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Solar medium-low temperature thermal utilization and effect analysis of boundary condition: A tutorial

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Abstract

© 2020 International Solar Energy Society Solar thermal utilization is an important part of renewable energy applications, and its development and application have received extensive attention. Based on the development status of medium and low temperature solar thermal utilization systems, this paper first introduces the application and performance research on subsystems of the solar system including heat collection, heat storage and heat utilization units. It then introduces the theoretical and experimental development of matching relationships and dynamic performance of the medium and low temperature thermal utilization systems. Lastly, a case study analysis of a solar thermal system performance was carried out under different thermal boundary conditions. The influence of the boundary conditions of the heat storage system and the actual weather conditions on its performance is explored. The results showed that the boundary conditions of constant temperature and the boundary conditions of constant heat flow density have their own advantages. The heat source conditions provided by heat transfer medium are similar to the constant heat flux boundary condition. The work of this paper provided a theoretical basis for the subsequent application and performance study of solar medium and low temperature thermal utilization systems.

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Solar Medium-low Temperature Thermal Utilization and Effect Analysis of Boundary Condition: A Tutorial

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Abstract: Solar thermal utilization is an important part of renewable energy applications, and its development and application have received extensive attention. Based on the development status of medium and low temperature solar thermal utilization systems, this paper first introduces the application and performance research on subsystems of the solar system including heat collection, heat storage and heat utilization units. It then introduces the theoretical and experimental development of matching relationships and dynamic performance of the medium and low temperature thermal utilization systems. Lastly, a case study analysis of a solar thermal system performance was carried out under different thermal boundary conditions. The influence of the boundary conditions of the heat storage system and the actual weather conditions on its performance is explored. The results showed that the boundary conditions of constant temperature and the boundary conditions of constant heat flow density have their own advantages. The heat source conditions provided by heat transfer medium are similar to the constant heat flux boundary condition. The work of this paper provided a theoretical basis for the subsequent application and performance study of solar medium and low temperature thermal utilization systems.

Key words: medium-low temperature thermal utilization; solar heat for industrial processes; solar heat collection system; heat storage system; thermal boundary condition

1 Introduction

With the increasingly sharp energy competition around the world, the development of renewable energy is regarded as the core task of the Fourth Scientific and Technological Revolution, which can promote the transformation of world energy structure and even economic structure. According to the '2018 BP Statistical Review of World Energy [1], the world energy market continued to transition to a low-carbon economy in 2017, and the electricity generation capacity of renewable energy increased by 17%. This is the largest increase on record, among which solar energy contributed one third of this increase. The policy support and low cost of solar power (5 cents per kilowatt hour) help worldwide solar capacity grow by 100 gigawatts. China remains the world's largest energy consumer, accounting for 23.2% of global energy consumption and 33.6% of the growth in global energy consumption. Among non-fossil energy sources, solar energy

consumption in China was ranked as the fastest growing (76%). In addition to the explosive growth of solar photovoltaic power generation, the installed capacity of solar thermal power generation will also reach to 5 GW during the period of the 13th Five-year Plan proposed by the Chinese government [2].

As solar thermal power generation technology becomes increasingly mature and widespread, the application potential of concentrated solar thermal utilization in other fields, however, is still rarely explored, especially in the field of industrial processes [3]. The total amount of industrial thermal load is huge, which consumed at least 15% to 30% of the total social energy consumption in most industrialized countries around the world [4, 5]. Taking China as an example, the total energy consumed by industrial production currently accounts for 60-70% of the total energy consumption of the whole country. The proportion was 70.4% in 2017, of which 50-70% was used for industrial processing heating, presenting a substantial energy saving potential [6]. The proportion of terminal energy consumption in different regions is shown in Figure 1 [7].

Fig. 1. Share of the total final energy consumption by region [3].

Currently, the heat load of most industrial process is generally less than 250 °C, with pressure under 8 atm. For example, a significant amount of heat is required in the process of the silk of tobacco industry under the temperature of 150-200 °C, drying and heating Chinese herbal medicine or tea products under the temperature of 60-90 °C, and de-enzyme process for Puer tea under the temperature of about 180 °C. Previous studies have shown that the solar energy could be very suitable for heating the carrier to the temperature below 250 °C, therefore, as a potential alternative of fossil energy, solar energy can form a tendency of solar heat for industrial processes (SHIP) [8]. Currently SHIP is in its early stages of development and is seen as the most promising direction for solar thermal utilization in India [9] and in Australia [10]. By the end of 2017, the number of the process thermal systems in the global solar energy industry reached 283, and the installed capacity in 2015 reached 100 MWth. The collector area and investment of the newly added SHIP projects in the last five years are shown in Figure 2. We can see that the newly added heat collector area was the largest in 2017, while the investment cost decreased, indicating that the cost of the system kept falling along with the technological development. The projects were mainly distributed in Mexico (66 projects), India (46 projects), Australia (26 projects) and Germany (26 projects) [11]. In addition to the power generation, the projects in the major developing countries [9] and developed countries

[5,10] were mainly applied in the areas of food processing [12,15], textile [12,14], air-conditioning [13], agricultural and forestry products drying [16], and chemical synthesis and process [17]. However, solar heat for industrial processes in China is growing much slower [7]. The statistical data for the global SHIP map (see Figure 3) shows that the global SHIP utilization is mainly concentrated in the northern hemisphere and low-latitude regions [18].

Solar thermal utilization can be divided into low-temperature thermal utilization (below 80 °C), medium-temperature thermal utilization (80-250 °C) and high-temperature thermal utilization (above 250 °C). The low-temperature thermal utilization is relatively mature, and it is also the most widely used form of application in, such as the solar heating systems [19]. The medium- and high-temperature thermal utilizations require concentrating light devices to achieve a relatively high temperature, and these devices are mainly used in solar concentrating thermal power generation [20]. Processes at low and medium temperatures have a high share of heat demand in the field of light industry [21] such as food & beverage, tobacco, pulp & paper, and also in some heavy industries [22] such as machinery, mining, and transport equipment manufacturing, as shown in Table 1 [23]. In recent years, the SHIP has been regarded to be one of the most potential fields for solar thermal application [24]. 283 operating solar thermal projects for process heat have been reported worldwide, with a total capacity of about 88 MWth (125,000 m²) [11]. The International Energy Agency Solar Heating and Cooling (IEA SHC) Task 49 specially focuses on the solar heat integration in industrial processes, which includes three subtasks: a) process optimization, integration and intensification; b) design guidelines, case studies and dissemination; and c) process heat collector development and testing [25]. This cooperation program provides technical support and popularization for the SHIP systems.

Fig. 2. The global collectors area and investment costs for SHIP systems [11] (The figures in the circle stand for the number of SHIP projects in operation).

The production processes of cleaning and disinfecting cans, kegs and bottles require hot water above 60 °C and low-pressure hot steam above 100 °C. Therefore, cascaded utilization of energy

can be realized, which can preheat boiler feed water and produce steam. The common combinations of SHIP and process technology are to provide direct heat source for process equipment, and to preheat devices such as steam boilers and heating system with indirect heat transfer of heat storage devices [7, 26].

Fig. 3. Map of the globally statistical SHIP applications [18].

