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Review article

Heat transfer enhancement technology for fins in phase change energy storage

Xueliang Zhu, Yan Li *, Qunzhi Zhu *

College of Energy and Mechanical Engineering, Shanghai University of Electric Power, Shanghai 200090, China

A R T I C L E I N F O

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A B S T R A C T

In the process of industrial waste heat recovery, phase change heat storage technology has become one of the industry's most popular heat recovery technologies due to its high heat storage density and almost constant temperature absorption/release process. In practical applications, heat recovery and utilization speed are particularly critical. Developing fins with reasonable structure or optimizing the original fins can speed up the process of heat storage and utilization to a certain extent. To improve the heat transfer enhancement design efficiency of fins and expand their application and reference range, this paper summarizes the current development process of new fins in the industry and the optimization results of the size, shape, and arrangement of fins by related researchers. It aims at these research results and how to design the shape, size, quantity, and arrangement of heat exchange fins from four proposed ideas and methods. The strengthening mechanism of heat transfer effect enhancement is analyzed. The research results show that the multi-branch structure is the research direction of the new fin design in the future; the non-uniform fin array has better heat transfer performance than the uniform fin array; the targeted arrangement of fins can maximize the heat transfer effect; there is an optimal number of fins in the heat accumulator; the parameters such as the size and number of fins and the size parameters of the heat accumulator are not independent of each other, but influence and restrict each other. The design process needs to be considered comprehensively; the topology optimization method, multi-objective response surface method, Taguchi method, orthogonal test method, etc., are commonly used in optimizing the shape and size of fins, which are simple and efficient.

Energy is an enduring topic. Improving its utilization efficiency is significant for environmental problems and solving energy shortages. China's energy utilization rate, including processing, transportation, and use, is only 33 %, and considering the efficiency of energy extraction, its total efficiency is less than half that of developed countries. In the proportion of up to 70 % of industrial energy consumption, half of the industrial waste heat is wasted because it cannot be used. It can be seen that how to improve the utilization rate of waste heat is a problem worthy of in-depth consideration. The known industrial waste heat recovery and utilization conditions are relatively harsh. Applying heat storage technology can overcome these difficulties to a certain extent and improve the industrial waste heat utilization rate. In terms of waste heat recovery, the development of heat storage technology is relatively mature, simple, easy to implement, and low cost, which is the best choice for heat energy recovery. Today's heat storage technologies mainly include sensible heat energy storage, latent heat energy storage (phase change energy storage), and thermochemical energy storage. Compared with sensible heat energy storage and thermochemical energy storage, phase change energy storage has more advantages in practical applications:

(1) Higher heat storage density (about 5–10 times that of sensible heat storage), which means a smaller heat storage system volume [1].
(2) The temperature remains almost unchanged during the phase transition. Compared with the instability of chemical reactions, phase change heat storage can provide a more stable heat source, which is efficient and safe.

Due to these unique advantages, phase change heat storage technology is widely used in current industrial production and daily life. In addition to the recovery and utilization of industrial waste heat [2], it also includes "peak shifting and filling valleys" of electricity, energy saving of buildings and air conditioners [3], thermal utilization of solar energy [4], cooling of electronic devices, etc. [5,6].

Although phase change heat storage technology has the advantages

* Corresponding authors.
E-mail addresses: yli@shiep.edu.cn (Y. Li), zhuqunzhi@shiep.edu.cn (Q. Zhu).

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Nomenclature

q_w	heat flux (W/m^2)
N/n	fin number
g	gravity acceleration (m/s^2)
T_w	wall temperature (K)
L	length (m)
T	temperature (K)
R	radius (m)
t	time (s), thickness (m)
P_h	screw pitch (m)
p	fin pitch (m)
H	the total length of the tube (m)
d	diameter (m)
w	thickness (m)
K	thermal conductivity (W·m^{-1}·K^{-1})
C_p	specific heat (J·kg^{-1}·K^{-1})
L	latent heat capacity of PCM (kJ/kg)
α	liquid fraction
ρ	density (kg/m^{-3})
V	volume
Z	average temperature
ρ_s	density of design variables
P	intermediate punishment index
r_f	filter radius
ρ_f	density of filtered design variables
ρ_p	density of required design variables
q	average heat storage rate (W)
Q	heat storage capacity (J)
x, y, z	x, y, z-component in a Cartesian coordinate system

Greek symbols

σ	Thickness of fin (m)
θ	Branch angle (°)
β	Angle (°)

Subscripts

in	inner
f	fin
o	outer
int	initial
e	end

Abbreviations

PCM	phase change materials
RSM	Response Surface Method
HTF	heat transfer fluid
CFD	Computational Fluid Dynamics
LHTES	Latent Heat Thermal Energy Storage
HSU	Heat Storage Unit

that these sensible heat storage and thermochemical heat storage do not have but is limited by the low thermal conductivity of phase change materials (PCM), the temperature distribution uniformity of phase change heat storage system and transient thermal response is not ideal. There are many studies in the industry on improving the heat transfer characteristics of the phase change heat storage system, and specific results have been achieved. It mainly includes preparing composite phase change materials with better thermophysical properties, developing functional thermal fluids that combine heat storage and heat release, and the structural design and heat transfer enhancement of heat storage devices. The working temperature and working environment limit the design and development of the first two. In contrast, the design of the heat storage device is not subject to such constraints and can be applied to a broader range of industrial application scenarios.

In the current heat storage unit design, adding fins as the expansion surface is a suitable method, with intense heat exchange performance, low cost, simple manufacturing, mass production, and no side effects. Moreover, adding fins is better than adding other thermally enhanced materials. Mahdi et al. [7] and Lohrasbi et al. [8] compared the heat transfer enhancement by adding fins and nanoparticle-enhanced phase change materials. The results consistently show that the addition of fins hardly affects the maximum heat storage capacity of the heat accumulator. Moreover, the charge and release rate of the phase change accumulator after adding fins is significantly higher than that of adding nanoparticle-enhanced phase change materials.

In addition, the heat transfer performance of adding reasonably designed fins in practical industrial applications is also quite impressive. The heat recovery system must have a stable waste heat supply in waste heat recovery and utilization. The unstable heat source in a large-scale heat pump system can easily damage the compressor and reduce the system's operating efficiency [9]. The thermal utilization of solar energy also has similarities in this regard. In the actual system operation, the thermal energy should not be accumulated in the pipeline to cause local overheating. However, it should be taken away in time to form a stable heat source supply. To form a stable heat source supply, it is necessary to charge and release heat simultaneously, which requires the fin structure to have high heat exchange efficiency. Therefore, a good fin structure design is crucial for the recovery and utilization of heat energy.

For example, Mahdi et al. [7] studied the optimal fin configuration under the condition of simultaneous charging and discharging in triple tubular accumulators and used the Response Surface Method (RSM) to optimize. The final fin configuration made the liquid fraction of the phase transition equilibrium state of the PCM as high as 0.65, which indicates that the fin structure and arrangement have excellent heat exchange efficiency and can efficiently convert unstable heat sources into stable heat sources. Regarding "shifting peaks and filling valleys" and building energy conservation, the baffled phase-change thermal storage electric heating device designed by Hu et al. [10] adopted the optimal number of plate fins. The device stores heat when electricity consumption is low and releases heat when electricity consumption peaks. It solves the mismatch of supply and demand time and dramatically improves the heat storage efficiency of the heat accumulator due to the design of the optimal number of fins and achieves the energy-saving effect. In the cooling of electronic devices, such as the cooling of battery packs in electric vehicles [11–14], the rapid heat dissipation of electronic devices such as laptop computers and air conditioners, etc. All have higher requirements for the charging and discharging rate of the phase change heat accumulator. In this regard, a reasonable choice of fin shape and structure may be able to solve this problem to a certain extent. For example, Jaworski [5] found that using pin fins in the rectangular heat accumulator has better heat dissipation efficiency because the needle structure is a thin-walled tube with a smaller diameter. After filling the PCM in the tube, the phase change accumulator has two characteristics: a large heat transfer surface and high heat capacity, so it has better heat transfer efficiency. It is very effective for cooling problems of electronic devices.

In addition, to achieve more efficient and broader industrial applications, in the past few decades, many researchers have developed a wide variety of fins, and the size and structure of the fins have been optimized again. There are also many in-depth studies on the number, array, installation method, arrangement position, etc. These studies' design ideas and optimization methods are worth learning and referencing. Starting from the design of various parameters of fins, this paper discusses some representative academic research results. It comprehensively sorts design ideas, optimization methods, and heat transfer enhancement mechanisms. On these bases, this paper puts forward the starting ideas of 4 kinds of fin designs and provides the theoretical methods of 4 kinds of fin designs commonly used in the industry. It is hoped that the relevant content can help researchers to improve the design efficiency of heat transfer enhanced fins and provide a reference for the optimal design of the matching finned heat exchanger. This article will discuss the design logic of fin heat transfer enhancement shown in Fig. 1.

1. Shape design of fins

1.1. New fin design

As an industrial product with a simple structure and low cost, fins can be seen everywhere in heat exchangers in daily production and life. The heat transfer enhancement of the heat exchanger by adding fins is mainly to strengthen the heat conduction by increasing its heating area or use its structure to strengthen the natural convection and speed up the heat flow between the high and low-temperature media, to optimize its heat exchange effect. In the current selection of heat exchange fins, the more common ones are rectangular fins, annular fins (circular fins), triangular fins, helical fins, branch-shaped fins, etc., and different shapes of fins have different effects on heat transfer. Generally speaking, the

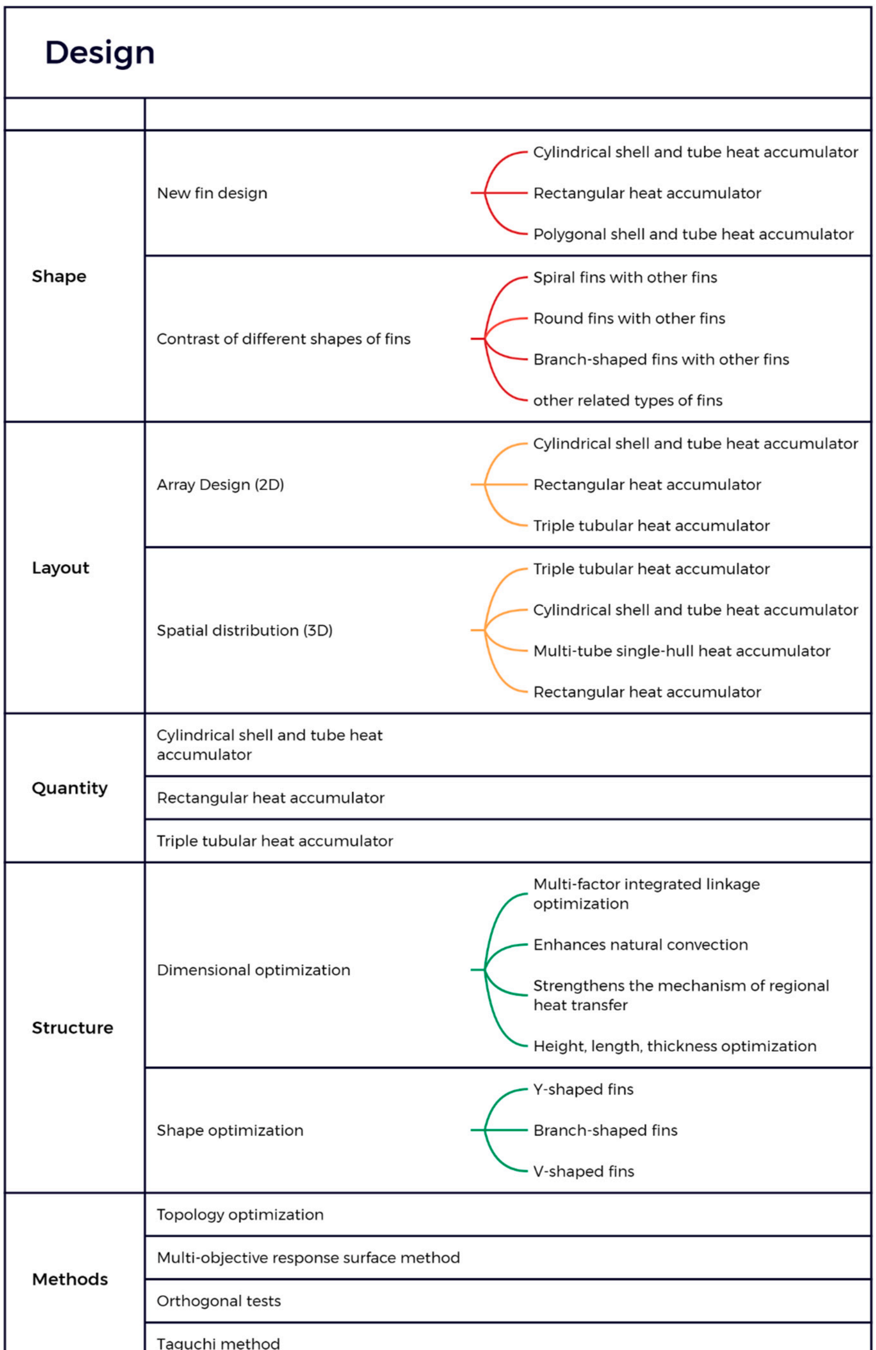

Fig. 1. Design logic diagram for enhanced finned heat transfer.

accumulator models used in the new fin design are mainly divided into cylindrical shell-and-tube accumulators, rectangular accumulators, and regular polygonal shell-and-tube accumulators. Among them, the design details of various new types of fins in different accumulator models are shown in Table 1.

1.1.1. Cylindrical shell and tube heat accumulators

Wang et al. [15] designed a new type of helical fin type phase change heat storage unit, which can shorten the melting time of PCM by 12.21 % compared with the same volume of flat fins; and with the increase of the thickness, number and helical period of the helical fins, the melting time of the PCM decreased significantly. The heat transfer performance is also improved to a certain extent.

