators with reflector gaps. Applied Optics 1978;17:1668–9.
[83] Oommen R, Jayaraman S. Development and performance analysis of compound parabolic solar concentrators with reduced gap losses—oversized reflector. Energy Conversion and Management 2001;42:1379–99.
[84] Rönnelid M, Karlsson B. Experimental investigation of heat losses from low-concentrating non-imaging concentrators. Solar Energy 1996;57:93–109.
[85] Rabl A. Comparison of solar concentrators. Solar Energy. 1976;18:93–111.
[86] Tripanagnostopoulos Y, Yianoulis P, Papaefthimiou S, Zafeiratos S. CPC Solar collectors with flat bifacial absorbers. Solar Energy 2000;69:191–203.
[87] Rönnelid M, Perers B, Karlsson B. Construction and testing of a large-area CPC-collector and comparison with a flat plate collector. Solar Energy 1996;57:177–84.
[88] Eames PC, Norton B. Thermal and optical consequences of the introduction of baffles into compound parabolic concentrating solar energy collector cavities. Solar Energy 1995;55:139–50.
[89] Altuntop N, Arslan M, Ozceyhan V, Kanoglu M. Effect of obstacles on thermal stratification in hot water storage tanks. Applied Thermal Engineering 2005;25:2285–98.
[90] Rhee J, Campbell A, Mariadass A, Morhous B. Temperature stratification from thermal diodes in solar hot water storage tank. Solar Energy 2010;84: 507–11.

[91] Dehghan AA, Barzegar A. Thermal performance behavior of a domestic hot water solar storage tank during consumption operation. Energy Conversion and Management 2011;52:468–76.

[92] Andersen E, Furbo S, Fan J. Multilayer fabric stratification pipes for solar tanks. Solar Energy 2007;81:1219–26.

[93] Shah LJ, Andersen E, Furbo S. Theoretical and experimental investigations of inlet stratifiers for solar storage tanks. Applied Thermal Engineering 2005;25:2086–99.

[94] Furbo S, Andersen E, Thür A, Jivan Shah L, Dyhr Andersen K. Performance improvement by discharge from different levels in solar storage tanks. Solar Energy 2005;79:431–9.

[95] Han YM, Wang RZ, Dai YJ. Thermal stratification within the water tank. Renewable and Sustainable Energy Reviews 2009;13:1014–26.

[96] Shah LJ, Furbo S. Entrance effects in solar storage tanks. Solar Energy 2003;75:337–48.

[97] Dehghan AA, Hosni MH, Shiryazdi SH. Experimental evaluation of the thermal behavior of a vertical solar tank using energy and exergy analysis. Proceeding of IMECE2005 2005 Florida, USA. (Article).

[98] Barzegar A, Dehghan AA. Transient thermal behavior of a vertical solar storage tank with a mantle heat exchanger during no-flow operation. Journal of Applied Fluid Mechanics 2009;2(1):55–69 (Article).

[99] Alizadeh S. An experimental and numerical study of thermal stratification in a horizontal cylindrical solar storage tank. Solar Energy 1999;66:409–21.

[100] Helwa NH, Mobarak AM, El-Sallak MS, El-Ghetany HH. Effect of hot-water consumption on temperature distribution in a horizontal solar water storage tank. Applied Energy 1995;52:185–97.

[101] Hasan A. Thermosyphon solar water heaters: effect of storage tank volume and configuration on efficiency. Energy Conversion and Management 1997;38:847–54.

[102] Farrington RB, Bingham CE. Testing and analysis of load-side immersed heat exchangers for solar domestic hot water systems, technical report. Golden, CO, USA: Solar Energy Research Inst: October 1, 1987. (Article).

[103] Spur R, Fiala D, Nevrala D, Probert D. Performances of modern domestic hot-water stores. Applied Energy 2006;83:893–910.

[104] James & James. Planning and installing solar thermal systems: a guide for installers, architects and engineers; 2005.

[105] Parent MG, Van der Meer TH. Hollands KGT. Natural convection heat exchangers in solar water heating systems: theory and experiment. Solar Energy 1990;45:43–52.

[106] Buenconsejo Jr NS. Experimental study of annular free convection heat exchangers: the flow characteristics in the annular gap. Australia: School of Mechanical and Manufacturing Engineering, University of New South Wales; 1994 (Article).

[107] Nasr A, Morrison GL, Behnia M, Computational study of flow and heat transfer characteristics of annular and vertical cavities, In: Proceedings of 2nd CASCM' 97; 1997. (Article).

[108] Morrison GL, Nasr A, Behnia M, Rosengarten G. Analysis of horizontal mantle heat exchangers in solar water heating systems. Solar Energy 1998;64:19–31.

[109] Baur JM, Klein SA, Beckman WA. Simulation of water tanks with mantle heat exchangers. In, Proceedings of ASES annual conference, Solar: 93;1993.p.286–91. (Article).

[110] Shah LS, Furbo S (1996),Optimization of mantle tanks for low flow solar heating systems, EuroSun 96, Special Issues S-9601. (Article).

