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Review Article

A review on the modification of circular fin and tube heat exchangers through new innovative fin shapes

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Abstract
Heat exchanger tubes with circular fins have been widely used in industrial and commercial applications over the past decade and are considered one of the major researcher's concerns in a range of engineering applications for their affordable cost and potential future to improve heat transfer. This review summarises the recently optimised circular fins (needle, trimmed, crimped, perforated) and how they affect heat transmission, the pressure generated and weight. Where varying fin-related parameters (fin height, fin material, the gap between fins, and number of fins) and flow-related parameters (inlet air velocity or manufacturing fin design) are aimed to optimise the performance in a different manner. In conclusion, designs of novel fin shapes remained a challenging field and demanded more studies to reinforce the existing papers to accomplish the optimum required performance.

Keywords
Circular finned tube, Novel fin shapes, Serrated fins, Segmented fins, Perforated fins, Spiral fins.

1.Introduction
Heat exchangers play a crucial role in industrialised domains such as residential and power engineering due to their widespread use in different applications such as compressors, boilers, air coolers, ventilators, heating, ventilation and air conditioning (HVAC), and freezers. A heat exchanger is an instrument that transfers temperature from one fluid to another, or from one fluid to a solid, and vice versa with the condition of temperature variations. Many researches have focused on improving heat exchanger efficiency. Finned tube heat exchangers are a commercially effective construction to accomplish that aim. The heat exchanger's gas side has the highest thermal resistance. The gas heat transmission is 10-50 times less than that for the liquids [1]. Therefore, gases generated 80% of thermal resistance [2]. As a result, various approaches, both passive and active, have been used to improve heat transfer and minimize resistance on the gas side. Active approaches need more power to increase heat transfer [3], which raises capital and operating costs, and even some active methods produce noise due to vibration.

On the other hand, passive techniques enhance heat transmission by modifying heat exchanger geometry or adding fins. As a consequence, most profitable improvement strategies are passive [4].

Fins are one of the most widely used passive approaches in commercial applications that demand a fluid-gas heat exchange. Moreover, they are classified as internal or external due to their position in or on the tubes, with the latter being the subject of this study. The external fin focused on lowering the high thermal impedance of gases compared to liquids and had the larger ability to extend the area of fins in contrast to the limited area inside the tube. One of the most used individually finned tubes is circular fins due to their high heat exchange, adjustable structure, ease of manufacture, maintenance, and cleaning. Although the wide usage of circular fins, the new applications demanded lowering the cost as most as possible and reducing the weight along with achieving a higher heat exchange rate. Therefore, a need was raised for designing a new modified circular fin to accomplish the new commercial requirements.

Literature recorded many of these new novel fin geometries such as segmented, serrated, perforated,

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crimpled, spiral, and needle fins. Modified circular fin classifications are presented in *Figure 1*.