Although the SHIP has a great potential for decreasing the fuel consumption, there are also some barriers existed to prevent the development of large-scale SHIP systems. In fact, several SHIP systems have been integrated into the production process. The actual capacity installed so far is relatively small even though a very large potential of harnessing solar energy for meeting the process heating demand in industries has been reported [27]. Several reasons are summarized here includes: it is difficult to integrate solar energy systems with existing or optimized heat flow processes; industrial production systems above the scale are much smaller than small-scale systems, and the latter requires customized solutions; the cost of the fixed asset needs to be paid in advance [27]; the lack of adequately competent designers and design software [28]; the lack of a simple heat storage device [29]; and the lack of adequate policy and regulatory support compared to photovoltaic and concentrated solar thermal (CSP) systems [30]. For low-concentration (solar concentrating ratio below 10) SHIP with the operating temperature range of 60 °C - 250 °C, the application is much less explored in comparison to the low-temperature thermal system, and the degree of attention is not as good as that of solar thermal power generation. The core problems that need to be solved are mainly reflected in two aspects: 1) insufficient market competitiveness resulted from low energy flow density of solar radiation, low economic efficiency of low-concentration light, and short operating life of concentrating devices; and 2) the existence of solar radiation intermittent, periodic and unstable characteristics, vulnerability to the season, day and night. It is generally challenging to solve these core problems to significantly increase its competitiveness [27]. This is contrary to the steady energy demand for industrial heat throughout the year, which requires heat storage or auxiliary heat source to improve solar energy guarantee rate and the system's adjustable flexibility. It is especially important for trough solar thermal power generation systems [30]. Therefore, in order to obtain a wider development of solar industry thermal utilization, it is necessary to tackle the problem of low-concentration concentrating technology and energy storage technology.

Table 1 Application of solar heat for industrial processes [23]

Sector	Process	Temperature Range (°C)	Sector	Process	Temperature Range (°C)
Chemicals	Distillation	100–200	Dairy	Sterilization	100–120
	Compression	105–165		Drying	120–180
	Cooking	80–100		Boiler feed water	60–90
Foods & beverages	Thickening	110–130	Tinned food	Sterilization	110–120
	Blanching	60–100		Cooking	60–90
	Scalding	45–90		Bleaching	60–90
	Evaporating	40–130	Timber By- products	Thermo diffusion beams	80–100
	Cooking	70–120		Drying	60–100
	Pasteurization	60–145		Pre-heating water	60–90
	Smoking	20–85	Wood	Preparation pulp	120–170
	Cleaning	60–90		Fixing	160–180
	Sterilization	100–140		Pressing	80–100
	Tempering	40–80		Steaming	70–90
	Drying	40–200		Compression	120–170
Paper	Bleaching	40–150		Cooking	80–90
	Cooking	110–180		Drying	40–150
	Drying	95–200	Bricks & Blocks	Curing	60–140
Fabricated	Pickling	40–150		Plastics	120–140
				Preparation	
Metal	Degreasing	20–100		Distillation	140–150
	Electroplating	30–95		Separation	200–220
	Phosphating	35–95		Extension	140–160
	Drying	60–200		Drying	180–200
Rubber&	Drying	50–150	Automobile	Blending	120–140
Plastic	Surface treatment	20–120		Water heating	-90
Machinery & Equipment	Cleaning	40–90	Leather	Cleaning	-120
Textiles	Bleaching	40–100		Other processes	-90
	Coloring	40–130	Metal	Heating	-180
	Drying	60–90		Washing	-160
	Washing	50–100	Pharmacy	Different processes	70–180

139 The paper is organized as follows. *Section 1* introduces the background of solar thermal
140 utilization. In *Section 2*, the development of application and performance research of heat collection
141 systems are summarized. *Section 3* presents a review on the development of heat storage systems

including selection of phase change materials, structural design and performance research. In *Section 4*, the performance and capacity matching of the heat collection-storage coupling and heat-collection-storage-utilization systems are summarized. *Section 5* aims at providing a case study of the application of a heat-collection and heat-storage coupling system. This section will also explore the impact of the thermal boundary conditions under the actual weather conditions on the thermal performance of the system. The main findings are summarized in *Section 6*.

2 Development of heat collection system

2.1 Development of solar collectors

Fig. 4. Operation temperature of different types of solar collector for SHIP [13].

The core component of a solar thermal utilization system is the solar collector, which converts the solar radiation into the heat of the heat transfer medium. In the field of low-temperature heat utilization, flat plate collectors (FPC) and vacuum tube collectors (ETC) are mainly used [31]. Pandey et al. [32] found that solar FPCs can be used to trap solar thermal energy and use it for heating applications like water heating, room heating and other industrial applications. Some optical and heat transfer enhancement techniques can extend the temperature to a higher level, such as use nanofluid[33], transparent insulation material for FPCs (up to 150 °C) [34], and the use of inert gas or an ultra-high vacuum (up to 150 °C) [35]. ETC combined with Compound parabolic collectors (CPC) can achieve 3-10 times of concentrated light, and can provide temperatures up to 150 °C [36]. The concentrating collectors are used for medium and high temperature heat utilization, while parabolic trough concentrating (PTC), linear Fresnel concentrating (LFC), non-imaging concentrating (CPC) are commonly used for medium and low concentrating in industrial applications [24], as well as some novel types such as semi-passive tracking collector [13]. Different collectors can provide different heat transfer working temperature zones, as shown in Figure 4 [37]. Kalogirou [38] reviewed and compared FPC, CPC, ETC, PTC, LFC, parabolic dish collector (PDC) and heliostat field collectors (HFC), and simulated the output of SHIP systems for different temperatures. PDC and HFC are two-axis tracking concentrating, which is mainly used for solar

thermal power generation [20]. It was found that in the SHIP field, considering the maturity and economy of technology, PTC is most suitable for heating in the required temperature zone [39].

2.2 Selection of solar collectors

Since concentrated collectors have a great potential in various fields of industrial applications such as mining, food process, chemical synthesis, textiles, machinery and equipment manufacturing, pulp and paper. the collectors used in each case should be selected carefully. Several aspects such as suitable operation temperature, higher efficiency, integration pattern with the industrial process, cost and space availability should be considered.

For industrial processes that highly depend on the temperature range, the solar collector should provide a specific thermal demand with a certain energy grade. According to different temperature ranges, there are mainly several types of collectors used in the SHIP fields, including FPC, ETC, PTC, CPC and LFC. Mahesh et al. [40] presented that the traditional flat plate, evacuated tube and high efficient parabolic trough collector can provide temperature ranges of 60–80°C, 80–150°C and 150–400°C, respectively. For the specific thermal demand, it is necessary to select the collector that can provide a narrow temperature range.

The heat transfer fluid could be water, air or oil. Air is commonly utilized in the food process and drying for agricultural and forestry products [41]. Water is used for heating applications or preheating of boiler feed water, while oil often used for an indirect heat exchange system with a thermal energy storage unit [42]. **The integration patterns mainly include:** (a) Pre-heating of water where heat energy is required for evaporation. However, the condensate can return a high temperature so the efficiency of the non-concentrating collector is reduced [3]. (b) Integration on the distribution level where high temperatures (mostly steam) are required so only concentrating solar collectors can be used. This provides the greatest flexibility as it is not linked to a specific process [43]. (c) Another important way to integrate solar thermal energy is direct coupling with a specific process. **However, when the process is changed or stopped, it can be challenging to use the energy which is rather inflexible [44].** Benoit et al. [45] presented the variations of the convective heat transfer coefficients in the stable operation temperature ranges of the HTFs and discussed recent developments on new HTFs working at least 700 °C.