[111] Knudsen S, Morrison GL, Behnia M, Furbo S. Analysis of the flow structure and heat transfer in a vertical mantle heat exchanger. Solar Energy 2005;78:281–9.

[112] Shi J, Wang JD, Yu GZ. Chemical engineering handbook. Beijing: Chemical Engineering Press; 1996.

[113] Wang X, Wang R, Wu J. Experimental investigation of a new-style double-tube heat exchanger for heating crude oil using solar hot water. Applied Thermal Engineering 2005;25:1753–63.

[114] 〈http://energy.gov/energysaver/articles/heat-transfer-fluids-solar-water-heating-systems〉.

[115] O' Dell MP, Mitchell JW, Beckman WA, Design method and performance of heat pump with refrigerant-filled solar collectors.

[116] Soin RS, Raghuraman S, Murali V. Two-phase water heater: model and long term performance. Solar Energy 1987;38:105–12.

[117] Soin RS, Rao KS, Rao DP, Rao KS. Performance of flat plate solar collector with fluid undergoing phase chance. Solar Energy 1979;23:69–73.

[118] Radhwan AM, Zaki GM. Analysis of refrigerant-charged solar collectors with phase change. Heat Recovery Systems and CHP 1993;13:429–39.

[119] Farrington R, Deangelis M, Morrison L and Dougherty D, Performance evaluation of a refrigerant-charged thermosyphon solar DHW system. In Proc. AS/ISES annual meeting, Philadelphia, published by the ISES, American Section 1981; 4: p. 676–680.

[120] Joudi KA, Al-tabbakh AA. Computer simulation of a two phase thermosyphon solar domestic hot water heating system. Energy Conversion and Management 1999;40:775–93.

[121] Esen M, Esen H. Experimental investigation of a two-phase closed thermosyphon solar water heater. Solar Energy 2005;79:459–68.

[122] Fanney AH, Terlizzi CP. Testing of refrigerant-charged solar domestic hot water systems. Solar Energy 1985;35:353–66.

[123] Akyurt M. AWSWAH - the heat pipe solar water heater. Journal of Engineering and Applied Sciences 1986;3:1/2:23–38 ; (United States);.

[124] Kamal W. Performance evaluation of a refrigerant-charged solar collector.In energex, 84. Regina, Saskatchewan: Pergamon Press; 1984 193-197. (Article).

[125] 〈http://www.heatpumpcentre.org/en/aboutheatpumps/heatpumpworkingfluids/Sidor/default.aspx〉.

[126] Payakaruk T, Terdtoon P, Ritthidech S. Correlations to predict heat transfer characteristics of an inclined closed two-phase thermosyphon at normal operating conditions. Applied Thermal Engineering 2000;20:781–90.

[127] Kim M-H. Performance evaluation of R-22 alternative mixtures in a breadboard heat pump with pure cross-flow condenser and counter-flow evaporator. Energy 2002;27:167–81.

[128] Chaichana C, Aye L, Charters WWS. Natural working fluids for solar-boosted heat pumps. International Journal of Refrigeration 2003;26:637–43.

[129] Charters WWS, Megler VR, Urma I, Aye Lu. Propane as working fluid is domestic heat pumps. In: IIR/IIF-Melbourne 1996, Refrigeration, Climate Control and Energy Conversation. 1996 February 11-14. Melbourne, Australia: Melbourne: International Institute of Refrigeration; 1996.

[130] Urma I, Propane as a replacement for domestic application, MEngSc thesis, Department of Mechanical and Manufacturing Engineering, The University of Melbourne; 1997. (Article).

[131] Stoecker WF. Comparison of ammonia with other refrigerants for districts cooling plant chillers. ASHRAE Transactions 1994;100(2):1126–35 (Article).

[132] Kim M-H, Pettersen J, Bullard CW. Fundamental process and system design issues in CO_2 vapor compression systems. Progress in Energy and Combustion Science 2004;30:119–74.

[133] Nekså P, Rekstad H, Zakeri GR, Schiefloe PA. CO_2-heat pump water heater: characteristics, system design and experimental results. International Journal of Refrigeration 1998;21:172–9.

[134] White SD, Yarrall MG, Cleland DJ, Hedley RA. Modelling the performance of a transcritical CO_2 heat pump for high temperature heating. International Journal of Refrigeration 2002;25:479–86.

[135] Cecchinato L, Corradi M, Fornasieri E, Zamboni L. Carbon dioxide as refrigerant for tap water heat pumps: a comparison with the traditional solution. International Journal of Refrigeration 2005;28:1250–8.

[136] Kim SG, Kim YJ, Lee G, Kim MS. The performance of a transcritical CO_2 cycle with an internal heat exchanger for hot water heating. International Journal of Refrigeration 2005;28:1064–72.

[137] Laipradit P, Tiansuwan J, Kiatsiriroat T, Aye L. Theoretical performance analysis of heat pump water heaters using carbon dioxide as refrigerant. International Journal of Energy Research 2008;32:356–66.

[138] Zhang XR, Yamaguchi H. An experimental study on evacuated tube solar collector using supercritical CO_2. Applied Thermal Engineering 2008;28:1225–33.