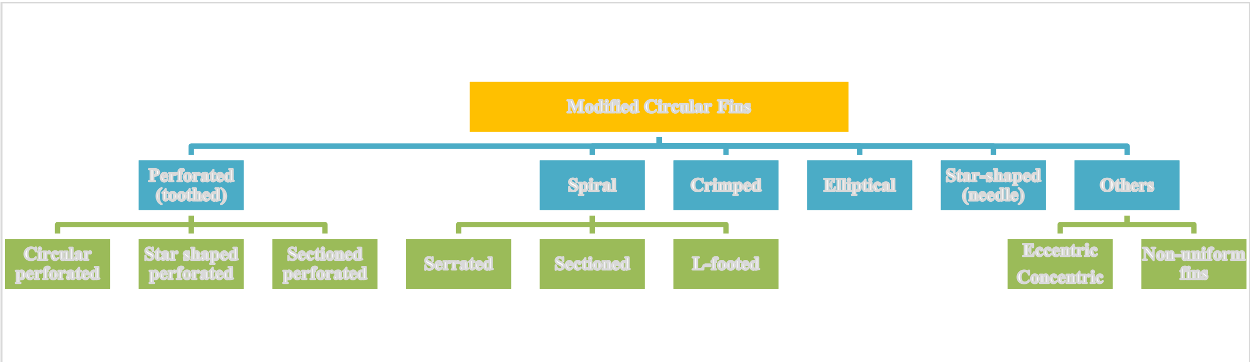

Figure 1 Modified circular fin classifications

The modified fin geometries are aimed to enhance heat exchanger functionality, as presented in *Figure 2*. By disrupting the thermal boundary layer, altered fin designs such as the aforementioned novel fins boost flow turbulence and heat transfer coefficient (h). The fin segmentation promotes turbulence and enhances gas accessibility to the fin root area, therefore balancing the effect of flow in fin length. A 20% increase in h over basic fins was explained by [5]. An increase in h is also noticed as in [6] by 7.61% using eccentric than the concentric one. While [7] represented an increase by 7.07% with perforated fins utilizations.

Various studies [8, 9] investigated an increase in Nusselt number (Nu) with the modified novel circular fins where it observed a significant increase in Nu by 23 % and 10.3%, respectively.

The proposed new fin designs, on the other hand, generate new challenges. One of these concerns is an increase in pressure loss, which demands more fan power to push the gas over the fins, as observed by [10]. However, some articles provided a better finding toward reducing the pressure loss that increased along with enhancing the heat transfer by interrupting the boundary, as mentioned by Bošnjaković et al. [2, 11−14], where the pressure loss was found to be lowered compared to plain (circular) fins. Nemati et al. [15] also achieved the same observation where a combined domain with circular and elliptical fins was utilized and reduced the pressure loss by 31%. As well as, a lower Euler number (Eu) was observed for serrated fin without twisting by [16]. Lowered the pressure drop by 43.09% [6].