The design of innovative structures or the selection of more suitable optical materials can ensure a solar collector to operate in a higher efficiency [46], and this will be discussed in Section 3.3. In addition, the economic efficiency was considered. For example, Mohan et al. [47] analyzed three different solar collectors such as FPC, ETC and CPC. It can be concluded that a lowest payback period of 6.75 years was achieved by solar thermal power with evacuated tube collector with a gross area of 216 m².

2.3 Performance of solar collectors

From the theory of **photo-thermal** conversion, it can be seen that the thermal performance of solar energy is related to the type, structure, optical efficiency, working temperature, solar radiation and ambient temperature. The main indicator is the instantaneous collector efficiency [8]. **Various types of experimental set-up [48] and mathematical models [49] of the Parabolic Trough Solar Collectors have been reviewed, and the modelling methods were compared [50].** Zou et al. [51]

proposed a special small-sized parabolic trough solar collector, and extensive experiments were conducted to evaluate the characteristics of the proposed PTC. The results showed that the PTC could collect solar radiation efficiently. Kumaresan et al. [52] carried out experimental and numerical studies of thermal performance enhancement in the receiver part of solar parabolic trough collectors. Conrado et al. [48] presented various types of mathematical models, simulation and numerical methods, and experimental set-up of the PTCs. Based on the performance studies of various types of specific collectors, some empirical formulas were obtained, as shown in Table 2 [53], where η_i is the instant efficiency of solar collector system, T_m is the average temperature of collector, T_i is the inlet temperature of heat transfer fluid, T_a is the temperature of environmental air, and I is the instant solar irradiance. It can be seen that, except for the coefficients, the form of the formula and the parameters involved are almost identical. Therefore, in most cases, it does not require customized design and manufacturing once the performance curve is determined.

To enhance the thermal performance and expand the temperature range, appropriate optimization of structure or materials has been conducted. For the new collectors with a special structure, the main focus is on the optimization of its structure, optics and heat transfer to improve the efficiency [54], and obtain the heat collecting performance through experiments, simulation or advanced prediction methods [55]. Yang et al. [56] present a dynamic 3D volume element model of PTC considering the semi-finite optical model, and analyzed the influence factors on the performance described by the first and second laws in order to optimize the structure and operating parameters. Gong et al. [57] explored the enhanced heat transfer performance of a PTC with pin fin arrays inserting into the absorber tube, by using Monte Carlo ray tracing method. Bellos et al. [58] analyzed the performance of PTC with internally finned absorbers for various operation conditions by using SolidWorks Flow Simulation. For high-temperature collectors, multi-objective performance optimization with non-dominated sorting genetic algorithm has been used [59]. A new design of a collector with a spherical shaped reflector and the tracking components was also reported, and its overall efficiency can reach to 50% and the temperature can go up to 200 °C [60]. A LFC with two-axes tracking and cavity receiver was developed and its temperature can reach 250 °C [53].

Table 2 Efficiency equation of different types of collectors [53]

Solar collector	Tracking	Operating temperature (°C)	Efficiency equation
Flat plate	Non-tracking	30–100	$\eta_i = 0.732 - 3.411*(T_m - T_a)/I - 0.294*(T_i - T_a)^2/I$
Evacuated tube	Non-tracking	50–100	$\eta_i = 0.62 - 3.395*(T_m - T_a)/I - 0.002*(T_i - T_a)^2/I$
Compound parabolic concentrator	Non-tracking	60–240	$\eta_i = 0.644 - 0.749*(T_m - T_a)/I - 0.005*(T_m - T_a)^2/I$
Parabolic trough	Single axis	60–300	$\eta_i = 0.71 - 0.3581*(T_m - T_a)/I - 0.0019*(T_m - T_a)^2/I$
Arun 160 (Dish)	Double axis	up to 250	$\eta_i = 0.65 - 0.40*(T_m - T_a)/I$
Linear Fresnel reflector	Single axis	up to 300	$\eta_i = 0.635 - 0.0265*(T_m - T_a)/I - 0.00043*(T_m - T_a)^2/I$

2.4 Research gaps

Solar heating shows a huge potential for industrial processes, while it has not been applied much in the engineering practice. Even if there are commercially customized solar collectors, the coupling characteristics of the heat collecting unit and other units in the system is still an unsolved problem. Much progress has been made on single heat collecting unit, but limited research has been carried out to match heat storage, heat mode and their coupling characteristics. It is necessary to analyze the coupling relationship between each subsystem and exploit the heat transfer and flow laws in line with the actual situation.

3 Development of heat storage system

3.1 Development of heat storage devices

Thermal storage technology (TES) can alleviate the conflict between thermal energy supply and the demand in terms of time, intensity and space. Common short-term heat storage includes sensible heat storage, phase change heat storage, and chemical reaction heat storage (reversible endothermic/exothermic reaction) [61]. Among them, sensible heat storage is the most mature and commercially viable heat storage method at present. Phase change heat storage is a hot topic in the recent decades, such as the PCMs and applications [62], the heat transfer problems [63] and some researches concerning the PCMs special for solar heating system [64]. Liu et al. [65] reviewed the development in high temperature TES over the past decade with a focus on sensible and latent heat storage and discussed high temperature corrosion and economic aspects of these systems. Miró et al. [66] found that the on-site TES systems have received the most attention in the manufacturing industry.

The sensible heat storage is divided into three types: liquid sensible heat storage, solid sensible heat storage, and liquid-solid combined sensible heat storage [67]. HTF is the same as heat storage material called direct heat storage. It is simpler and more efficient because it reduces one heat exchange step and avoids poor heat exchange between HTF and heat storage material. Molten salt, mineral oil and synthetic oil can be used as the working fluid. In addition, indirect heat storage needs to comprehensively consider the cost of antifreeze and heat storage materials. These two schemes were applied to the Solar Energy Generating Systems (SEGS) trough system in the United States for on-site experiments. The indirect heat storage showed the potential for later development [68]. The application of high temperature conditions created conditions for the Rankine cycle efficiency to increase to 40% [69]. Considering the structure, function, cost and technical maturity of the system, direct heat storage has a great potential in the field of SHIP applications, and heat transfer/heat storage materials such as mineral oil is very suitable for temperature below 300 °C. The direct heat storage usually takes a double tank system; one hot tank is used for melting while one cold tank is used for solidification [70]. However, sometimes the heat storage takes a single tank system, where the middle oblique temperature layer separates the hot and cold working medium into upper and lower parts [71]. The single tank system saves initial investment and is designed to replace the two tank system. Kim et al. [72] presented single tank-based thermal energy storage to reduce costs and installation space. The single tank used a porous block to make easy passage of the molten salt and reduce the flow pressure drop. Zhang et al. [73] explored the performance of a solar and coal dual-source boiler with a thermocline energy storage, and found that the energy storage efficiency improved by 0.5-0.8% per hour.

Phase change heat storage has the advantages of large heat storage density, easy process control,

stable temperature during heat storage/heat release process, etc. Extensive research has been performed, especially for applications where heat supply is discontinuous or supply and demand are not coordinated. It also has advantages in concentrating solar thermal power generation systems. The current research focuses on the development of new phase change materials (PCMs) and characterization of their physical parameters, thermal stability, and thermal conductivity enhancement, the structure of thermal storage equipment, and PCMs and equipment materials compatibility.