The traditional fin geometry and size must be designed for specific use cases. You et al. [16] adopted a topology optimization method to design the fin shape in the phase change heat storage unit. Under arbitrary boundary conditions and geometric conditions, the branch-shaped fins can be obtained with free extension and development and compared with the traditional square fins. Conical fins, the fins designed in this way, have better heat storage performance. This is because the optimized extension of the fins in the radial direction helps the heat transfer to the PCM area away from the heat transfer channel faster, avoiding the PCM that cannot contact the heat exchange material to rely on its own lower thermal conductivity for heat exchange. In addition, the branch-shaped structure distribution can better transfer heat to the gap between the fins. In addition to the enhancement of radial heat transfer, due to the many branches, and dense structure of the branches, the longitudinal heat transfer is also enhanced, and the temperature distribution is more uniform than before.

Zhao et al. [17] optimized the shape of the longitudinal fins in the cylindrical shell-and-tube heat accumulator (the longitudinal fins in this paper refer to longitudinal rectangular fins unless otherwise specified). Comparison of Y-fins formed by topology optimization and longitudinal fins before optimization, the heat storage time is shortened by 70 %, and the heat release time is shortened by 81 %. The detailed structure of the fin is shown in Fig. 2(a). Zhang et al. [18] also performed topology optimization on longitudinal fins in cylindrical shell-and-tube accumulators and reconstructed longitudinal fins through design variable threshold screening and contour filling methods. Compared with the original longitudinal fin, the heat transfer efficiency of the optimized fin is increased by 40 %, and the phase transition time of the heat charging and discharging process is significantly shortened. Ge et al. [19] performed topology optimization on the longitudinal fins in the shell-and-tube accumulator, and the optimized fin structure was shaped like an airplane. Through experiments comparing the four aircraft-shaped fins with the four longitudinal fins before topology optimization, it is found that the fins after topology optimization have better heat transfer performance, and the heat release time is shortened compared with before optimization increased by 57.1 %. The detailed structure of the fin is shown in Fig. 2(b).

Aly et al. [20] designed a sawtooth-shaped corrugated fin, compared it with the longitudinal flat straight fin, and analyzed the solidification rate of PCM. The result shows that the total solidification time of the PCM in the phase change heat accumulator is reduced by 30 % to 35 %. Furthermore, the solidification rate will further increase with the increase in corrugations and the height of corrugations. However, due to the low heat transfer rate at the corrugated concave surface, the solidification rate will decrease when there are fewer fins, so it is best to use a large amount in the phase change accumulator to offset the low heat transfer rate at the concave surface.

Li et al. [21] punched a circle of circular holes on the annular fin and found that there are optimal hole diameters, hole numbers, and hole positions through simulation. Under specific configuration parameters, compared with non-porous annular fins, the melting time of PCM is shortened by 5.49 %, and the heat storage capacity is increased by 0.21 %. The principle is that the pore size increases, the natural convection

Table 1
The design of various new fins in different heat storage models.

Accumulator type	Reference	Fins configurations	Research contents	Finding description
Cylindrical shell and tube	[15]	New spiral fins	Numerical/ melting	The PCM melting time can be reduced by 12.21 % compared to the same volume of flat fins.
	[16]	New tree fins	Numerical/ melting	The optimized extension of the fins in the radial direction facilitates faster heat transfer to the PCM area away from the heat transfer channels.
	[17]	New Y-shaped fins	Numerical/ melting/ solidification	A 70 % decrease in heat storage time and an 81 % decrease in heat release time.
	[18]	New fork-shaped fins	Numerical/ melting/ solidification	The optimized fins have a 40 % increase in heat transfer efficiency compared to the original longitudinal fins.
	[19]	Aircraft shaped fins	Numerical/ Experimental/ melting	Compared with before optimization, the exothermic time was reduced by 57.1 %.
	[20]	Serrated corrugated fins	Numerical/ solidification	Compared to flat fins of the same configuration, PCM has a 30 % to 35 % reduction in total solidification time.
	[21]	New annular fins with holes	Numerical/ melting	Compared to solid ring fins, the PCM has a 5.49 % reduction in melting time and a 0.21 % increase in heat storage capacity.
	[22]	New T-shaped fins	Numerical/ melting	The melting time of the PCM is reduced by 33 % with the addition of T-shaped fins compared to longitudinal fins.
	[23]	New triangular fins	Numerical/ solidification	Compared to conventional longitudinal rectangular

(continued on next page)

Table 1 (*continued*)

Accumulator type	Reference	Fins configurations	Research contents	Finding description
				fins, a 30.98 % reduction in PCM solidification time.
	[24]	New spider web fins	Numerical/ solidification	8 bifurcated spider web structure with a 47.9 % reduction in solidification time compared to longitudinal fins
Rectangular	[25]	New multi-branch conical fins	Numerical/ melting	The melting rate of the new multi-branched conical fins with a volume ratio of 0.1 is equivalent to that of a plate fin with a volume ratio of 0.15.
	[26]	Fractal fins of ferns	Numerical/ melting	Compared to conventional fins, the melting time is reduced by 40.3 %, and the average heat storage rate increases by 88.3 %.
	[27]	New stepped fins	Numerical/ melting	At a step ratio of 0.4, the melting rate increases by 56.3 % at 800 s compared to conventional equal-volume horizontal fins.
Regular polygon shell and tube	[28]	Koch fractal fins	Numerical/ melting	Its heat storage rate is 6 times faster than that of longitudinal fins.
	[29]	New longitudinal triangular fins	Numerical/ solidification	The fins are capable of a 30 % increase in solidification rate compared to conventional rectangular fins.

increases, and the thermal conductivity decreases. The farther the hole is from the heat transfer fluid (HTF) pipe, the stronger the heat conduction and the weaker the convection. There is an optimal balance between heat conduction and convection. The detailed structure of the fin is shown in Fig. 2(c).

Al-Mudhafar et al. [22] designed a T-shaped fin and compared it with a longitudinal fin in a shell-and-tube accumulator. The study found that the melting time of PCM after adding T-shaped fins was reduced by 33 % compared with longitudinal fins for the same PCM volume. This is due to the increase of heat penetration depth and the heat transfer specific surface area, which reduces the thermal resistance and accelerates the heat transfer between the HTF and the PCM. The detailed structure of the fin is shown in Fig. 2(d). Yao et al. [23] designed a new type of triangular fins for the shell-and-tube heat accumulator. The shape of the fins is based on the longitudinal rectangular fins, which change the cross-section along the flow direction of the pipes. It gradually becomes a larger diamond shape instead of a rectangle. Through simulation research, it is found that the new triangular fin can reduce the solidification time of PCM by 30.98 % compared with the traditional longitudinal rectangular fin. Wu et al. [24] fabricated a novel cobweb-shaped fin and compare it to a longitudinal fin of the same volume for the PCM solidification process.

In contrast, the cobweb-shaped fins are superior to the solidification heat transfer enhancement of PCM due to the elimination of the heat transfer lag zone. Wu et al. [24] proposed an 8-branched spider web structure to further improve its exothermic performance, which reduced the solidification time of PCM by 47.9 % compared to longitudinal fins. The detailed structure of the fin is shown in Fig. 2(e).

1.1.2. Rectangular heat accumulator

Xie et al. [25] used the topology optimization method to design the shape of the plate fins in the rectangular phase change accumulator. They obtained a new type of multi-branch conical fins through topology optimization. According to the simulation results, in the new multi-branch conical fin heat storage unit with a volume ratio of 0.1, the melting rate of PCM is equivalent to that of a plate fin with a volume ratio of 0.15. Moreover, in the new multi-branch conical fin heat storage unit with a volume ratio of 0.15, the melting rate of PCM is faster than that of plate fins with a volume ratio of 0.2. Inspired by the leaves of multi-forked ferns, Deng et al. [26] designed a fern-like fractal fin and used RSM to optimize the structural parameters of the fin. The optimized fin aspect ratio is 0.94, and the branch angle is 54.7°. Compared with traditional fins, the melting time of PCM is reduced by 40.3 %, and the average heat storage rate increases by 88.3 %. The detailed structure of the fin is shown in Fig. 2(f).

As a critical parameter in thermal energy storage, the average heat storage rate is mathematically expressed as:

$$q_{hsu} = \frac{Q_{hsu}}{t_e - t_{int}} \tag{1}$$

Among them, t_{int} and t_e are the start time and end time of the heat charging process, respectively, Q_{hsu} is the total heat storage of the heat storage unit, and the mathematical expression is:

$$Q_{hsu} = Q_{fin} + Q_{PCM} \tag{2}$$

It can be summed up in one sentence. The total heat storage is equal to the heat storage of the metal fins plus the heat storage of the PCM. Conversely, when the heat storage is changed to heat release, q_{hsu} can also be quantified as the average heat release rate.

Nakhchi et al. [27] developed a new type of stepped fin. The shape of the fin is stepped upward or downward. In this study, different step ratios were set for comparative research. When using down stepped fins with a step ratio of 0.4, compared with traditional equal volume horizontal fins, the melting rate of PCM increased by 56.3 % at 800 s and 65.5 % at 3600 s. The detailed structure of the fin is shown in Fig. 2(g).

1.1.3. Regular polygon shell and tube heat accumulator

Inspired by snowflakes, Li et al. [28] designed Koch fractal fins with a similar structure to snowflakes. By comparing the heat transfer with longitudinal fins in the regular hexagonal vertical shell-and-tube accumulator, it is found that the heat release rate of Koch fractal fins is faster than that of longitudinal fins, and its temperature distribution is more uniform. Due to its larger specific surface area, point-to-surface heat flow path, and minor thermal resistance, Koch fractal fins can store heat up to six times faster than longitudinal fins. The detailed structure of the fin is shown in Fig. 2(h). Like the Koch fractal fin, Liu et al. [29] designed a new type of longitudinal triangular fin, which differs only in shape from the Koch fractal fin. In the same heat accumulator model, compared with the traditional rectangular fin, the solidification rate of this fin can be increased by 30 %.

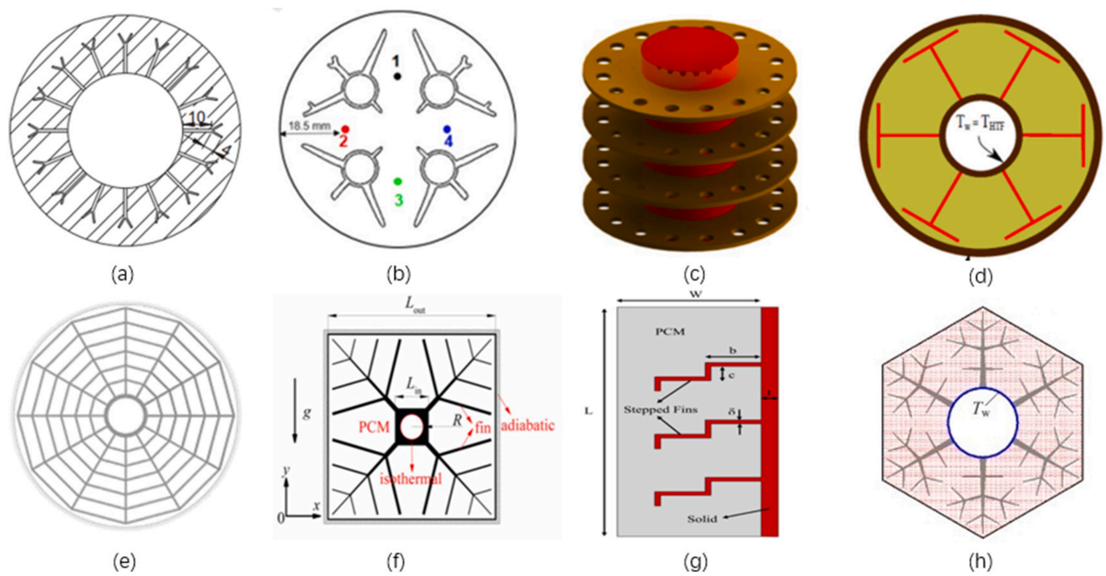

Fig. 2. Design of the new fin shape (a) Y-shaped fin formed after topology optimization [17] (b) Aircraft-shaped fin formed after topology optimization [19] (c) Circular perforated fin [21] (d) T-shaped fin [22] (e) Spider-web fin [24] (f) Fern-shaped fin [26] (g) Downward stepped fin [27] (h) Snowflake fin [28].

1.2. Comparison of different fin shapes

Fins of different shapes have different heat exchange performances, and the heat exchange level of the fin itself will also be different under different heat exchange conditions. In the current comparison of heat transfer performance of fins with different shapes, the most studied are spiral fins, circular fins (annular fins), branch fins, longitudinal fins, and needle fins. The heat transfer performance comparison details of these fins with different shapes are shown in Table 2.

1.2.1. Comparison of helical fins and other fins

Zhang et al. [30] established a three-dimensional model considering natural convection for shell-and-tube heat exchangers. They studied the heat transfer of fin structures with different helicity when the heat storage device was placed horizontally and vertically. The study shows that due to the presence of vortex, fins with a slight helix angle are suitable for the vertical direction, and large helix angles are suitable for the horizontal direction. For example, when the fin structures are annular fins (0°) and double helical fins (25°), the heat storage device placed in the vertical direction has better heat transfer performance. While for the quad-spiral fins (57°) and longitudinal fins (90°), the heat storage device has more superior heat transfer performance when the heat storage device is placed horizontally. The comparison details of fins with different helicity are shown in Fig. 3.

Helical fins were also studied by Ghalambaz et al. [31]. They compared the effects of helical fins and straight fins on the melting performance of PCM, and found that the melting time of PCM with helical fins was reduced by 18 % compared with the straight fins with the same volume ratio. Furthermore, when the number of helical fins was increased from 2 to 6, the heat storage rate was increased by 25 %. Sun et al. [32] conducted a comparative study on the effect of installing spiral fins, straight fins, and no fins on the solidification time of PCM in the shell-and-tube accumulator. The solidification time was shortened by 12.7 % and 22.9 %, and the heat release rate was increased by 12.4 % and 22.8 % for the finned and finless ones.

Zonouzi et al. [33] also studied the effects of installing one helical fin, four longitudinal straight fins, and no fins on the melting process of PCM in a shell-and-tube accumulator placed vertically. The length of the helical fins in this simulation equals the sum of the lengths of the four longitudinal straight fins. It is found that when the HTF is 355 K, compared with the longitudinal straight fins, the helical fins can reduce the melting time of PCM by 21.5 % due to the existence of vortex.