Some investigations are related to the latent heat storage (LHS) and sensible heat storage (SHS). The low-temperature solar utilization of a heating water system for daily applications was explored [74]. It is found that the latent heat storage with paraffin wax as PCM can store 40% more heat energy than the sensible heat storage. Fan et al. [75] conducted a review covered a variety of PCMs, operating conditions, heat exchange and thermal energy storage arrangements. Alva et al. [76] found that the thermal conductivity of the PCM can be improved by dispersing high thermal conductive particles like carbon, graphite or metals into the PCM. Tay et al. [77] conducted that the use of additional metallic material and increasing heat transfer surface area such as fins can improve the heat transfer rate of the PCM. Vasu et al. [78] summarized that the compatibility of the container materials with PCM plays an important role because PCMs are normally encapsulated in containers.

3.2 Selection and thermal properties of PCMs

In the design process of a phase change heat storage system, it is first necessary to select or prepare a suitable phase change material. In addition to considering the thermal conductivity of its key physical parameters, the selection and preparation of phase change energy storage materials in combination with a medium temperature collector system must meet several physical and chemical properties [63]. Wei et al. [79] found that the selection of phase change materials for TES systems should meet thermal performance, physical performance, dynamic performance, chemical performance, economic performance, and technical performance requirements. Molten salt is the first choice for the field of high-temperature heat utilization in solar energy. It is usually composed of alkali metal or alkaline earth metal and halide, SO_4^{2-} , CO_3^{2-} , NO_3^- and PO_4^{3-} . In contrast, the alkali metal nitrate system can better meet the above requirements, and be particularly suitable for medium and high temperature heat storage media [80]. Zhang et al. [81] adopted the eutectic molten salt as the PCM to make the utilization of solar energy at medium temperature of around 200 °C more efficiently, and it will be helpful for the application using molten salt for solar energy storage. Pflieger et al. [82] and Jin et al. [83] revealed that Hitec (NaNO_3 - NaNO_2 - KNO_3) and Solar salt (NaNO_3 - KNO_3) are the two molten salts that have been extensively used in Concentrated Solar Power (CSP). At the same time, the viscosity of the specific components of emerging ternary nitrates ($\text{Ca}(\text{NO}_3)_2$ - NaNO_3 - KNO_3 , LiNO_3 - NaNO_3 - KNO_3 and $\text{Ca}(\text{NO}_3)_2$ - LiNO_3 - KNO_3 mixtures) were measured in the high temperature areas, which have shown their potential applications in CSP systems.

In addition, in order to make molten salt act as a flow heat transfer medium, new formulations or additives that have lower melting points and a wider temperature range of use have been developed. Raade [84], for instance, developed a eutectic mixed nitrate of five components with a melting point reduced to 65 °C. Tian et al. [85] prepared the physical performance enhanced molten salt by adding Mg particles. Zhou [86, 87] carried out a study of a eutectic salt of LiNO_3 - NaCl , and

found that it was very suitable for the medium temperature range because the chemical stability was below 400 °C. Cunha et al. [88] presented a comprehensive review of PCMs with phase transition temperatures between 0 and 250 °C. Organic compounds and salt hydrates seem to be more promising below 100°C and eutectic mixtures are more suitable for applications where temperature varied from 100 to 250 °C. Bradshaw et al. [89] disclosed a series of patents for low-melting mixed molten salt heat transfer and heat storage media. For industrial thermal loads at temperatures below 250 °C, the phase transition temperatures of these molten nitrates are suitable.

Faced by a wide variety of phase change heat storage materials (at least 160,000), it is a very complicated task to identify the most suitable heat storage material for a particular application. It is also a key issue to promote the high-temperature heat utilization of solar energy. In the field of solar thermal power generation, due to the production needs, the impact of the selection of heat storage materials on system performance and cost have been analyzed from an operational perspective. Ren [90] prepared 36 kinds of mixed carbonate molten salts, and analyzed their thermal physical properties, as well as the cost. Thirteen types of ternary carbonate salts were recommended for latent heat storage. Bradshaw et al. [91] evaluated the use of various binary nitrates as single phase heat transfer fluids for solar tower absorbers and analyzed the effects of the chemical and physical properties of molten salts on thermoelectric systems. However, this experimental study was directed to the case where a liquid molten salt was used as a heat transfer heat storage medium, and did not include the physical properties in the solid state. Ruegamer et al. [92] compared the investment cost and the levelized cost of energy of different heat storage materials and heat storage systems with different heat storage scales in the trough concentrating power generation system. Jacob et al. [93] investigated the capital cost (CAPEX) of each system including the tank, storage material, encapsulation cost (if applicable) and allowances for construction and engineering. The liquid sodium system was found to result in a higher embodied energy and CAPEX than other previously studied TES systems.

When a material is actually selected, it is difficult to ensure that the parameters of the thermal performance are dominant. For example, a material with a large thermal conductivity may have a small specific heat capacity. Fernandez et al. [94] used the CES database to evaluate and optimize materials for the 150-200 °C sensible heat storage based on the minimum cost principle. However, this method is only suitable for the material selection with well-defined analytical solutions. However, for the latent heat storage process including phase transition and phase interface migration, the applicability of this method is poor, and the numerical model is usually used to simulate the heat transfer process. Pointner et al. [95] discussed four different numerical models that can be used to investigate phase change processes within latent heat storage and compared their performance in terms of accuracy, convergence behavior and computational efficiency.

3.3 PCM thermal conductivity enhancement and structure optimization of LHS system

Most of the available PCMs have relatively low thermal conductivity. The thermal conductivity of inorganic PCMs such as hydrated salts and molten salts is only about 0.5 W/m·K, which directly affects the heat storage and heat release rates of the PCMs [96]. Low thermal conductivity is one of the most important factors restricting the large-scale application of phase change heat storage. Therefore, strengthening the physical parameters such as PCM thermal conductivity and enhancing the heat transfer of the heat storage devices have always been the focus of research in the field of

phase change heat storage. At present, there are several ways to enhance phase change heat transfer, such as improving the aspect ratio of heat storage materials, filling materials with high thermal conductivity, microencapsulation packaging technology, and combining with heat pipe heat pumps.

Al-Maghalseh et al. [97] discussed the heat transfer intensification methods including application of fins, filling materials, nano-fluids, nano-particles, microencapsulation and the thermal conductivity enhancement method. Nomura et al. [98] reported the synthesis of a core-shell type microencapsulated PCM (MEPCM) consisting of Al-25 wt% Si microspheres (mean diameter of 36.3 μm and melting temperature of 577 $^{\circ}\text{C}$) as the core (PCM) and Al_2O_3 as the shell. The results showed that the heat capacity measured for this material was five times higher than that of conventional solid SHS materials. Zhang et al. [99] developed a novel microencapsulated PCM by in situ polymerization to produce high encapsulation capacity and leakage-prevention performance. Li et al. [100] established a new high-temperature packed-bed thermal energy storage system (PBTES) with macro-encapsulation of molten salt PCM.