1.2.2. Comparison of circular fins with other fins

Abdulateef et al. [34] experimentally investigated the difference in heat transfer performance between circular fins and longitudinal fins in shell and tube heat exchangers at the same specific volume. They found that the heat charging time of circular fins was reduced by 69 % compared to finless, while longitudinal fins were reduced by 55 % compared to finless heat transfer. The maximum heat storage capacity of circular fins is 52 % higher than that of longitudinal fins. Elmaazouzi et al. [35] also compared the heat transfer performance of circular fins, longitudinal fins, and honeycomb fins in a vertical shell and tube accumulator by simulation. The same fin thickness was added as a constraint. The complete melting time of longitudinal fins was reduced by 4 % and 8.5 % compared to circular and honeycomb fins, respectively, when all fins were 1 mm thick.

Tay et al. [36] established three heat exchange models for the shell and tube heat exchange system. The first model is embedded with pin-shaped fins in the HTF pipe, the second model is embedded with circular fins, and the third model does nothing and is surrounded by PCM with ordinary copper pipes. The study shows that under the same specific volume, due to the larger heat transfer surface area of the circular finned tube, the heat transfer rate is higher than that of the pin-shaped finned tube, and the total energy storage density of the heat storage system will not be affected. Hamdani et al. [37] studied the heat transfer characteristics of circular fins and longitudinally long fins in cylindrical shell-and-tube heat accumulators. The study found that under the influence of natural convection, the melting rate when adding longitudinal fins to PCM was two times faster than that of adding circular fins. Beemkumar et al. [38] encapsulated rectangular fins, pin fins, and circular fins in a PCM-equipped sphere and studied their heat transfer efficiency. The results showed that compared with pin fins and rectangular fins, the circular fin has a larger contact-specific surface area, so its heat transfer efficiency is also the best.

1.2.3. Comparison of branch fins with other fins

Wu et al. [39] conducted a comparative study on branch-shaped fins and longitudinal straight fins' charging and discharging performance. Through experimental comparison, it was found that the melting time of branch-shaped fins was slightly shorter than that of straight vertical fins. However, due to its strong thermal conductivity, the rate of the

Table 2
Details of the comparison of the heat transfer performance of different fin shapes.

Accumulator type	Reference	Fins configurations	Research contents	Finding description
Cylindrical shell and tube	[30]	fins with different helix degrees	Numerical/ melting/ solidification	The small spiral angle fins have good heat transfer performance when placed vertically, while the large spiral angle has good heat transfer performance when placed horizontally.
	[31]	Spiral fins/ longitudinal straight fins	Numerical/ melting	The PCM melting time for spiral fins is reduced by 18 % compared to longitudinal fins.
	[32]	Spiral fins/ longitudinal straight fins/ no fins	Numerical/ solidification	The 4 spiral fins take 12.7 % and 22.9 % less time than the 4 longitudinal fins and no fins.
	[33]	Spiral fins/ longitudinal fins/no fins	Numerical/ solidification	Compared to longitudinal straight fins, spiral fins can reduce the melting time of PCM by 21.5 % more due to the presence of vortex.
	[34]	Round fins/ longitudinal fins	Experimental/ melting	Compared to longitudinal fins, round fins' maximum heat storage capacity is 52 % higher.
	[35]	Round fins/ longitudinal fins/ honeycomb fins	Numerical/ melting	The longitudinal fins have a 4 % and 8.5 % reduction in total melting time compared to the circular and honeycomb fins.
	[36]	Round fins/ pin fins/no fins	Numerical/ melting	The heat transfer rate of round-finned tubes is higher than that of pinned fins.
	[37]	Round fins/ longitudinal fins	Numerical/ melting	The longitudinal fins do not restrict natural convection, so the melting rate of the PCM is 2 times faster than that of the circular fins.
Spherical	[38]	Round fins/ rectangular fins/pin fins	Numerical/ melting/ solidification	Round fins have a larger contact-specific surface area, so their heat transfer efficiency is also the best.
Cylindrical shell and tube	[39]	Forked fins/ longitudinal fins	Experimental/ melting/ solidification	Compared to longitudinal straight fins, the increase in melting rate is not significant, but the increase in exothermic rate is very significant.
	[40]	Forked fins/ longitudinal fins	Numerical/ melting	Fork fins provide better and deeper lateral heat transfer than longitudinal fins.
Rectangular	[41]	Forked fins/ rectangular fins	Numerical/ melting/ solidification	Due to the multi-branch structure, the heat storage efficiency of the forked fins is higher than that of the rectangular fins, and the more branches there are, the stronger the heat transfer.
Spherical	[42]	Hollow cylindrical fins/solid cylindrical fins	Experimental/ melting	The solid cylindrical fins have a 23 % reduction in heat filling time, while the hollow cylindrical fins have a 28 % reduction in heat filling time.
Cylindrical shell and tube	[43]	T-shaped fractal branch fins/Y-shaped fractal branch fins	Numerical/ melting	The T-shape has a 5.6 % reduction in melting time compared to the Y-shape at equal volume.
Rectangular	[44]	Downward stepped fins/ upward stepped fins	Numerical/ melting	The melting characteristics of downward-stepped fins with a larger step ratio are significantly better than those of upward-stepped fins.
Cylindrical shell and tube	[45]	Circular fins/ longitudinal fins/linear wound fins	Numerical/ melting	In terms of heat storage rate, wire-wound fins are 20.95 % and 35.96 % higher than circular and longitudinal

(continued on next page)

Table 2 (*continued*)

Accumulator type	Reference	Fins configurations	Research contents	Finding description
				fins, respectively.
Triple shell and tube	[46]	Longitudinal straight fins/ longitudinal L-shaped fins	Numerical/ melting/ solidification	Longitudinal L-shaped fins have almost the same melting rate as longitudinal straight fins, but the solidification rate is 2–5 % faster.
Rectangular	[47]	Snowflake fins/ longitudinal fins	Numerical/ solidification	Adding snowflake fins resulted in a solidification rate almost 7.8 times faster than no fins, while longitudinal fins were only 4.5 times faster.
Cylindrical shell and tube	[48]	Circular perforated fins/circular solid fins	Numerical/ melting	When charging the heat, the annular perforated fins are more advantageous when the accumulator is placed vertically. At the same time, in all other directions, the solid fins are 5 % more efficient than the perforated fins.

exothermic process of the branch-shaped fins is more prominent. In practical applications, the appropriate number of fins should be selected to maximize the heat transfer effect. Luo et al. [40] also conducted a comparative study on the branch-shaped fins and the longitudinal fins. The results showed that the branch-shaped fins could accelerate the phase change process in the phase-change heat storage device. The principle is that the branch-shaped fins can disperse the entire PCM area into multiple independent areas to make its temperature distribution more uniform. The lateral heat transfer of the branch-shaped fins is better than that of the longitudinal fins and has better in-depth heat transfer capability. Shukla et al. [41] compared the heat transfer enhancement capabilities of twig-shaped fins and rectangular fins in a rectangular heat accumulator, in which the fins absorb heat from the front plate, transfer it to the PCM, and then the heat is transferred to the rear plate. They found that the branch-shaped fins have better energy storage efficiency than the rectangular fins due to their multi-branched structure. The heat transfer enhancement effect is also more apparent. In contrast, changing the length ratio of the branch-shaped fins and the angle of the fins brings little or negative benefits.

1.2.4. Comparison of other related types of fins

Nallusamy et al. [42] studied the performance difference between the inner hollow cylindrical fin and the inner solid cylindrical fin in the spherical capsule regarding heating performance. They found through experiments that when the temperature of the HTF increased from 650 °C to 700 °C, the heating time of the inner solid cylindrical fin was

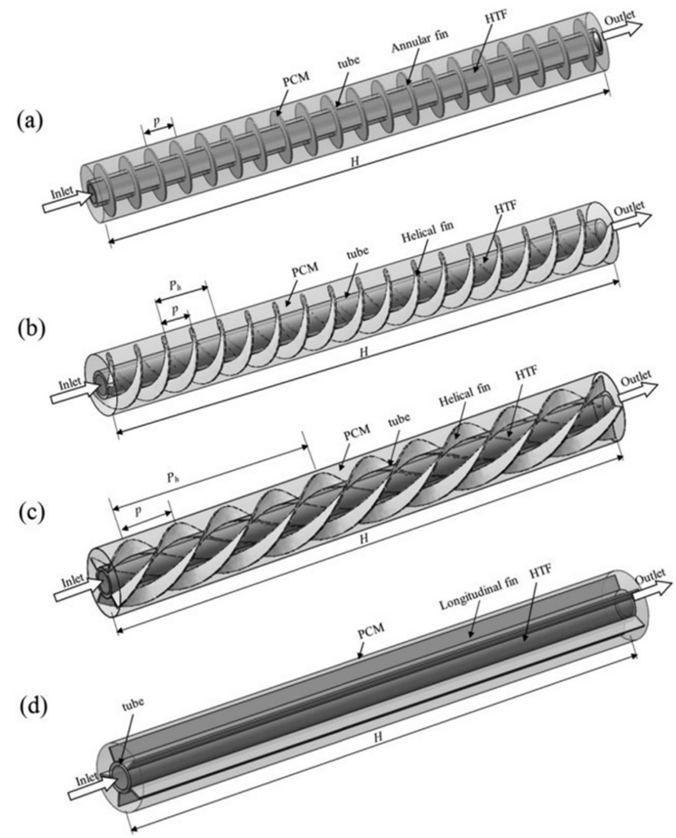

Fig. 3. Fin structure with different helix degrees [30] (a) Circular fin (b) Double spiral fin (c) Quadruple spiral fin (d) Longitudinal fin.

shortened by 23 %. The heating time of the inner hollow cylindrical fins is shortened by 28 %. The inner hollow cylindrical fins are more sensitive to temperature rise. Deng et al. [43] compared T-shaped and Y-shaped fractal branch fins and found that the melting time of T-shaped fins was reduced by 5.6 % compared with Y-shaped fins under the same volume. Studies have shown that T-fins with a medium branch level, a large length index, and a lower width index can better balance heat conduction and convection. This study's optimized T-shaped fractal branch fin reduces the charging time by 52.9 % compared with the traditional plate-shaped fin. Nakchi et al. [44] conducted a comparative study of stepped fins in a rectangular heat storage unit through a simulation study. This study found that the melting characteristics of downwardly stepped fins with larger step ratios were significantly better than those of upwardly stepped fins due to natural convection and recirculation flow at the fin surface.

Khan et al. [45] conducted a comparative study of annular fins, longitudinal fins, and wound fins in cylindrical phase change heat accumulators. Through the melting simulation analysis of PCM, it is found that compared with annular fins and longitudinal fins, the heat storage rate of wound fins is 20.95 % and 35.96 % higher than that of circular fins and longitudinal fins, respectively. The related fin structure is shown in Fig. 4. Patel et al. [46] compared the longitudinal straight fins and the longitudinal L-shaped fins in the triple tubular heat accumulator. The simulation results showed that the melting rates of the longitudinal L-shaped fins and the longitudinal straight fins were almost the same. However, The solidification rate of L-shaped fins is 2 %–5 % faster than that of straight fins.

Sheikholeslami et al. [47] compared the designed snowflake fins with longitudinal fins in a rectangular heat storage unit concerning the solidification performance of PCM. Simulation studies found that the solidification rate of the rectangular heat storage unit with snowflake fins is nearly 7.8 times faster than that of the finless rectangular heat

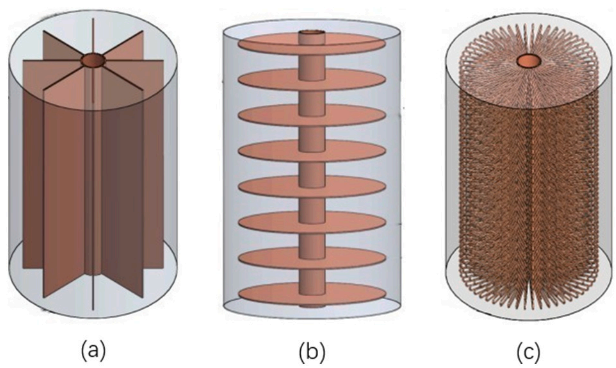

Fig. 4. Comparison of the structure of longitudinal fins, annular fins, and wound fins [45].

storage unit, while the longitudinal fins are only 4.5 times faster. Kalapala et al. [48] studied the heat transfer performance of annular perforated fins and solid annular fins in shell-and-tube accumulators at different inclination angles. The research results show that when the accumulator is charged, the annular perforated fins are more advantageous when the accumulator is placed vertically, increasing the heat storage rate in the initial melting stage. Solid fins are 5 % more efficient than perforated fins in all other directions. The dominant heat transfer mechanism when the accumulator releases heat is heat conduction, and the heat transfer efficiency of the two is not much different.

2. Arrangement of fins

2.1. Array design of fins

The fin array predominantly affects the PCM's temperature uniformity and the final heat transfer effect. In the actual design process, the phase change heat accumulators installed with fins can be classified into cylindrical shell-and-tube heat accumulators, rectangular heat accumulators, triple-tube heat accumulators, etc. The general appearance of these heat accumulators is shown in Fig. 5 and Table 3 for the design details of the fin arrays in different phase change heat accumulators.

2.1.1. Cylindrical shell and tube heat accumulators

Ling et al. [49] studied the heat storage performance of the shell-and-tube phase change heat accumulator with annular fins. They found that the thermal conductivity of the fins has little effect on the heat storage efficiency, while the fin spacing is the primary factor that affects the heat transfer rate. If economy is considered, a material with a lower thermal conductivity can be selected. Shahsavar et al. [50] studied the distribution interval of annular fins in a horizontally placed cylindrical shell-and-tube heat exchanger, conducted multiple tests for five specific annular fins, and determined the optimal position of each fin in turn. The results show that: for the melting process, there is an optimal fin position distribution in the heat exchange structure; the non-uniform fin distribution is better than the uniform distribution, and the melting time of the PCM with the optimal fin array under the same heat exchange conditions is 23.9 % lower than that of the uniform fin array.

In the study, Kuboth et al. [51] pointed out that the fin density increases linearly and exponentially along the pipe outlet direction, and when the ratio is optimal, the total heat release increases by 3 %. Pu et al. [52] studied the effect of different arrangements of annular fins on the melting of PCM for a vertically placed cylindrical shell-and-tube accumulator. The simulation includes a full tube uniform distribution (middle fin), all uniformly distributed in the upper half (upper fin) or lower half (lower fin) area and an arithmetic distribution where the fin spacing gradually increases by one unit (arithmetic fins). The research results show that the arithmetic fin has the shortest melting time, 2827 s, followed by the middle and lower fin. The melting time of PCM after adding the upper fins was the longest, 4862 s. The related fin structure is shown in Fig. 6. Sarani et al. [53] tested 17 different fin cases and found that the solidification rate of structurally discontinuous fins is slower than that of structurally continuous fins. Discontinuous fins are selected for situations requiring a longer energy release time. Better than continuous fins.

2.1.2. Rectangular heat accumulator

Ramana et al. [54] investigated the effect of the arrangement of vertically placed pin fins in a rectangular phase change accumulator on the heat transfer effect. By arranging the fins in a smooth and staggered manner, they found that the staggered fin arrangement possessed better heat transfer. Shafiq et al. [55] investigated the design comparison of different fin combination arrays in a rectangular heat storage unit. If the fin volume was controlled, the combined arrangement of straight fins and angular fins creates an enlarged heat dissipation surface, which increases the melting rate by 18 % and the heat storage rate by 19.8 %. However, the combined arrangement of straight fins of different lengths and thicknesses resulted in even better heat transfer performance and a 39 % increase in melting rate. Details of the related structures are shown in Fig. 7.

Hu et al. [56] studied the effect of vertical plate fin spacing on the

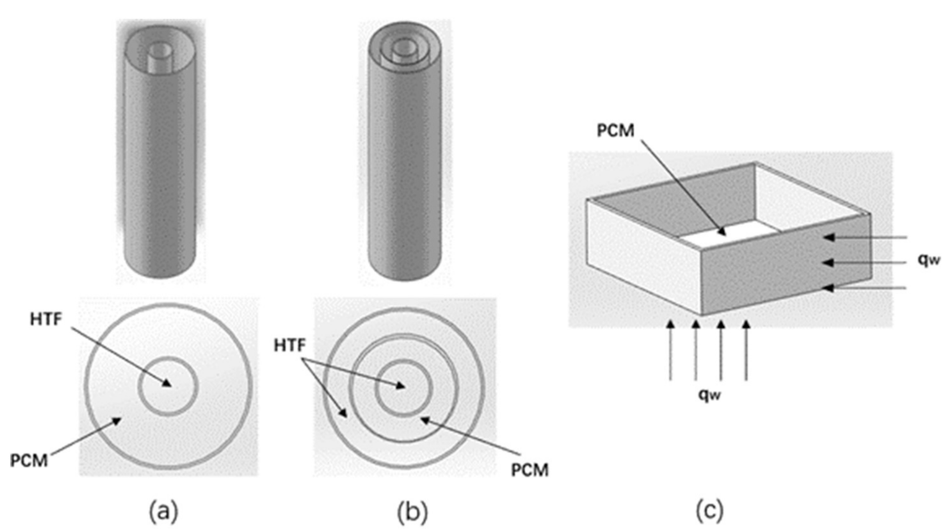

Fig. 5. (a) Cylindrical shell and tube heat accumulator (b) triple tube heat accumulator (c) rectangular heat accumulator.

Table 3
Details of the fin array design.

Accumulator type	Reference	Fins configurations	Research contents	Finding description
Cylindrical shell and tube	[49]	Annular fins	Numerical/ melting	Fin thickness has little effect on heat transfer; the fin spacing is the main factor, with smaller spaced fin arrangements providing better melting enhancement.
	[50]	Annular fins	Numerical/ melting	The optimum fin array should be non-uniform rather than uniform, and the melting time of a PCM with non-uniform fins is 23.9 % lower than with uniform fins.
	[51]	Annular fins	Numerical/ melting	When the fin arrangement density increases linearly and exponentially in the direction of the pipe exit, the total heat release from the PCM increases by 3 % compared to a uniformly distributed fin with a constant arrangement density.
	[52]	Annular fins	Numerical/ melting	Compared to the whole tube, the upper half of the tube has an even concentrated distribution and the shortest melting time of 2827 s for the arithmetic fins (fins spaced in an equal series with a tolerance of 1).
	[53]	17 different fins	Numerical/ solidification	The solidification rate of PCM is slower for structurally discontinuous fins (multiple rectangular fins that have been split and arranged evenly) than that of structurally continuous fins (a whole rectangular fin that has not been split).
Rectangular	[54]	Pin fins	Numerical/ melting/ solidification	The staggered arrangement of the fins provides better heat transfer than an in-line arrangement.
	[55]	The array of different fin combinations	Numerical/ melting	The combined arrangement of straight and angular fins increases the melting rate and heat storage rate of the PCM. However, by combining straight fins of different lengths and thicknesses, the melting rate of the PCM can be increased by another 21 %.
	[56]	Plate fins	Numerical/ solidification	The non-uniformly arranged fins have a lower temperature difference inside the PCM than a uniform distribution and a 14 % higher heat storage rate.
Triple shell and tube	[57]	Annular fins	Numerical/ melting	The staggered arrangement of the fins can increase the melting rate of the PCM by 37.2 % and also enhance the PCM's sensitivity to changes in flow and temperature of the HTF.
	[58]	Longitudinal fins	Numerical/ melting	The PCM ring is divided into three layers—inner, middle, and outer—with 4 fins evenly distributed in each layer. The staggered 90° arrangement between the layers allows the PCM to reduce melting time by 71 %.
	[59]	Longitudinal fins	Numerical/ solidification	The placement of the equal-length longitudinal fins of the inner and outer shell at 45° intervals reduces the unit's rotation's effect on the

(continued on next page)

Table 3 (*continued*)

Accumulator type	Reference	Fins configurations	Research contents	Finding description
				PCM's melting time while effectively reducing the PCM's solidification time.

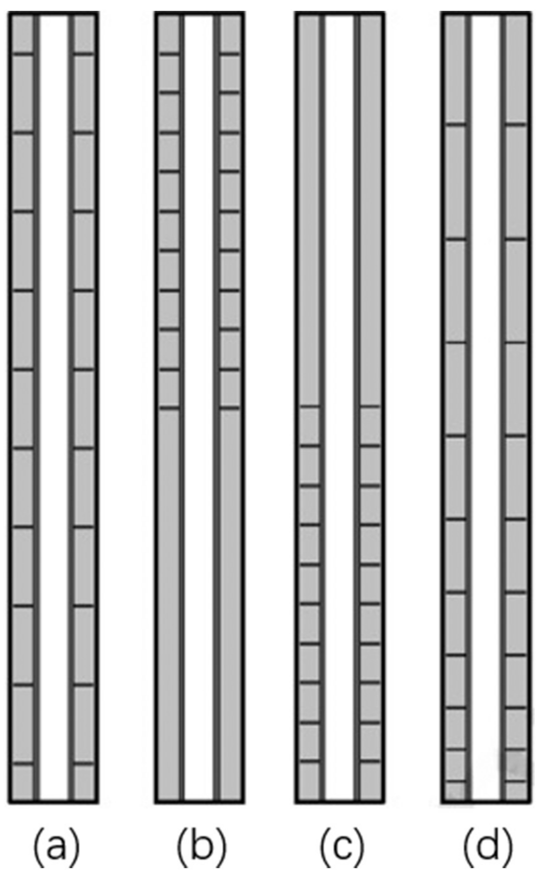

Fig. 6. Different distribution of annular fins [52] (a) Uniform distribution over the whole tube (middle fin) (b) All uniform distribution in the upper half of the area (upper fin) (c) All uniform distribution in the lower half of the area (lower fin) (d) Gradually increasing arithmetic distribution (arithmetic fin).

heat transfer effect in a rectangular container. The study showed that the temperature distribution of the non-uniform fin array is more uniform, and the average temperature difference is within 1 °C. The average temperature difference of the evenly distributed plate fins is about 3 °C, and the heat storage rate of the non-uniformly distributed fin array is 14 % higher than that of the uniform array.

2.1.3. Triple tube heat accumulator

Sun et al. [57] simulated the effect of a staggered arrangement of circular fins on the inner shell of the annular space in a triple tubular accumulator on the melting rate of PCM. It was found that the staggered fin arrangement could increase PCM's melting rate by 37.2 %. Moreover, this arrangement can enhance the ability of the Reynolds number and Stephen number of the HTF to influence the melting process.

Xu et al. [58] added two circular tubes, one large and one small, in the annular space of the traditional triple shell-and-tube heat accumulator, which divided the PCM area into three small annular spaces. Four adjacent longitudinal fins with an included angle of 90° are placed inside. Then the internal array is rotated by 45° and placed in the middle area to form the middle array, and the middle array is rotated by 45° and placed in the outer area to form the outer array. After testing, this type of fin array can reduce the melting time of PCM by 71 %. The research of Zhao et al. [59] showed that a suitable fin array could reduce the error caused by the installation. By adding longitudinal fins in the annular space area of the triple tube heat exchanger and arranging the equal length longitudinal fins of the inner shell and the outer shell 45° apart, the influence of the rotation of the device on the melting time of PCM can be reduced to a certain extent. Compared with the model of Al-Abidi et al. [60], it can effectively shorten the solidification time by 13.8 %.

2.2. Spatial position distribution of fins

Unlike the arrangement and combination of fins on a two-dimensional plane, the spatial distribution of fins focuses on the on-demand distribution of fin positions in three-dimensional space to achieve better heat exchange effects. The spatial distribution design of the fins can also be classified according to the type of heat accumulator to be installed. See Table 4 for details on optimizing the spatial distribution of different fins.

2.2.1. Triple tube heat accumulator

Mat et al. [61] designed a novel triple-tube thermal storage unit with inner and outer fins in the annular space, which divides the annular space into non-fully spaced longitudinal forming chambers. The researchers designed various cases for simulation analysis and comparison in this study. It was found that the addition of inner and outer fins in the horizontal tube along the axial direction resulted in the formation of multiple small chambers, and the overall natural convection was divided into multiple small areas of natural convection. Since the overall natural convection was weakened, the liquefied PCM transferred to the upper part of the triple-tube accumulator was reduced, resulting in a more uniform temperature distribution within the accumulation unit. Mat et al. [62] also investigated the effect of installing longitudinal fins in the inner shell, outer shell, and both inner shell and outer shell of a triple tube heat exchanger on the melting process of PCM, respectively. They showed that the three installation methods did not result in significant differences in the melting rate of PCM, but the melting time of PCM was reduced by 43.3 % compared to the triple tube without fins.

The dominant mechanism of heat transfer at different positions of the same heat exchanger is not the same. Mahdi et al. [63] performed corresponding structural optimization for the fin structure in different regions of the horizontally placed triple tube heat exchanger. The simulation results show that during the heating process, the upper half of the PCM in the annular space is dominated by natural convection, and the lower half is dominated by thermal conductivity. In the case of the same specific volume of fins, increasing the number and length of fins in the lower half area and reducing the number and length of fins in the upper half area can improve the overall heat transfer level. This optimization makes the heat transfer capacity of the triple tube heat exchanger better than that of the nanoparticle-bound PCM using the same specific volume. The details of the relevant fin structure are shown in Fig. 8. Joybari et al. [64] tested the effect of different installation methods of longitudinal fins on the natural convection in space under the condition of simultaneous charge and discharge in triple shell-and-tube accumulators. The research results show that when the inner tube flows hot fluid and the outer tube flows cold fluid, one fin is installed in the upper half of the outer shell of the annular space, three fins with an included angle of 45° are installed in the lower half of the inner shell, and the natural convection effect is good. Mahmoud et al. [65] used the computational fluid dynamics method to optimize the structure of annular fins in a triple shell-and-tube heat accumulator placed vertically. An arrangement with one side down and one up sloping is better. The melting rate of the optimized PCM is increased by

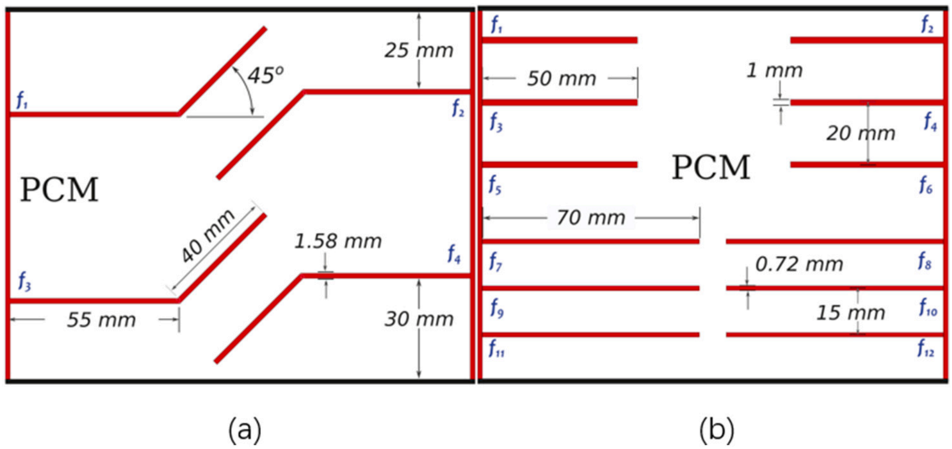

Fig. 7. Combined array of different fins in a rectangular thermal storage unit [55] (a) Combined arrangement of straight and angular fins (b) Combined arrangement of straight fins of different lengths and thicknesses.

88 %, and the charging rate is increased by 34 %.

2.2.2. Cylindrical shell and tube heat accumulators

Nie et al. [66] studied the spatial distribution of longitudinal fins in cylindrical shell-and-tube accumulators. They found that all longitudinal fins concentrated in the lower half area would increase the solidification time of PCM. This phenomenon became more evident with the increase in the number and length of fins.

Deng et al. [67] studied the effect of the length, number, and distribution of longitudinal fins on the heat transfer effect in a horizontally placed cylindrical shell-and-tube accumulator. They pointed out that the length, number, and distribution of fins are not unrelated, and the different optimal fin position exists for different lengths and numbers of fins. In this study, when the number of fins is less than 6, the fins are preferably placed in the circular area's lower half. When the number of fins exceeds 6, it is better to arrange them diagonally. When the number of fins is equal to 6, if the length of the fins is equal to 0.5 or 0.95, it is better to arrange them diagonally, and when the length of the fins is equal to 0.75, it is better to place them in the lower half area. Jyotirmay [68] and others adopted the vertical installation of rectangular fins on the vertical placed cylindrical shell-and-tube accumulator to enhance its heat storage performance. The primary method is to release the longitudinal space restrictions on natural convection without affecting its thermal conductivity, strengthening the natural convection effect. Installation of longitudinal fins will not affect its radial heat transfer uniformity.