Increasing the aspect ratio of the heat storage material and filling the material with high thermal conductivity are common strengthening methods. Fixed high thermal conductivity inserts, adding additives, introducing metal skeletons or adding honeycombs, fins, brushes are among the main methods used to enhance the thermal conductivity of PCMs [75, 101], especially the addition of carbon fiber [102-104], metal foam or powder [105-106], preparation of micro-nano composite [107-110] or EG [111-113]. For example, Fukai et al. [103] studied the effect of random carbon fiber and carbon fiber brush on the thermal conductivity of paraffin, and the results showed the thermal conductivity of the mixture can be improved significantly. Wang et al. [112] proposed a phase-change composite of double-layer carbon grid structure. The thermal conductivity of the original paraffin wax increased to 7.5 times when the EG content of expanded graphite reached 20%. Xiao et al. [111] analyzed the melting and solidification processes before and after the addition of metal foam in molten salt, and found that increasing foamed copper and foamed nickel reduced the solidification time by 28.8% and 19.3%, respectively. Das et al. carried out a numerical simulation and investigated the melting process of organosilane PCMs both in horizontal [109] and vertical [110] shell-and-tube LHS devices. The results showed that the addition of nanocomposites increased the thermal conductivity of the PCM to a certain extent. Mixing 1% by volume (vol%) of different morphological nanomaterials can increase the melting rate by 2% with spherical nanoparticles, increase the melting rate by 27% and 40% with single-walled carbon nanotubes and graphite nanosheets respectively, and the corresponding melting time reduced by approximately 15% and 25%. Ibrahim et al. [114] revealed that heat transfer enhancement in latent heat thermal energy storage (LHTES) systems can be achieved through either geometric configuration and/or thermal conductivity enhancement. The use of extended surfaces such as fins or heat pipes is a common technique for heat transfer enhancement. Cunha et al. [88] explored the heat enhancement methods including extended metal surfaces, using carbon, metal matrices and conductive powders, and direct heat transfer techniques. Merlin et al. [115] presented an experimental thermal energy storage using a phase change material associated with various configurations of conductive structures such as finned exchangers, graphite powder and Expanded Natural Graphite (ENG) matrix. The results showed that the thermal conductivity of the ENG/PCM composite was about 100 times greater than the conductivity of the PCM alone. Chang et al. [116] introduced concentric rod and eccentric rod as turbulators, and analyzed the flow and convective heat transfer characteristics of molten salt in a

parabolic trough receiver. The result showed that both concentric rod insert and eccentric rod insert can enhance the heat transfer performance effectively. Sandeep et al. [117] examined the feasibility of various heat transfer augmentation techniques for a PTC receiver. It was concluded that for PTC application, using insert with base fluid was beneficial in the laminar region and nanofluid with insert was justified for the turbulent regime. Tao et al. [96] introduced that the available enhancement methods can be classified into three categories, including using high thermal conductivity additives and porous media to enhance PCM thermal conductivity; using finned tubes and encapsulated PCMs to extend heat transfer surface; and using multistage or cascaded LHS technique and thermodynamic optimization to improve the heat transfer uniformity.

Except for improving the **PCM itself, structure optimization of the latent heat** storage device is also important to improve heat transfer performance. Ogoh et al. [118] explored the influence of the number and distribution of fins in cylindrical phase change storage tanks on heat storage characteristics. Liu et al [119] optimized the structure of a shell-and-tube heat exchanger with a metal foam and analyzed the structural parameters of the metal foam and the influence of the heat transfer medium inlet temperature on the heat storage characteristics. Al-Abidi et al. [120] conducted a structural reinforcement analysis of the internal fins and outer fins of a three-tube heat storage heat exchanger by using two-dimensional numerical modeling and explored the effect of the heat exchanger material on the heat storage rate for the number, length, thickness, and number of fins. Mat et al. [121] carried out simulation and experimental investigation on a three-tube phase change heat exchanger, and studied the influence of different heating forms and fin arrangement inside and outside. The results showed that the inner and outer fins can greatly reduce the melting time. Pizzolato et al. [122] proposed an optimization method based on topology and CFD to simulate the important role of natural convection in heat transfer enhancement in phase change storage tanks. Yang et al. [123] used simulation to study the role of inner ring fins in shell-and-tube heat storage tanks and optimized the structure and size. Shinde et al [124] conducted an enhanced heat transfer analysis on the phase change heat storage device of the organic Rankine cycle solar thermal power plant and established a three-dimensional simulation model to obtain the optimal number, thickness, and height of the fins. Veeraragavan et al. [125] explored the influence of lateral heat losses from a PCM storage tank on the performance of a storage integrated concentrating solar power system by means of an analytical model. It revealed that a high absorber area enabled the system with a high thermal efficiency.

For the performance of the heat storage device, the heat storage of the direct heat storage is relatively simple to calculate. The theoretical and experimental challenges related to the thermal performance of the double tank and the single tank systems are mainly temperature stratification. Senthil Kumar [126] analyzed the effect of the addition of PCM encapsulated in spherical capsules at the top of a hot water storage tank for stratification enhancements during the charging process. The results showed that PCM capsules increased the stratification capability as the temperature difference between the inlet HTF and PCM melting temperature was increased. Bouhal et al. [127] carried out the thermal stratification inside a standard hot water storage tank. It was shown that thermal stratification inside the storage tank is dependent on the positions of the flat plate. Wang et al. [128] designed a novel equalizer in order to improve the thermal stratification of a heat storage tank. It was shown that the equalizer performed best, and the storage tank had a better thermal stratification when the flow rate was 3 L/min. Moncho-Estevé et al. [129] used Computational Fluid

Dynamics (CFD) to study thermal stratification during the charge process in a cylindrical water tank.

Different phase change heat storage devices and different PCMs have distinct thermal properties. Therefore, the heat transfer performance of the phase change heat storage unit is characterized by the heat storage efficiency and includes the heat storage/release rate index and the exergy efficiency based on the second law of thermodynamics.

3.4 Research gaps

As an obstacle for PCM in SHIP application, its low conductivity is always considered as a foremost factor. Very limited investigations have been performed to address the PCM based heat performance enhancement. While the effects of thermal conductivity and phase-change temperature on storage system performance are generally separately studied, it is worth combining these two factors in future investigations.

4 Matching and performance of heat collecting/storage/utilization coupling systems

4.1 System matching relationship and performance

The static matching of the heat collection-storage-utilization units and the dynamic matching relationship on the heat transfer rate determine the operating performance of the SHIP system. Due to the slow development of the SHIP system, the field of solar thermal power generation has attracted more attention. Therefore, the current research status is mostly in the field of power generation.

In terms of static matching of sub-unit size, Powell et al. [28] established a heat transfer model of a trough solar thermal power generation system from the heat collection, heat storage to the boiler heat transfer. The operation dynamics of the system were respectively analyzed using a simple control strategy and an optimal control strategy. It was found that the solar energy guarantee rate of the system increased from 47% to 70% on sunny days after the increase of 8 h heat storage capacity of a TES. Baghernejad et al. [130] optimized the operation of a hybrid thermal power plant with conventional energy and solar energy, and combined the energy balance and analysis method to optimize the important parameters such as the scale of solar energy field, the rate of return, the consumption of conventional energy, and the length of solar energy operating time. Mohamadi et al. [131] carried out a theoretical modeling and experimental study on a combined solar thermal storage device. The optimized results showed that different heat storage media should be selected with different time period in order to increase heat storage without increasing the heat collection area. Heller et al. [132] established a heat transfer model for a solar thermal power generation system with a thermal storage device in a rock bed with an inclined temperature layer, and simulated the performance of the whole generation and heat storage system. Behar et al. [20] carried out a detailed modeling that comprised the heat transfer in the receiver tube, optics of the collector and the heat losses from the piping for the solar field. Zhao et al. [133] carried out a survey on achieving a higher solar-to-electricity conversion efficiency through solar-fossil hybrid thermal power systems as compared to a solar-only power plant. The results indicate that the moderate-temperature solar and fossil hybridization technology can provide a promising direction for efficient utilization of low-grade solar heat.