Moreover, this study shows that: compared with the increase of HTF mass flow rate, the enhanced heat transfer effect produced by the increase of fluid inlet temperature is more significant. Qin et al. [69] studied the effect of identical fins on the melting time of PCM when the same fins were located at the top, middle, and bottom of the vertical cylindrical shell-and-tube accumulator. The melting time of the PCM was reduced by 3.6 % when the fins were placed on the bottom compared to when the fins were placed on the top. It indicates that the different arrangements of the identical fins have a specific influence on the heat exchange capacity of the heat storage unit.

Wang et al. [70] established a three-dimensional cylindrical shell-and-tube phase change heat storage device model. By simulating the case of adjacent angles of three rectangular fins on the inner tube surface, it was found that when the fin angle was between 60° and 90°, the natural convection effect was the best, and the melting rate was the fastest. Liu et al. [71] designed six Y-shaped fins arrangements with non-uniform distribution along the direction of gravity. Through the simulation comparison in the established cylindrical shell and tube accumulator model, it is found that the time required to melt the PCM in the lower half region is almost half of the total melting time. Among the six types of fins, two short straight fins in the upper half area and 2 Y-shaped fins in the lower half area can minimize the total melting time. Compared with four long straight fins, Melting time was reduced by 21.5 %. Huang et al. [72] arranged six tree-branch fins in a horizontally placed shell-and-tube heat accumulator according to a particular gradient, in which two tree-shaped fins were placed in the upper half area, and four tree-shaped fins were placed in the lower half area. Compared with the evenly distributed branch-shaped fins, the branch-shaped fins distributed according to a specific gradient can enhance the heat transfer in the lower part and prolong the natural convection in the upper part. Compared with the uniform distribution, the melting time of the gradient distribution is shortened by 9 %. However, the solidification time is increased by 57.4 % due to the non-uniformity of the temperature distribution. The details of the relevant fin structure are shown in Fig. 9.

Khan et al. [73] studied the effect of three longitudinal fin orientations with the same angle on the melting process of PCM in a horizontally placed shell-and-tube accumulator. Through simulation and experiments, the change process of the angle between the longitudinal fins in the upper half area and the horizontal direction gradually changes from 90° to 0°. The results show that the best melting performance of PCM is obtained when the longitudinal fins in the upper half are at an angle of 30° to the horizontal. Yu et al. [74] studied the positional distribution of longitudinal fins in a horizontally placed shell-and-tube accumulator. Through RSM design optimization, it was found that the fin angle arrangement with a gradient distribution has better melting performance. Since this arrangement can synergistically enhance natural convection and heat conduction, the PCM melts faster and has a more uniform temperature distribution. The melting time of PCM was reduced by 30.5 % compared to the uniform distribution.

Tang et al. [75] studied the non-uniform spatial distribution of longitudinal fins. For the horizontally placed cylindrical shell-and-tube heat accumulator, the fins are all concentrated in the lower half area, which could synergistically enhance natural convection and heat conduction. When the adjacent angle between the fins is 25°, and the length of the fins is 40 mm, the melting time of PCM is shortened by 83.9 %, and the heat storage density is increased by 466 % compared with those without fins. Saldi et al. [76] studied the influence of the inclination angle of two annular fins on the melting of PCM in the vertical cylindrical shell and tube heat storage device. They found the maximum heat transfer rate when the inclination angle of the two fins is −20°. The slightly worse heat transfer rate is the case where one angle is 20° and the other is −20°. Parsazadeh et al. [77] studied the effect of the inclination of annular fins on the melting process of PCM in vertical shell-and-tube accumulators. The RSM method was used to simulate and analyze. They found that local vortex would be generated under the fins with

Table 4
Optimization of the spatial position distribution of the different fins.

Accumulator type	Reference	Fin configuration	Research content	Finding description
Triple shell and tube	[61]	Longitudinal fins	Numerical/ melting	Natural convection in small spaces with multiple zones is more evenly distributed than natural convection in large spaces.
	[62]	Longitudinal fins	Numerical/ melting	There is no significant difference in the melting rate of PCM when the fins are installed in the inner shell, the outer shell, and the inner and outer shell, respectively.
	[63]	Longitudinal fins	Numerical/ melting/ solidification	Increasing the number and length of fins in the lower half and reducing the number and length of fins in the upper half enhances heat transfer.
	[64]	Longitudinal fins	Numerical/ melting/ solidification	1 fin in the upper half of the annular housing and 3 fins at an angle of 45° to each other in the lower half of the inner housing provide better natural convection.
	[65]	Annular fins	Numerical/ melting	Depending on the different flow directions of the HTF, an arrangement in which the housing fins are angled downwards on one side and upwards on the other in the annular space is preferable.
Cylindrical shell and tube	[66]	Longitudinal fins	Numerical/ solidification	The concentration of all the fins in the area's lower half increases the PCM's solidification time.
	[67]	Longitudinal fins	Numerical/ melting	There is a link between the various parameters of the fins, and there are optimal positions for different lengths and numbers of fins.
	[68]	Longitudinal fins	Experimental/ melting/ solidification	The longitudinal installation enhances the natural convection effect and does not affect the radial heat transfer uniformity.
	[69]	Annular fins	Numerical/ melting	Due to natural convection, the melting time of the PCM is 3.6 % less when the fins are placed at the bottom than when they are placed at the top.
	[70]	Longitudinal fins	Numerical/ melting	Natural convection works best, and melting rates are fastest when the fin angle is between 60° and 90°.
	[71]	Y-shaped fins	Numerical/ melting	The time required to melt the lower half of the PCM is half of the total melting time.
	[72]	Tree fork fins	Numerical/ melting/ solidification	In contrast to a uniform distribution, a gradient distribution enhances heat transfer in the lower part and prolongs natural convection in the upper part.
	[73]	Longitudinal fins	Numerical/ experimental/ melting	The best melting performance is achieved when the longitudinal fins in the upper half of the area are at an angle of 30° to the horizontal.
	[74]	Longitudinal fins	Numerical/ melting	The arrangement of the fins with a gradient distribution of the angle has better melting performance.
	[75]	Longitudinal fins	Numerical/ melting	Natural convection and heat transfer are synergistically enhanced when the fins are all concentrated in the lower half of the area.
	[76]	Annular fins		

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Table 4 (*continued*)

Accumulator type	Reference	Fin configuration	Research content	Finding description
			Numerical/ melting/ solidification	The maximum heat transfer rate is achieved when both fins are tilted at an angle of −20°.
	[77]	Annular fins	Numerical/ melting	Local vortices are generated under the positively inclined fins, enhancing the convection effect and facilitating the melting of the PCM.
Multi-tube single-shell	[78]	Annular fins	Numerical/ melting	When placed vertically, the annular fins impede the natural convection of the liquid phase PCM.
	[79]	Longitudinal fins	Numerical/ melting	The horizontal-vertical cross-distribution of the fins reduces melting time by 42.1 %.
Rectangular	[80]	Longitudinal fins	Numerical/ melting	The PCM has a better melting rate when the fins are inclined at 90° to 115°.

positive inclination angles. The vortex could enhance the convection effect, which is conducive to melting PCM. Moreover, in the case study, a positive tilt angle of 35° had the shortest charging time.

2.2.3. *Multi-tube single-shell heat accumulator*

Kim et al. [78] tested the heat transfer effect of a multi-tube single-shell heat accumulator with annular fins installed in the inner tube when placed horizontally and vertically. They found that when the heat accumulator is placed horizontally, the heat storage time is shortened by 10 % compared to when it is placed vertically. This is because when the heat accumulator is placed horizontally, more vigorous natural convection can be generated than when it is placed vertically. The reason is that the annular fins in the heat accumulator, when placed vertically, will hinder the natural convection of the liquid phase PCM.

Dandotiya et al. [79] studied the effect of the arrangement of longitudinal rectangular fins on the heat transfer effect in a multi-tube heat exchanger. The model is a vertical cylindrical shell containing four vertical cylindrical tubes placed at the vertices of a square. Case one uses two longitudinal rectangular fins to connect the four cylindrical tubes diagonally in pairs, and the second is to divide the square area into four small square areas equally by two longitudinal fins. Studies have shown that diagonally connected cylindrical tubes with fins can reduce melting time by 26.3 %, while horizontal and vertical cross-distribution can shorten by 42.1 %.

2.2.4. *Rectangular accumulators*

Borahel et al. [80] studied the inclination of rectangular fins on the vertical wall of a rectangular accumulator. Through simulation, it was found that PCM had a better melting rate when the inclination of the fins was between 90° and 115°.

3. Choice of the number of fins

There is an optimal number of fins for a specific heat exchange environment. It is also classified according to the phase change heat storage units with fins installed. The details of the optimal number of fins for different heat accumulators are shown in Table 5.

3.1. *Cylindrical shell and tube heat accumulators*

Yang et al. [81] studied the effect of annular fins on the thermal performance of shell-and-tube heat storage units. Using the RSM method, they obtained the optimal number of annular fins in the Latent Heat Thermal Energy Storage (LHTES) unit. The studies have shown an optimal number of fins for a specific heat exchange environment. With the increase in the number of fins, although the heat conduction is enhanced, which is beneficial to the movement of the phase interface, the occurrence and development of local natural convection are limited due to the smaller and smaller space of the cavity (the interval between the fins). In addition, since the total volume of the fins is fixed, the increase in the number of fins will lead to a decrease in the thickness of the fins, an increase in the heat transfer resistance, and a decrease in the heat transfer effect. Therefore, too many fins weaken the heat transfer enhancement. See Fig. 10(a) for the effect of the specific number of fins on the melting time. It can be seen from this figure that when the number

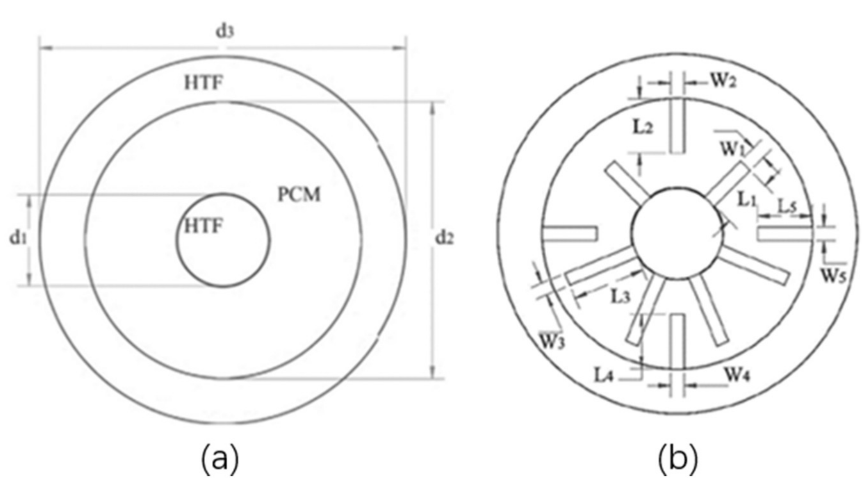

Fig. 8. Design of fin distribution based on the dominant heat transfer mechanism in different zones [63] (a) Finless triple tube heat exchanger (b) Triple tube heat exchanger with an optimized fin space distribution.

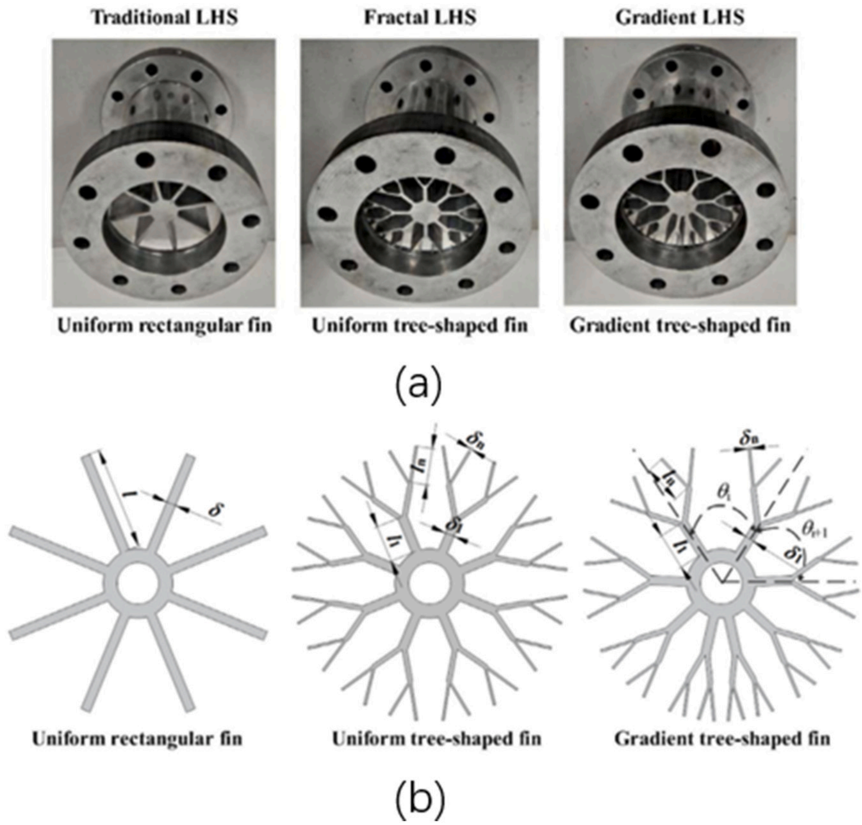

Fig. 9. Design of the gradient distribution of the forked fins [72] (a) Photographs of the three different latent heat storage units; (b) Cross-sections of the three different fins.

of fins is less than 31, increasing the number of fins can reduce the melting time of PCM. However, when the number of fins is greater than 31, the melting time increases with the number of fins, so the optimal number of fins in this cylindrical shell and tube accumulator is 31.

Nobrega et al. [82] conducted an experimental study on the heat transfer of longitudinal fins in shell-and-tube accumulators. They pointed out that the optimal number of fins and fin width are the key factors affecting the heat transfer effect. The shell-and-tube accumulator fitted with five longitudinal fins has 2.3 times the solid fraction at the end of the solidification process compared to finless tubes. When six longitudinal fins were installed, the solid fraction at the end of the solidification process was only 2.26 times that of the finless tube. Details of the finned tubes used in this experiment are shown in Fig. 10(b). Shinde et al. [83] studied the effect of fins' number, height, and thickness on the solidification process of PCM in medium-temperature shell-and-tube accumulators. The outlet temperature of HTF and the solid phase fraction of PCM were simulated, and it was found that the latent heat storage system had the best heat transfer performance when the number of longitudinal fins was 24, the thickness was 1 mm, and the height was 7 mm. Tao et al. [84] simulated the effect of the number of fins added in the lower half of the horizontally placed shell-and-tube accumulator on the melting process of PCM. Through comparative analysis, it is found that when the number of fins in the lower half area is 7, the heat transfer performance of the heat accumulator is the best. When the number of fins continues to increase, the heat transfer performance of the system begins to decline.

3.2. Rectangular heat accumulator

Amagour et al. [85] designed a rectangular phase change accumulator, which uses plate-shaped fins inside the accumulator, HTF pipes run through the plate-shaped fins, and PCM is placed in the interval. Through simulation research, it is found that there is an optimal number of plate-shaped fins. When the number of fins is higher than 13, increasing the number of fins will not significantly improve the heat transfer effect of the accumulator and will increase the cost consumption. Xu et al. [86] studied the optimal number of plate-shaped fins installed on the vertical wall of a rectangular heat accumulator. Through genetic algorithm and Computational Fluid Dynamics (CFD) simulation experiments, it was found that the optimal number of plate-shaped fins depends on the length of the fins, and different lengths of fins correspond to different optimal numbers of fins.

3.3. Triple tube heat accumulator

Jmal et al. [87] designed a special triple-tube accumulator with one end closed, HTF flows in from the inner tube, and out from the outer tube, PCM is placed in the annular space. The effect of adding annular fins to the PCM region on solidification was investigated. The results show that there is an optimal number of fins N = 9. Continuing to add fins, the PCM annular space will limit the heat convection, and the enhancement effect on the solidification process is negligible. Abdulateef et al. [88] installed triangular fins in the traditional triple tube heat accumulator, and the fins are located in the outer shell of the annular space. Simulation studies found that the optimal number of fins is 8. In the optimal aspect ratio and fin length, PCM's melting and solidification times were shortened by 163 min and 425 min, respectively, compared to the triple-tubular accumulator without fins.

4. Structural optimization of fins

4.1. Fin size optimization

There are also optimal sizes for the fins in the heat accumulator, and

Table 5
Optimum number of fins present for different heat accumulators.

Accumulator type	Reference	Fins configurations	Research contents	Finding description
Cylindrical shell and tube	[81]	Annular fins	Numerical/ melting	The optimum number of fins is 37. Continued increases will result in a reduction in fin thickness, an increase in thermal resistance, and a reduction in heat transfer.
	[82]	Longitudinal fins	Experimental/ solidification	The optimum number of fins and the optimum fin width are critical factors in heat transfer effect.
	[83]	Longitudinal fins	Numerical/ solidification	This latent heat storage system has the best heat transfer performance when the number is 24.
	[84]	Longitudinal fins	Numerical/ melting	The optimum number of fins is 7. As the number of fins increases, heat transfer performance begins to decline.
Rectangular	[85]	Plate fins	Numerical/ melting/ solidification	When the number of fins is higher than 13, increasing the number of fins will not significantly improve the heat transfer effect.
	[86]	Plate fins	Numerical/ melting/ solidification	The optimum number of plate fins depends on the length of the fins.
Triple shell and tube	[87]	Annular fins	Numerical/ solidification	There is an optimum number of fins, N = 9. If the number of fins continues to increase, the PCM annular space will limit the thermal convection.
	[88]	Triangular fins	Numerical/ melting/ solidification	The optimum number of fins is 8 for the best melting and solidification results.

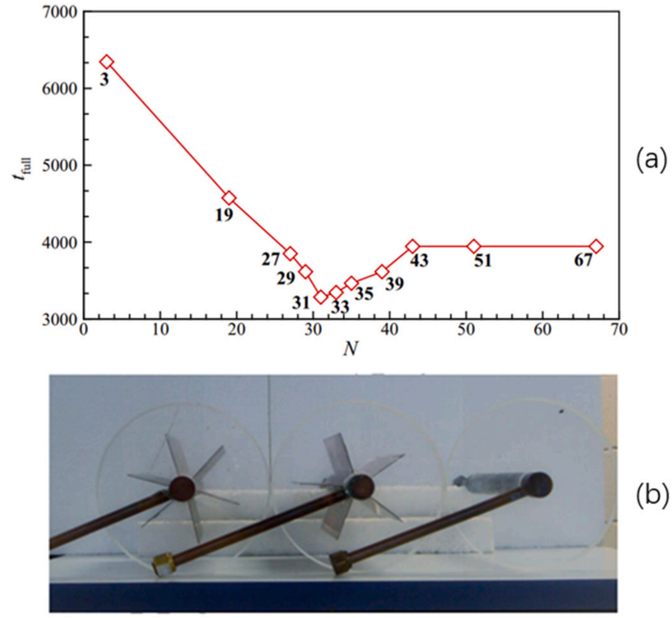

Fig. 10. (a) Effect of a specific number of fins on melting time in the study by Yang et al. [81] (b) Latent heat storage unit with different number of fins used in the experiment by Nobrega et al. [82].

each optimal size parameter has a corresponding design concept. Different optimization methods can be used for different design concepts, and these methods can be roughly divided into the following categories.

4.1.1. Size optimization based on multi-factor comprehensive relationship

Mostafavi et al. [89] pointed out that in a cylindrical shell-and-tube thermal storage system, even a tiny fin can bring a significant gain, and the gain from continuing to increase the fin size is small. On the contrary, it will cause a reduction in heat storage and an increase in manufacturing costs. To further understand the relationship between fin size and heat transfer rate, Mostafavi et al. [90] derived the temperature distribution equation of fins. The results showed that: in a given time, the optimal fin size is a function of the thermal conductivity of the fins. Its physical expression is that in a given total heat transfer time, the final heat transfer effect of the fins is the combined effect of two heat transfer modes: PCM absorbs heat directly from the fins, and PCM absorbs heat from the heat source pipe. Both should be considered when designing the fin size. Lissner et al. [91] studied the influence of fin size on heat transfer rate and heat storage capacity, they found that under the condition of fixed fin volume fraction, there is an optimal fin height, which can provide the most optimal heat transfer rate. The smallest possible fin spacing and thickness results in a more uniform temperature distribution and the optimal design of the fin size also depends on the flow conditions.

Wang et al. [92] studied the effect of circular fin parameters on the heat transfer performance of the shell-and-tube heat storage device. Unlike previous studies on the change of a single parameter, the combined effect of the fin spacing and the change of the inner and outer diameter of the tube was considered in the study. They found that the outer diameter of the tube has a more significant impact on the heat storage rate. When the fin spacing is small, larger fin height and width can improve the heat exchange performance of the thermal storage device. Ghalambaz et al. [93] optimized the size of the helical fins in a vertically placed cylindrical shell-and-tube accumulator. By studying 12 different cases, the effects of different fin heights, thicknesses, and helical pitches on the accumulator charging process were explored. According to different optimization factors, the overall charging time of the accumulator can be reduced by 6 %–42 %, and the heat storage rate can be increased by 6 %–63 % compared with the same volume of straight longitudinal fins. Biwole et al. [94] studied the relationship between the number and size of rectangular fins in a rectangular heat storage unit. They found that increasing the number of fins can achieve better melting characteristics. However, it will shorten the temperature stabilization time of the heat exchange wall. The same effect can be achieved by using thinner and longer fins, and the temperature of the heat exchange wall can be stabilized for a longer time.

4.1.2. Optimization based on natural convection heat transfer mechanism

Vogel et al. [95] established a mathematical model to study the effect of natural convection in vertical finned tube accumulators. The results show that when the tube spacing is small, and the fin volume fraction is large, the effect of natural convection can be ignored. Otherwise, natural convection will significantly influence the melting process. Gurturk et al. [96] studied the relationship between the heat transfer surface area and natural convection of a vertical cylindrical longitudinal finned shell and tube accumulator. This study showed that the heat transfer surface area is not as large as possible. Excessive heat exchange surface area is not conducive to natural convection due to surface friction coefficient. See Fig. 11 for specific details of its case studies. It shows that the heat transfer surface area of Fin 1_C is 2.5 times that of Fin 4, but compared with the addition of Fin 1_C, the melting process of PCM by adding Fin 4 is accelerated by 5 %. It shows that the heat exchange surface area is not as large as possible. Although Fin 5 and Fin 4's heat exchange surface area is the same, by inverting Fin 5 so that the fin heat exchange surface is located in the upper half of the pipe, the acceleration of the melting process of PCM by Fin 4 is still 5 % faster than that of Fin 5. For the same reason, natural convection mainly occurs in the upper half. Fin 4 has only half of the heat exchange surface in the upper half, while the inverted Fin 5 has all the heat exchange surfaces in the upper half. Due to large-area friction, the development of natural convection is seriously affected.

4.1.3. Optimization of unequal-length fins based on the dominant heat transfer mechanism in each region

Zhu et al. [97] studied the effect of unequal-length circular fins on the vertical cylindrical shell and tube thermal storage system. The overall melting and solidification rates of PCM have been improved to a certain extent by structural optimization of fin lengths with different installation heights. For example, increasing the length of the bottom fins can improve the slow heat transfer at the bottom. Partitioning the heat exchanger as a whole helps to reduce heat resistance. Increasing the length of the end fins can effectively shorten the solidification phase transition time. The relevant case studies are shown in Fig. 12. Najim et al. [98] also studied the length optimization of circular fins in a vertically placed cylindrical shell and tube heat exchanger, and the results showed that as the flow direction of the HTF changed, the excellent design of fin size could lead to a faster charging heat rate and more uniform temperature distribution. In this simulation, the non-uniform-sized fin distribution can reduce the melting time of PCM by 10.4 % and increase the heat storage rate by 9.3 % compared to the uniform-sized fin distribution. However, considering that non-uniformly sized fins will also generate higher flow resistance and affect the natural convection of the melting process. Continuing to increase the number of fins may negatively affect the melting process, reducing the melting rate.

Joshi et al. [99] aimed to maximize the heat storage capacity as the optimization goal. They studied the vertical cylindrical shell-and-tube heat accumulator to minimize the fin volume ratio without affecting its melting rate. The final optimization result is also the fins of unequal

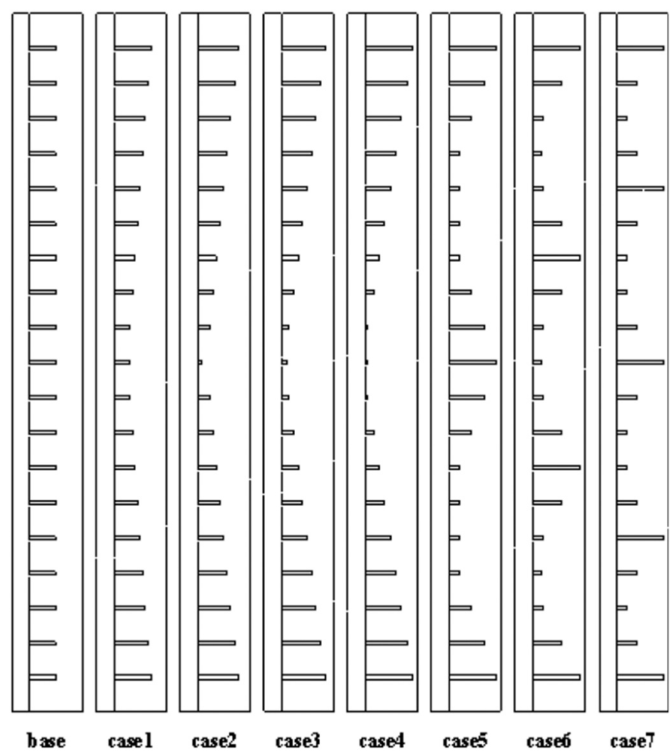

Fig. 12. Experimentally listed structure of unequal fin cases [97] (the BASE case is the most basic equal-length annular fin).

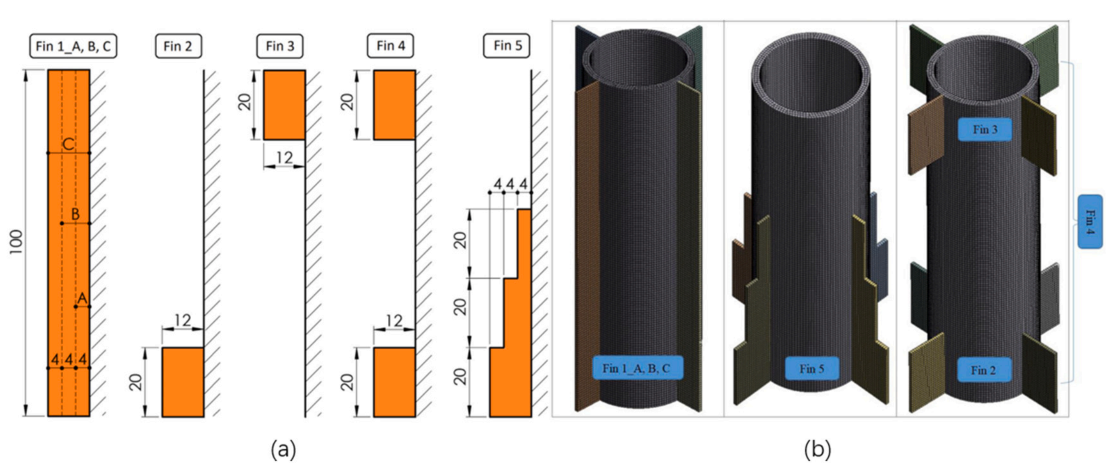

Fig. 11. Five cases with different sizes and locations of heat transfer surface area [96] (a) Fin heat transfer surface area setting (b) Mesh stereoscopic structure of the numerical model.

length. After reducing the volume ratio of the fin to PCM by half in this experiment, the melting fraction of PCM can be improved by 6.34 %. Through experiments, Singh et al. [100] studied the unequal-length annular fins in cylindrical shell-and-tube accumulators. They found that along the flow direction of HTF, reducing the length of the fins can effectively reduce the melting time of PCM. The unequal length fin arrangement can reduce the melting time of PCM by 30 % compared to no fins. Ji et al. [101] studied the effect of the lengths of the upper and lower rectangular fins on the vertical wall of a rectangular heat accumulator on the melting process of PCM. Studies have shown that when the ratio of the length of the upper fin to the length of the lower fin is 0.25, the total melting time of PCM can be saved by about 25 %.