In terms of dynamic matching such as operating parameters, Menéndez et al. [134] conducted a mathematical modeling and control study for a trough-type concentrating solar thermal system with a molten salt heat storage device. Zhao et al. [135] established a physical model and a dynamic simulation model of a heating system composed of a plate collector, a phase change heat storage, an auxiliary heat source and a plate heat exchanger. The real-time parameters in 7 operation modes were simulated, which provided a theoretical reference for design optimization. Li et al. [136] investigated the dynamic efficiencies of an open-loop air receiver and a thermal energy storage unit numerically. Meanwhile, an air receiver and thermal energy storage experimental platforms were built to validate the simulation models. Ni et al. [137] studied the influence of intensity of solar radiation on the daytime performance of a solar-ORC system.

For solar thermal systems with heat storage device, the degree of flexibility of the system is increased, the operation mode is more diverse, and the control strategy is more complex. Cole et al. [138] analyzed the thermal storage in the building heating, solar thermal power generation and even the chemical industry to increase the flexibility of the system. Multi-parameter constraint control is required to maximize the benefits of heat storage devices. Khalilzadeh et al. [139] optimized the daily operation of solar thermal power plants with heat storage devices, and established a series of models with nonlinear control equations and verified methods. Taking the maximum daily generation time and maximum benefit as the optimization objectives, the results of two single objectives and one multi-objective optimization were compared. The first single objective optimization showed that the heat storage unit increased the power generation time by 7 hours per day, and the second single objective optimization showed that the heat storage unit provided a 13.5% profit increase. The multi-objective compromise optimization showed that the heat storage unit can increase the power generation time by 5 hours and increase the profit by 8.1%. Arkar et al. [140] presented the design and thermal response modeling of a solar air heating system, which consists of an air vacuum tube solar collector and latent heat storage. The analysis showed that a solar heating fraction of 63% can be achieved in an energy efficient, lightweight building.

4.2 Dynamic simulation of system performance

The establishment of dynamic heat transfer models of solar energy systems can reliably reflect the heat transfer behavior and performance of the system. It is also the basis of system matching and optimization. To obtain the real melting (heat storage) and solidification (heat release) behaviors, it is required to explore the influence of the solar energy collecting field on the heat storage and examine the influence of the terminal heat on the heat release. Theoretical modeling, numerical simulation and experimental investigation are often used to pursue this goal. Here the heat storage process is taken as an example to illustrate the current research progress. As the melting and solidification processes of PCMs is a complex heat transfer process involving two phases, there are generally three physical phenomena in the process of phase transition: phase transition, heat conduction and convection. Generally, the heat transfer in the liquid region is dominated by convection, and the solid area is dominated by heat conduction, making it more complex than a single conduction or convection process. Also, the solid-liquid phase interface is a nonlinear motion process, which is difficult to obtain precise analytical solutions. The exact solution can be obtained, such as the representative Neumann solution [141] if only considering the first type of thermal boundary conditions for a single component PCM within a one-dimensional infinite region. For

finite boundaries and multicomponent PCMs, considering the process of natural convection, or the second or third type of thermal boundary conditions, the approximate analytical solutions can only be obtained by means of isotherm transfer, heat balance integral and periodic solution. However, the approximate solution is too complex to be used conveniently in practical problems.

Since it is difficult to predict the mathematical analytic solution and the approximate analysis solution, in many cases, only numerical simulations were used. Many studies adopted numerical simulations to examine the heat transfer characteristics of phase change materials, which has gradually become an efficient and useful research method in the field of phase change energy storage. Numerical simulation methods such as finite difference method and finite element method are more convenient and effective in solving moving boundary problems. Software simulation has become a conventional method for optimizing heat storage device. For example, Yang et al. [123] used software simulation to study the effect of annular fins in a tubular heat storage tank, and optimized its structure and size. Pizzolato et al. [122] proposed an optimization method based on topology and CFD to analyze the important role of natural convection in heat transfer enhancement in a phase-change heat storage tank. Hachicha et al. [142] used a numerical-geometrical method based on the ray trace and finite volume method to determine the solar flux distribution around the absorber tube with a high accuracy. Calise et al. [143] investigated an innovative solar-geothermal polygeneration system and developed a dynamic simulation model in TRNSYS. Allouche et al. [144] carried out a simulation study using TRNSYS to meet the cooling needs of a 140 m³ space during the summer season in Tunis, Tunisia. It was found that the application of a relatively small hot storage tank led to the highest solar fraction (92%). Fornarelli et al. [145] examined a latent heat storage system for concentrated solar plants (CSP) by means of CFD simulations. The results revealed that the enhanced heat flux, due to natural convective flow, reduced about 30% the time needed to charge the heat storage.

Numerical simulations and corresponding verification tests are usually performed under the first type of thermal boundary conditions at constant wall temperature. However, there are only a few studies that combined solar energy collection unit with phase change heat storage unit and these studies mainly focused on the low temperature heat utilization of solar energy, especially the household solar water heating systems. A number of studies also focused on the solar thermal power systems in the field of high temperature utilization of solar energy. Due to the slow process of medium-high temperature heating and cooling, the cost of one-time investment for the experiment platform is also high. In order to save the cost of experiments and shorten the research period, numerical analysis methods are usually adopted. Shinde et al. [124] analyzed the enhanced heat transfer of the phase change heat storage device of an organic Rankine cycle solar thermal power plant, and established a 3D simulation model in order to obtain the optimal number, thickness and height of fins. Nithyanandam et al. [146] proposed a rigorous calculation model for a phase-change heat storage device for solar concentrated power generation. The dynamic heat storage reaction under the thermal boundary condition of variable inlet temperatures was then simulated. Although this paper mainly focuses on the matching relation between the heat collection and heat storage units and the dynamic characteristics of the whole system, it also provided a theoretical reference for numerical modeling under variable temperatures and thermal boundary conditions.

4.3 Research gaps

To analyze the matching relationship and dynamic characteristics of the heat transfer in the PCM-based storage system, it is inevitable to deal with the motional solid-liquid boundary problem even though difficult to solve. Because the state parameters such as liquid fraction are time-varying and can be obtained via simulation, it is possible to solve the motional solid-liquid boundary by combining experiment evaluation with numerical simulation. The key issue to build a such a hybrid model is to establish the mapping relationship between the phase change process, boundary condition and operation time. It is also necessary to establish a dynamic model to study the heat transfer characteristics of the heat-collecting and heat-storing coupled system, as well as the heat-releasing and heat -utilization coupled system.