4.1.4. Optimization is based on the height, length, and thickness of the fin itself

It is also an idea to start from the influence of the fin itself on the heat transfer effect. Details of the optimization studies in terms of fin height, length, and thickness are shown in Table 6.

Singh et al. [102] studied the effect of fin height on the melting process in shell-and-tube accumulators and pointed out that a suitable fin height benefits natural convection. When the height of the fins is equal to 0.75 times the difference between the tube's outer diameter and the inner diameter, the melting time of the PCM is reduced by 47 %. Pakrouh et al. [103] analyzed the heat transfer characteristics of the structure parameters of the pin-fin radiator, including the number of fins, thickness, height, etc. The research shows that increasing the number, thickness, and height of pin-shaped fins can accelerate the melting process of PCM, reduce the actual wall temperature of electronic devices, and protect the chip.

Borhani et al. [104] studied the effect of helical fin size parameters on the melting time of PCM in shell-and-tube accumulators. They pointed out that when the fin spacing is constant, the decrease in fin height caused by increasing fin thickness will increase the melting time. Increasing the fin's pitch and height will reduce the melting time when the fin thickness is constant. For spiral finned shell and tube accumulators, the height of the fins is one of the decisive factors. Asgari et al. [105] studied the effect of branch-shaped fins on the solidification process of PCM in cylindrical shell-and-tube accumulators, and their results showed that thinner fins could improve the solidification rate of PCM.

Fan et al. [106] studied the relationship between annular fin height and PCM melting in spherical containers. Through indirect experiments, the system's thermal performance increased with the increase of fin height. At the highest fin height, the melting time of PCM is reduced by 30 %. Hosseini et al. [107] studied the effect of the height of rectangular fins in the shell-and-tube accumulator on the melting process of PCM. They found that when the fins are high, the eddy generated by convection cannot disappear, but a larger eddy will be formed, while the eddies formed by the fins with smaller heights are also smaller. Increasing the height of the fins does increase heat transfer, but the heat storage decreases, so there is an optimum fin height. Modi et al. [108] studied the effect of longitudinal fin size parameters on the melting process of PCM in a horizontally placed shell-and-tube accumulator. Simulation studies showed that when the fin volume is the same, installing a small number of long fins is better than installing a large number of short fins, and thinner fins have better heat transfer performance.

Table 6
Optimization results based on the height, length and thickness of the fin themselves.

Accumulator type	Reference	Fins configurations	Research contents	Finding description
Cylindrical shell and tube	[102]	Longitudinal fins	Numerical/ melting	The right fin height is beneficial for natural convection.
Rectangular	[103]	Pin fins	Numerical/ melting	Increasing the thickness and height of the pin fins accelerates the melting of the PCM.
Cylindrical shell and tube	[104]	Spiral fins	Numerical/ melting	Fin height is one of the decisive factors.
	[105]	Tree fork fins	Numerical/ solidification	Thin fins improve the solidification rate of PCM compared to thick fins.
Spherical	[106]	Annular fins	Numerical/ melting	The thermal performance of the system increases with the increase of fin height.
Cylindrical shell and tube	[107]	Longitudinal fins	Numerical/ melting/ solidification	A vortex forms when the fins are higher, improving heat transfer.
	[108]	Longitudinal fins	Numerical/ melting	A small number of long fins are more conducive to melting than a large number of short fins.

4.2. Fin shape optimization

When the heat exchange conditions change, the original fin shape and structure must be further optimized to achieve the best heat transfer enhancement effect in the new heat exchange conditions. In the current fin shape optimization, many related optimizations are carried out for branch-shaped, Y-shaped and V-shaped fins. See Table 7 for details.

4.2.1. Y-shaped fins

Sciacovelli et al. [109] used CFD and RSM to study the optimal structural parameters of Y-fins in shell-and-tube heat exchangers. The novelty of this study is that shape optimization is carried out under transient conditions, and the optimization parameters are different for different heat transfer times. The optimal bifurcation angle of the Y-fin with single or double bifurcation varies with the running time. The double-bifurcated Y-fin has better heat transfer performance than the single-branched Y-fin. When the operating time of the LHTES device reaches the 2000 s, the double bifurcated Y-fins improve the heat transfer performance by 24 % compared to the single bifurcated Y-fins. The related Y-fin structure is shown in Fig. 13.

Li et al. [110] studied the effect of the angle between the two branches of a single Y-fin in a rectangular accumulator and the horizontal plane on the melting performance of PCM. The results show that the influence on the melting process is mainly concentrated in the early melting stage for equal-length fins with different angles. The uneven melting caused by convection cannot be improved in the later melting stage. After simulation optimization, the optimal angle is that one branch is horizontal and the other has an angle of -15° with the horizontal plane. Alizadeh et al. [111] used RSM to optimize the shape of Y-fins in vertical shell-and-tube accumulators. Through optimization, Y-shaped fins are gradually transformed into leaf-shaped fins. The simulation results show that with the increase of fin length and fin bifurcation angle, the system's average temperature is lower than before optimization, the solid phase fraction is higher, and the heat transfer during solidification is significantly enhanced.

Table 7
Fin shape optimization.

Accumulator type	Reference	Fins configurations	Research contents	Finding description
Cylindrical shell and tube	[109]	Y-shaped fins	Numerical/ melting	For Y-shaped fins, the optimum bifurcation angle decreases with increasing running time.
Rectangular	[110]	Y-shaped fins	Numerical/ melting	The influence of the different angles of the equal-length fins on the melting process is mainly concentrated in the first half of the process.
Cylindrical shell and tube	[111]	Y-shaped fins	Numerical/ solidification	Increasing the fin length and bifurcation angle will increase the solidification rate.
	[112]	Fractal fork fins	Numerical/ melting	Improving the fin's length ratio and thickness index plays a vital role in achieving an optimum balance between natural convection and thermal conductivity during the melting process.
	[113]	Tree fork fins	Numerical/ melting	There is an optimum length and width ratio for tree fins.
	[114]	Tree fork fins	Numerical/ experimental/ melting	The larger the bifurcation angle, the better the melting performance of the forked fins.
	[115]	Tree fork fins	Numerical/ melting	The non-uniform distribution of the PCM enables a significant reduction in the melting time of the PCM.
	[116]	V-shaped fins	Numerical/ melting/ solidification	The optimal fin branching direction does not affect the heat storage capacity, but can significantly increase PCM solidification rate.
Triple shell and tube design	[117]	V-shaped fins	Numerical/ solidification	The increase in the angle of the V-shaped fins increases the rate of heat release of the accumulator.

4.2.2. *Branch-shaped fins*

Yu et al. [112] used RSM to optimize the shape of fractal tree fins. Compared with the fractal tree fins before optimization, the melting time of PCM was reduced by 26.7 %, and the average heat storage rate was increased by 45.4 %. The study points out that improving the length ratio and thickness index of fractal tree fins plays a vital role in achieving the best balance between natural convection and heat conduction during the melting process. Zhang et al. [113] conducted a parametric study on tree-branch fins. In this study, they pointed out that in the shell-and-tube heat storage device, the optimal length ratio of tree-shaped fins is about 1.3, and the optimal width ratio is about 1. Peng et al. [114] conducted both simulation and experimental study on the melting performance of the branch-shaped fins in the shell-and-tube accumulator. Their results showed that PCM's melting rate increased with the branch-shaped fins' bifurcation angle. That is, the larger the bifurcation angle, the better the performance of heat transfer of the branch-shaped fins.

Huang et al. [115] simulated and optimized the branch-shaped fins in the horizontally placed shell-and-tube heat accumulator, using RSM to optimize the uniformly distributed branch-shaped fins in the annular space into a non-uniformly distributed hierarchical tree Fork shape. The optimized fin structure is a rectangular straight fin with a zero-layer branch and two Y-shaped fins with a first-layer branch in the upper half area. The lower half area consists of two bifurcated fins with two layers of branches and one bifurcated fin with three layers of branches. More layers in the structure mean more branches. Compared with the evenly distributed tree-branch fins, this structure shortens the melting time of PCM by 41.1 %. See Fig. 14 for details on the structure of the layered fins.

4.2.3. *V-shaped fins*

Lohrasbi et al. [116] optimized the shape of V-shaped fins in the cylindrical shell-and-tube accumulator. The simulation took the maximum solidification rate and heat storage capacity as the optimization goals, and they found that the optimal fin branch direction does not affect the heat storage ability. However, it can significantly improve the solidification rate of PCM. In addition, the increase in the thickness of the fins can accelerate the heat release, but the heat storage capacity will decrease. The length of the fins increases the heat release rate to a greater extent, and the decrease in heat storage can be ignored. Sheikholeslami et al. [117] studied the included angle of the V-shaped fins in the shell-and-tube heat storage unit. The results showed that the larger the included angle of the V-shaped fins within a specific range, the faster the heat release rate of the heat accumulator.

5. Design method

For different design ideas, there are corresponding design methods. According to different design ideas, four types of optional design methods are commonly used in actual fins design.

5.1. *Topology optimization*

The topology optimization method is often used in the shape design of fins, and its essence is the distribution method of materials in the design domain. From a certain point of view, it can be understood as a

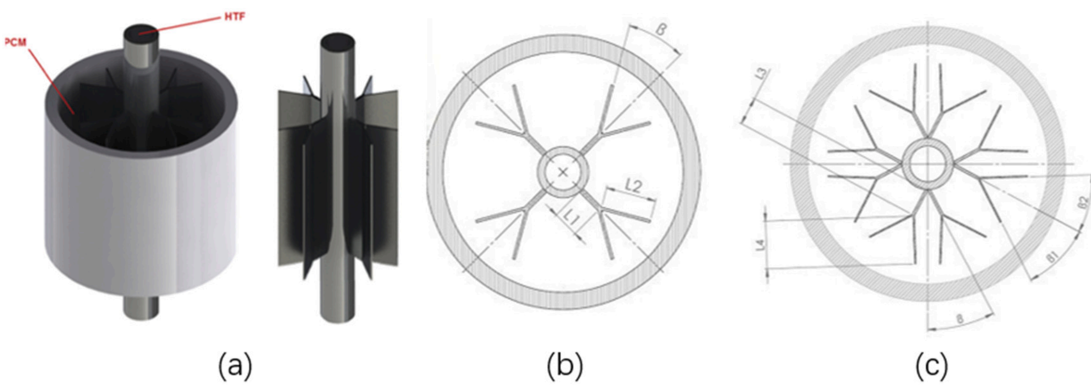

Fig. 13. Shape optimization design of Y-shaped fins in shell and tube thermal storage units [109] (a) LHTES system (b) Single bifurcated Y-shaped fins (c) Double bifurcated Y-shaped fins.

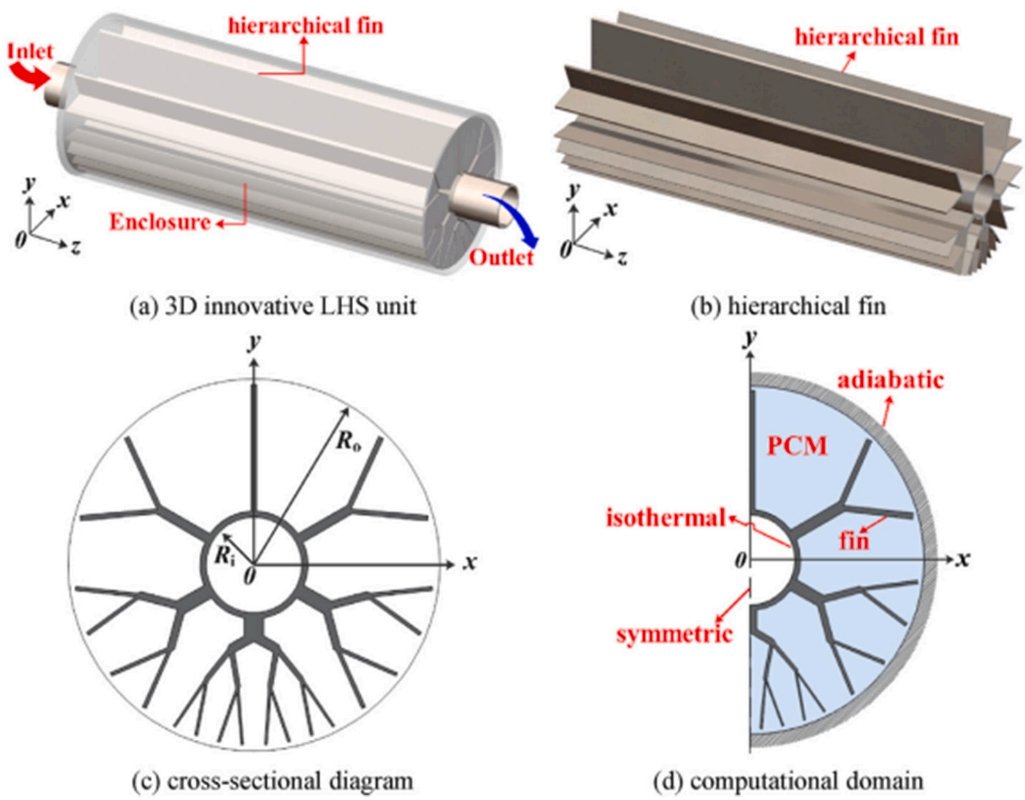

Fig. 14. Schematic diagram of a horizontal shell and tube thermal storage unit with layered forked fins after optimization using RSM [115].

more thorough shape optimization. The topology optimization does not require more constraints and only needs to be carried out under loading boundaries and internal connections, and the design freedom is high. In the optimization process, each node of the fin structure has a corresponding design variable, and the variable size distribution of all design nodes ultimately determines the shape of the fin. The specific operation steps are as follows.