5 A case study: performance of a system coupled heat collecting with heat storage

5.1 Boundary condition of heat transfer

According to the basic theory of heat transfer, there are three thermal boundary conditions, among which the first thermal boundary condition is the temperature boundary, in which the heat temperature of the heat material is constant. The second boundary condition is the boundary of heat flux, in which the heat flux is constant. The two boundary conditions are described in Eqs. (1-2).

$$t_w = C_1 \quad (1)$$

$$q_w = -\lambda \left(\frac{\partial T}{\partial n} \right)_w = C_2 \quad (2)$$

Where t_w is the wall temperature, C_1 and C_2 are the constants, n is the normal of the area boundary, q_w is the heat flux density in the outer wall and λ is the material thermal conductivity.

For the heat exchanger to storage heat, the latent heat and the sensible heat are combined and Eq.(3) is commonly used. K is the total heat transfer coefficient, where the corresponding thermal resistance is combined as R_1 , R_2 and R_3 , as shown in Fig. (4). Because the thermal conductivity of the PCM is far lower than the metallic thermal conductivity, R_2 and R_3 are far lower than R_1 and can be ignored. R_1 is the combination of the heating area thermal conductivity and the phase boundary thermal conductivity.

$$Q = KA \cdot \Delta t_m \quad (3)$$

$$K = 1 / (R_1 + R_2 + R_3) \quad (4)$$

where Q is the total heat transfer capacity, A is the heat transfer area, Δt_m is the logarithmic mean temperature difference, R_1 is the heat resistance of solid-liquid PCM, R_2 is the heat resistance of the tube wall, and R_3 is the convection heat resistance between the HTF and the tubes.

5.2 Effect of heat boundary conditions on heat transfer characteristic

5.2.1 The first boundary condition

A numerical simulation study was carried out for a specific heat collecting-storage-utilization system, whose model establishment, setting and reliability verification were all reported in a previous study [147]. The boundary conditions of the latent heat thermal energy storage were respectively set as constant wall temperature $t_w=270, 260, 250$ °C, where the temperature range was

matching with the constant temperature obtained from the actual solar heat collector. The initial temperature of the PCM was 25 °C, the melting and heat transfer processes of the PCM under three heat source temperatures were calculated. The temperature, mass fraction of the liquid phase, and heat transfer flux and other parameters variation during the 360 minutes were obtained and are shown in Figs. 5-7.

Fig. 5. PCM average temperature in the melting process.

Fig. 6. Mass fraction of liquid phase in the melting process.

From Figs. 5-6, it can be seen that the variation trend of the PCM temperature was coincident, and the velocity of the temperature elevation was gradually decreased since the three temperature differences of the wall was only 10 °C. Therefore, the temperatures in the three conditions were very close with each other. However, it is not obvious that the temperature difference occurred is because the PCM temperature nearly reaches the latent heat temperature before 210 min. The latent heat temperature is relatively steady and the phase changing is the main source to absorb heat in the later period, and hence, the temperature elevation is only a subsidiary. At 360 min, the liquid fraction of the three temperature levels from high to low were 0.696, 0.583 and 0.454, respectively.

The condition of the heat transfer in the present experiment was further investigated. The heat transfer rate of the thermal wall PCM in the heat transfer area with the computational domain of $8.2 \times 10^{-4} \text{m}^2$, is shown in Fig. 7. It can be seen that the heat transfer rate drastically decreased as the reduction of the difference in heat transfer temperature. The heat transfer rate slowly decreased after 90 min, and the difference of the three boundary conditions kept an obvious level. However, after 210 min, the rate in the wall temperature of 270 °C was smaller than that of 260 °C. After 26 minutes,

the heat transfer rate in the wall temperature of 260 °C was slightly lower than the rate in the wall temperature of 250 °C. This indicates again that the difference in heat transfer temperature was the main influencing factor. However, the temperature variation was associated with the height of the liquid fraction, which is a complex process containing sensible heat and latent heat. However, in the 360 min, the total heat transfer of the three temperature levels were 273.51, 222.48 and 204.92 MJ/m², respectively. Taking the liquid fraction rates and total heat transfer into account, in this period, the difference taken by the heat material in the temperatures of 250 °C and 260 °C was much lower, and the difference taken by the heat material in the temperatures of 260 °C and 270 °C was more obvious.

Fig. 7. Heat transfer flux variation from inner wall in the melting process.

On the basis that known conditions are heat transfer area, heat transfer rate and average heat transfer temperature difference, total heat transfer coefficient K can be elicited according to Eq. (3). The variation of K with time is shown in Fig. 8, which can reflect the heat resistance of the PCM to some extent. Visibly, the total heat transfer coefficient decreased slowly at the beginning, and then rose up and drop down for twice. The variation curves with this trend may due to that the temperature of the PCM close to the inner wall rapidly reached the latent heat temperature, while limited liquid molten salt was produced in the outer wall, resulting in solid in the liquid periphery to disable the liquid flowing. Moreover, the thermal conductivity of liquid PCM was 33.33% lower than that of solid PCM, which caused R_1 to increase and K to decrease slightly. As liquid fraction increases, natural convection began to take shape (as shown τ_1), and the mechanism of heat transfer was not only conduction, but the combination of natural convection and conduction. The liquid close to the tube inner wall flowed upward to melt the supernatant solid and move up the phase boundary. Thus, the heat transfer rate began to rise fast. However, after reaching the later inflection point τ_2 , about 40% of PCM has molten. The liquid molten salt mainly distributed in the upper tube, and the solid in the upper tube almost disappeared. Therefore, the writhing effected by density difference of solid and liquid phase was becoming weak. In the process when the liquid circular flowed along the phase boundary, the temperature decreased gradually, the power of melting reduced continually, and the natural convection transferred from upper tube to nether tube section. Subsequently, the main parameter of the liquid PCM reduced the effect on the conduction, leading to the reduction of the total heat transfer. So the main form of this stage is the liquid thermal conduction. The second inflection point τ_2 of K was mainly influenced by the liquid fraction, in the three conditions, the

rising rate of liquid fraction in the 250 °C was most slow, therefore, it's most late to get τ_2 , and the main switching point of two factors is not obvious as when the temperature is 270 °C, when the influence of two factors is almost equal, the curve after reaching τ_2 is gentle. The difference between τ_2 and τ_1 is the grand total time of natural convection, the total time under three temperatures is almost 3 hours.

Previous investigations have shown that natural convection plays an important role in the melting process, therefore, it's necessary to take natural convection into consideration for latent heat systems. It is beneficial to make τ_1 start earlier, and the duration of τ_2 longer. In the condition when the continuing period was constant, τ_1 under the temperature of 270°C was one hour earlier than that of the temperature 250°C. This phenomenon agreed with the qualitative conclusions of classical theories.

Fig. 8. Total heat transfer coefficient K in the melting process.

Through the comparison of the mass fraction of liquid phase, heat flux, and total heat transfer coefficient variation within 360 min, it was found that rising the heat source temperature by 10% can result in an obvious difference that the heat source temperature is 20°C higher than the phase transition temperature. Therefore, it is better to rise the tube wall temperature as much as possible to achieve more effective heat transfer.

5.2.2 The second boundary condition

To compare with the first boundary condition, constant heat flux which has the same total heat transfer of thermal material in the constant wall temperature of 270 °C in 360 min (the period of high radiation in one day) was selected as C_2 in the second boundary (i.e. heat flux $q_w = 13.208 \text{ KW/m}^2$). From numerical simulation, the temperature, liquid fraction and the variation of total heat transfer can be obtained and the results are shown in Figs. 9-12.

Since the total heat transfer in 360 minutes under the first boundary condition was the same as that of the second, the final temperatures of the first and the second were very close. It can be seen that both of the temperatures of HTF and PCM increased tardily in the condition of the constant heat flux, and the temperature difference always stabilized at a small range, and the temperature of

HTF over 270 °C was reached until the final 60 min. The temperature and the accumulated total heat transfer of the PCM in the heat storage process were increased slowly in the constant heat flux. However, in the condition of the constant wall temperature, both temperature and the total heat transfer emerged a trend that is slower growth. For the liquid fraction of the PCM, in the first half the fraction was 0 in the condition of constant heat flux, indicating a process of accumulating sensible heat. When the HTF temperature raised, the tin temperature was higher than that in the condition of constant wall temperature. Apart from 180 min, the PCM began to melt gradually and the natural convection began to form. The natural convection was only 130 minutes, which was far lower than 180 minutes in the condition of the constant wall at 270 °C.

Fig. 9. Average temperature of HTF and PCM in the melting process.

Fig. 10. Mass fraction of liquid phase in melting process.

Fig. 11. Accumulated heat transfer flux in the melting process.

Fig. 12. Total heat transfer coefficient in the melting process.

5.2.3 The boundary condition of actual solar radiation

In the actual production process, the quantity of heat contained in the thermal working substance is the heat storage thermal source. Without ministrant thermal source, the of the heat storage the HTF temperature rises gradually because of the constant wall temperature boundary. Ignoring the temperature difference at the entrance, the HTF temperature at the entrance and exit can be considered as constant; this means that the coupling influence of heat storage to the heat collector entrance temperature can be ignored. Therefore, only the single influence of heat collector on heat storage needs to be considered. The solar radiation of Kunming region in a common day was taken as a heat collector thermal source condition (see in Fig. 13), the temperature variation of the HTF can be calculated under the obtained characteristics of the parabolic trough solar concentrated collector, as shown in Fig.14.

Fig. 13. DNI in Kunming district on May 14, 2016.

It can be seen from Fig. 13 that solar radiation was a parabola going downwards after sunrise and before sunset, and solar radiation rapidly rose to 500 kW/m² at the beginning of sunrise and then decreased slowly. What need to consider is that the solar radiation was not keeping a strict parabola and is influenced by clouds covering.

Based on the previous results in [147], a relationship between the working substance temperature and heat transfer efficiency can be obtained in Fig. 14. According to the relation of energy and temperature rising and set solar radiation as a basis, fitting thermal boundary temperature can be calculated. The longest period of 360 min is set as constant, some parts truncated from working substance temperature are set as thermal boundary conditions in the numerical simulation, and the liner fit boundary formula is $t_w = 115.7 + 0.426\tau$ (°C). HTF and tube inside wall conducted heat by convection, and the tube inside wall temperature was calculated in the condition of convection. Since heat resistant was low, the temperature difference between tube inside wall and HTF was only about 2 °C, which was considered as assumption (3) under the simulation. Fig. 15-17 shows the boundary conditions of the heat storage.

Fig. 14. The relationship between total efficiency η and HTF temperature.

Fig. 15. Average temperature variation of heat resource and PCM in the melting process.

Fig. 16. Mass fraction of liquid phase under actual solar-driven heat source.

It can be seen that the temperature variation of the heat collection material increased in a linear trend. The liquid fraction kept on 0 in the first 210 min and then increased until to 0.62. It can see from Fig.17 that the heat transfer ratio was decreased with small variations.

Fig. 17. Heat transfer flux under actual solar-driven heat source.

5.3 The comparison among the effect of three boundary conditions to thermal performance

By focusing on the two boundary conditions given in above, then simulation results of relative parameters were obtained. According to the parameter defined in the reference [148], the heat storage, heat loss and heat storage efficiency were calculated, and the results are summarized in Table 3.

It can be seen that the heat storage efficiency of the first boundary condition in different periods was decreased. However, when the temperature boundary is discrepant, the heat storage efficiency basically kept stable and the heat was 62.13% of the total heat reserved in 360 min. The temperature of the PCM at this moment was lower than that at the melting point. The total heat transfer of the second boundary condition in 360 min was the same as the first boundary condition at the 270 °C. However, the heat loss was lower in the whole heat transfer process, and the efficiency was relatively higher. The heat storage efficiency in different period slightly decreased. In addition, in the condition of lower temperature, heat storage at the previous stage was little lower, and the heat transfer variation increased significantly as the increase in the temperature of heat storage. However, there was not an equal relationship between the solar radiation condition and the two boundary conditions.

Table 3 Comparison of heat storage efficiency

Heat boundary	$Q_{360\text{min}}$ (MJ)	$L_{360\text{min}}$ (MJ)	$\eta_{360\text{min}}$	$Q_{330\text{min}}$ (MJ)	$L_{330\text{min}}$ (MJ)	$\eta_{330\text{min}}$
$t_w=270^\circ\text{C}$	10.68	2.007	0.842	10.55	1.787	0.855
$t_w=260^\circ\text{C}$	9.92	1.881	0.841	9.745	1.676	0.853
$t_w=250^\circ\text{C}$	9.13	1.777	0.837	8.997	1.582	0.850
Constant heat flux	10.680	1.439	0.881	9.749	1.22	0.889
Solar driven	9.544	1.077	0.899	8.315	0.884	0.904
Heat boundary	$Q_{240\text{min}}$ (MJ)	$L_{240\text{min}}$ (MJ)	$\eta_{240\text{min}}$	$Q_{120\text{min}}$ (MJ)	$L_{120\text{min}}$ (MJ)	$\eta_{120\text{min}}$
$t_w=270^\circ\text{C}$	9.822	1.124	0.897	6.636	0.367	0.948
$t_w=260^\circ\text{C}$	9.043	1.056	0.895	6.062	0.347	0.946
$t_w=250^\circ\text{C}$	8.073	0.997	0.890	5.753	0.329	0.946
Constant heat flux	7.037	0.535	0.929	3.446	0.128	0.964
Solar driven	5.354	0.397	0.931	1.785	0.055	0.970

6 Conclusions

Based on the development status of solar medium and low temperature thermal utilization systems, this paper introduced the application and performance research on subsystems of the solar system, as well as the matching relationship and the overall dynamic performance. A case with respect to thermal performance of a heat storage driven by different heat collection modes is analyzed. The main conclusions are as follows.

- Solar heat for industrial processes showed a huge potential, which temperature could be

mainly distributed in 60-250 °C. The PTC is more attractive for medium temperature applications. The single performance of the collector has obtained much attention, however, there are only a few studies considered the heat-collecting storage-utilization as a whole system.

■ In the field of thermal energy storage, latent heat storage showed a good potential. The low conductivity of PCMs is an obstacle and many methods have been presented to enhance the heat transfer performance. The analysis of PCM charging and discharging performance was often carried out by numerical simulation, and experiments.

■ As for the matching relationship of the subsystems, it is necessary to establish a dynamic model to study the heat transfer characteristics of the heat-collecting and heat-storing coupled system, as well as the heat-utilization coupled system under different temperature.

■ The case illustrates that for a coupled solar system, the preferred choice of heat boundary condition in the numerical simulation is measured parameters, and the second choice is the constant heat flux corresponding to the same total heat flux as the actual circumstance.

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