(1) Define the objective function, such as using the average temperature in the design domain to characterize the heat transfer capacity of the fins. Its mathematical expression can be written as:

$$Max \Big/ Min\ Z(T(x), x) = \int_{\Omega} T(x)dx \Big/ V \tag{3}$$

where x is the position of the design variable, $T(x)$ is the temperature value of the node where the design variable is located, V is the volume of the design domain, and Z is the average temperature of the material in the design domain. The mathematical expression means that within the specified charging and discharging time, under the same heat source or cold source conditions, through topology optimization, after the specified time, the fins with higher or lower average temperatures in the design domain are considered to have higher heat transfer capabilities.

(2) Determine the constraints.

$$Subject\ to \begin{cases} F(T(x), x) = 0 \\ \int_{\Omega} \rho_s dv \leq \Phi \int_{\Omega} dv \\ x \in X = \{x_1, x_2, x_3, \dots x_n\} \end{cases} \tag{4}$$

where the first constraint that the temperature change for each design variable must satisfy the derived set of heat transfer equations, the second constraint ensures that the optimization process does not affect the total heat storage of the system by limiting the amount of heat transfer material used. In its expression, ρ_s represents the density of the design variables, the magnitude of which reflects the proportion of different materials in each design variable. Φ is the system's predetermined total volume proportion of heat transfer materials.

(3) Using the variable density interpolation method [118] to distinguish the PCM and the fin structure in the design domain, that is, the high thermal conductivity material (HCM). The core idea is to define the density of the design variable as a continuous variable with a value range of 0–1, where 0 represents PCM, and 1 represents HCM. Its definition is as follows.

$$\begin{aligned} K(x) &= K_{PCM} + \rho_S^P(K_{HCM} - K_{PCM}) \\ \rho(x) &= \rho_{PCM} + \rho_S(\rho_{HCM} - \rho_{PCM}) \\ C_P(x) &= C_{P_{PCM}} + \rho_S(C_{P_{HCM}} - C_{P_{PCM}}) \\ L(x) &= (1-\rho_S)L \\ \alpha(x) &= \alpha_{PCM}(T) + \rho_S(\alpha_{HCM} - \alpha_{PCM}(T)) \end{aligned} \tag{5}$$

The materials used in practical industry are either heat storage materials or heat transfer materials, that is, a distribution of either 0 or 1, so it is necessary to use the median penalty strategy commonly used in the field of topology optimization [119]. By defining the intermediate penalty index P (Generally, the value is 3–5), a penalty index is added to the thermal conductivity calculation to reduce the intermediate value's heat transfer contribution and the error.

(4) Considering the influence of natural convection, the Brinkman term is introduced, and permeability $\alpha(x)$ is defined to distinguish between solid and liquid phases.
(5) The Method of Moving Asymptotes (MMA) can be used for optimization [120]; in the iterative process, the solution equation system is approximated so that the result is closer to the real solution. At the same time, using this method will also introduce additional parameters to help the solution of the equation system to be closer to the real solution [121].
(6) Perform the filtering operation. Define the radius of the filter area, and use the average value of the design variables of all nodes within the range as the variable value of the node. This processing can effectively avoid the "cliff-like" density mutation between adjacent nodes and effectively enhance the stability of the heat transfer problem solution, to avoid the situation of failure to converge to the greatest extent. The operation is to solve partial differential equations of the Helmholtz type [122]. Its mathematical expression is as follows.

$$\nabla\cdot(-r_f\nabla\rho_f) + \nabla\rho_f = \rho_s \tag{6}$$

where r_f is the filter radius size and ρ_f is the filtered design variable, it is worth noting that after each iteration, the equation needs to be solved again to update the design variables. Moreover, the size of the filter radius should be selected based on experiments. If it is too large, the calculation time will be costly, and if it is too small, the fin structure may not be fully optimized.

(7) Perform post-processing operations. To clarify the fin shape boundary, a projection function [123] is usually introduced to post-process the intermediate values. The function expression is as follows.

$$\rho_P = \frac{tanh(\beta\eta) + tanh(\beta(\rho_f - \eta))}{tanh(\beta\eta) + tanh(\beta(1-\eta))} \tag{7}$$

In this mathematical expression, η represents the threshold of the mapping. In general, 0.5 can be chosen, β controls the steepness of this mapping, ρ_f is the filtered design variable obtained previously, and the resulting ρ_P is the final design variable required.

5.2. Multi-objective response surface method

RSM is commonly used in the structural optimization and spatial distribution of fins. It is an efficient mathematical-statistical method that works well when the optimization objective (target response) is controlled cooperatively by multiple design variables (optimization factors). Compared with similar artificial neural network algorithms, particle swarm optimization, and genetic algorithms, using RSM can significantly reduce the workload, save computational time and experimental costs, and the actual operation is relatively simple [124,125].

Fin optimization using RSM can be divided into two stages. The first stage utilizes second-order polynomials to evaluate the effect of design variables on the optimization objective. The mathematical expression is as follows [126].

$$y = a_0 + \sum_{i=1}^{n} a_i x_i + \sum_{i=1}^{n} a_{ii} x_i^2 + \sum_{i=1}^{n}\sum_{j=1}^{n} a_{ij} x_i x_j, i < j \tag{8}$$

In this expression, a is the tuning parameter and x is the design parameter.

In the second stage, the optimal desirability parameters are calculated. The expected value of this parameter is between 0 and 1. When the desirability parameter is 1, the system is optimal, and when the parameter is 0, the system is completely undesired. The relevant judgment of the expected value should be combined with its own experimental needs and the optimization goal, such as whether to maximize or minimize the liquid fraction after stabilization.

Among them, the evaluation criteria for maximization and minimization are inconsistent. When desired to maximize the optimization objective, the mathematical judgment expression is as follows.

$$\begin{cases} d_i = 0 \quad y_i \le Low_i \\ d_i = \left[\dfrac{y_i - Low_i}{High_i - Low_i}\right]^{wt_i} \quad Low_i \le y_i \le High_i \\ d_i = 1 \quad y_i \ge High_i \end{cases} \tag{9}$$

Among them, y_i is the calculated value of the second-order polynomial and d_i is the expected value. When desired to minimize the optimization objective, the mathematical judgment expression is as follows.

$$\begin{cases} d_i = 1 \quad y_i \le Low_i \\ d_i = \left[\dfrac{High_i - y_i}{High_i - Low_i}\right]^{wt_i} \quad Low_i \le y_i \le High_i \\ d_i = 0 \quad y_i \ge High_i \end{cases} \tag{10}$$

The above two piecewise functions are the RSM evaluation criteria for single-objective optimization. When there are many optimization objectives, it is necessary to consider the expected values of all objectives and optimize for the total expected value. The following mathematical expression can describe this step.

$$D = (d_1 \times d_2 \times d_3 \times \cdots \times d_n)^{1/n} = \left(\prod_{i=1}^{n} d_i\right)^{1/n} \tag{11}$$

For example, taking the angle α and β of the fins as optimization factors and taking the stable liquid fraction as the response, when the total expected value is maximized, α and β are the best fin angles obtained by RSM optimization.

5.3. Orthogonal test method

Compared to the integrative considerations of RSM, the orthogonal test is more straightforward and saves time and effort. However, its shortcomings are also apparent. Since only representative test points are selected, inevitably, there will still be specific errors.

As a multi-factor and multi-level design method, the core idea of the orthogonal test is to select the most representative from the comprehensive test (taking the combination of all factor levels, similar to the exhaustive method) according to the orthogonality. These selected points are "evenly dispersed, neat, and comparable." Because of this, the orthogonal test is one of the main methods to analyze the influence of factors.

Among them, factors refer to influencing factors, such as the number of fins, height, thickness, number of heat storage units with fins installed, etc. The level refers to the number of cases. For example, A factor has three cases, which are A1, A2, and A3. When A factor is the number of fins, the cases A1, A2, and A3 corresponding to the number of fins can be 6, 7, 8, etc.

The operation of the orthogonal test is relatively simple, and it only needs to construct an orthogonal table for factors and levels. The constructed orthogonal table has the following properties.

(1) Each number appears the same number of times in each column.
(2) Every pair of ordinal numbers consisting of any two columns occurs the same number of times.

For example, construct an orthogonal table with four factors and three levels, where A represents the number of fins, B represents the height of the fins, C represents the thickness of the fins, and D represents the number of heat storage units with fins installed. Then the following orthogonal test table can be constructed (Table 8).

5.4. Taguchi method

The purpose of Taguchi method is to make the quality of the designed test product stable and less disturbed by the research factor, and to use the relationship between product quality, cost and benefit to design. In one sentence, it is to manufacture high-quality products at low cost.

In the fin structure design, the specific steps are as follows.

(1) Select the optimization objective.
(2) Determine whether the optimization objective is minimized or maximized. (Can be analogous to RSM)
(3) List all factors that affect the optimization goal.
(4) Define the influence level of each factor.
(5) When necessary, design and conduct interference experiments to judge the influence of research factors.
(6) Column Orthogonal Table.
(7) Execute the test to record the data.
(8) Computational analysis. (Such as K1, K2, and K3, where K is the average value of all cases of the same factor at the same level, for example, K can be the heat storage rate or heat release rate, compare the sizes of K1, K2, K3, etc.)
(9) Confirmation test.

Table 8
Four-factor, three-level orthogonal table for fin design.

	A	B	C	D
Case1	A1	B1	C1	D1
Case2	A1	B2	C2	D2
Case3	A1	B3	C3	D3
Case4	A2	B1	C2	D3
Case5	A2	B2	C3	D1
Case6	A2	B3	C1	D2
Case7	A3	B1	C3	D2
Case8	A3	B2	C1	D3
Case9	A3	B3	C2	D1

6. Conclusion

To improve the heat transfer enhancement effect of fins on phase change heat accumulators and expand their application range, this paper reviews the research progress of fin heat transfer enhancement technology. It discusses fins' design method and heat transfer mechanism, including their shape, size, quantity, and layout. This paper analyzes different optimization design principles, advantages, and disadvantages. It also provides references and ideas for the enhanced heat transfer of fins in phase change accumulators in the future. The main conclusions are as follows:

(1) The primary goal of developing new fins should be to eliminate heat transfer lag areas. The shape should preferably be designed with a multi-fork structure. A fine multi-branched structure enables uniform temperature distribution, deeper heat penetration, and point-to-surface heat transfer.
(2) Generally speaking, the excellent heat transfer performance of fins in descending order is multi-bifurcated structure fins, single-bifurcated fins, spiral fins, ring fins, or rectangular fins (with different heat transfer performance depending on the arrangement), and pin fins.
(3) Non-uniform fin arrays are superior to uniform fin arrays in heat transfer for melting or solidification. Non-uniform arrangements can be arithmetic arrangements, exponential arrangements, etc. The optimum arrangement can be designed following the functional equation evolved from the heat transfer formula.
(4) It is very effective to design according to the primary heat transfer mechanism of the heat transfer zone. For example, in a horizontally placed shell and tube accumulator, the upper half of the melting process is dominated by convection and the lower half is dominated by heat conduction, so the heat exchange effect with more fins in the lower half is better.
(5) The arrangement of fins needs to consider the fins' various parameters, the diameter of the tube, the angle of adjacent fins, the installation direction, and many other non-independent factors.
(6) The optimum number of fins exists in the accumulator. Above or below the optimum number of fins, heat transfer performance will be reduced. The optimal number of fins is related to the various parameters of the fins, as well as to the optimal balance of direct and indirect heat transfer.
(7) For multi-branched fins, there is an optimum number of branches, branch angles, and aspect ratios. Thin fins should have better heat transfer performance than thick fins. The fin height should be appropriate. The choice of fin size should also consider the fins' surface friction coefficient and thermal conductivity.
(8) Topology optimization is effective and practical in the design of fin shapes. RSM is often used to optimize the size and shape of fins, and similar optimization methods are Taguchi and orthogonal tests.
(9) The above summative, comparative, and reflective assessments can go some way towards avoiding duplication of effort, enabling more efficient and more straightforward fin designs for enhanced heat transfer, and understanding some of the better solutions and hidden reasons for fin designs.

7. Research challenges and future directions

By combing and analyzing the technologies, methods, and heat transfer mechanisms related to fin heat transfer enhancement in recent years, it is easy to find that in addition to the experimental method and heat transfer mechanism, each well-designed experiment has a specific reference value for its research. However, it is very specific, less universal, and more challenging to promote.

For example, each design experiment is carried out based on a specific heat exchanger shape, placement, fin shape, heat exchange environment, size, height, thickness, spacing, helicity, inclination angle, etc. It is highly targeted. In future research, more comparative experiments may be conducted to study the heat transfer comparison of different fins, size parameters, and positions in the same heat accumulator under the same PCM volume.

Alternatively, the heat transfer performance comparison of the identical fins in different heat exchange environments, such as stable heat source, sine and cosine pulse heat load, etc., or other conditions are the same, consider the laminar flow and turbulent flow of HTF, such as an inner spiral tube, turbulent flow experiments, etc. or only study the influence of dimensionless parameters such as Stephan number, Reynolds number, Rayleigh number, etc., or conduct a combination comparison of various fins to find the best size and shape in combination, or consider other conditions are the same but the influence of different accumulators on the heat transfer of the same fin, such as an eccentric tube, arcuate section tube, conical, rectangular, cylindrical, funnel-shaped. Alternatively, consider the influence of different types of PCM on fin heat transfer, or add other thermal solid conductivity materials such as carbon nanotubes, metal foam, expanded graphite, etc. to study its influence on fin heat transfer, or study the best balance of convection and heat conduction in the heat exchange of fins in the accumulator, and the best balance of direct heat transfer and indirect heat transfer.

In conclusion, more comparative experiments may be needed to enable finned heat transfer enhancement technology to provide more general references and improve the research efficiency of relevant researchers.

In addition, many of the fin structures studied may be too complex for commercial production. Perhaps the feasibility of manufacturing, the practicality of fins, and their economic benefits can also be considered in future research.

Declaration of competing interest

We declare that there are no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Data availability

No data was used for the research described in the article.

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