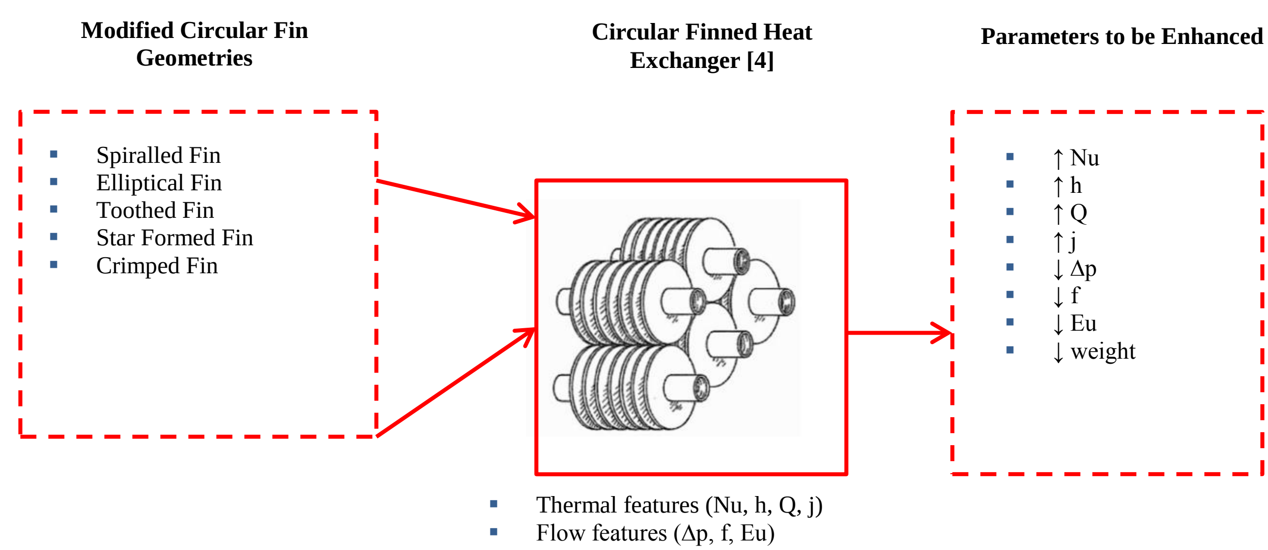

Figure 2 Pictorial representation of review analysis for modified circular fin

Because many applications necessitate a lighter heat exchanger, where the cost is also reduced. Therefore, weight is another critical factor to consider when developing modern heat exchangers, where the new novel modified circular fin shapes accomplished this aim as mentioned by Anoop et al. [17], where the serrated fins reduced the weight by 7.5% using serrated fins. While Martinez-Espinosa et al. [18] lowered mass in the range from 25% to 33.3% utilizing helically segmented fin with cut-off angles of 45° and 60°, respectively. On the other hand, Bošnjakovic et al. [2] reduced the mass by 23.8 % with needle fin utilization. Utilizing perforated fins with two-hole radii R 1.25 and R 1.85 [19] reduced the weight by 6.62%-9.49%, respectively. In line with this finding, other articles observed a reduction in weight [20, 21].

As a result, a balance between the demand for a higher heat exchanged with a lighter weight and raising pressure loss will be a challenge for future investigations.

Heat exchange and pressure collapse may be studied theoretically, experimentally, or numerically using computational fluid dynamics (CFD). Experimental approaches explain physical occurrences, whereas analytical methods control equations. CFD advocates algorithmic numerical ways to solve Navier–Stokes and related equations. CFD provides design evaluation without harming the plant. Studying using CFD is dependable. Multiple grid points or a mesh may handle variable geometries. Long computer time and thorough turbulence model selection [22]. This reduces plant expenses. CFD provides complete, graphical, detailed data [23−28]. The current study examined circular finned tube heat exchangers' analytical methods and fin design elements.

The main objective of this review is to present a comprehensive study of the developments in the traditional circular fin geometries and address their impact on heat transfer, flow features, and weight reduction. Further, challenges have been explained in this review.

This paper is organised into the following sections. Section 2 presented the methodology of the research. Section 3 illustrates the motivation of this review. Inclusion and exclusion have been added in section 4. Section 5 summarizes the approaches to analysis. Section 6 reports the experimental studies, while section 7 summarizes the numerical studies. Section 8 describes other studies for modified circular fins. The application of these modified circular fins is illustrated in section 9. Section 10 highlighted the challenges that face the modified circular fins. In the end, discussion and conclusion have been added in sections 11 and 12, respectively.

2.Methodology of research

2.1Questions of research

The major purpose of this research review is to examine the influence of modified circular fin designs on heat transmission, flow, and weight reduction. To do this, we posted the following research questions, as summarized in *Figure 3:*

Q1. What are the classifications of all the modified circular fins?

Q2. What are the current achievements in heat transfer, flow characteristics, and weight reduction with respect to the recent circular fin modifications?

Q3. There are so many scenarios of heat transfer modification. What would be the optimum configurations to accomplish that?

Q4. What are the major challenges to be addressed in circular fin modification?

Section 11 addresses all research inquiries.

2.2Selection criteria

The focus of this study is to provide a comprehensive classification of modified circular fins. This research employed a systematic literature review based on the preferred reporting items for systematic reviews and meta-analyses (PRISMA) technique [23]. An initial survey was conducted to locate literature on modified circular fins. The publications were obtained from databases such as Springer, Science Direct, Scopus, Web of Science, and others. The first level of filtering used keyword detection; the second level is based on the publications' titles and abstracts. The final group of publications was later determined by reviewing the whole article. As shown in *Figure 4*, the screening method discovered that the number of publications falling within the scope of this research was relatively small. *Figure 5* shows the distribution of publishers, with 46% of Elsevier journals picked, 7% of Springer, 7% of MDPI, 6% of ASME, 6% of Taylors & Francis publishing, 3% of NTNU, and 25% from other publishers. *Figure 6* depicts the year-wise distribution of the selected papers. *Table 1* contains the publishing of journal articles, books, reports, theses, and conference proceedings.

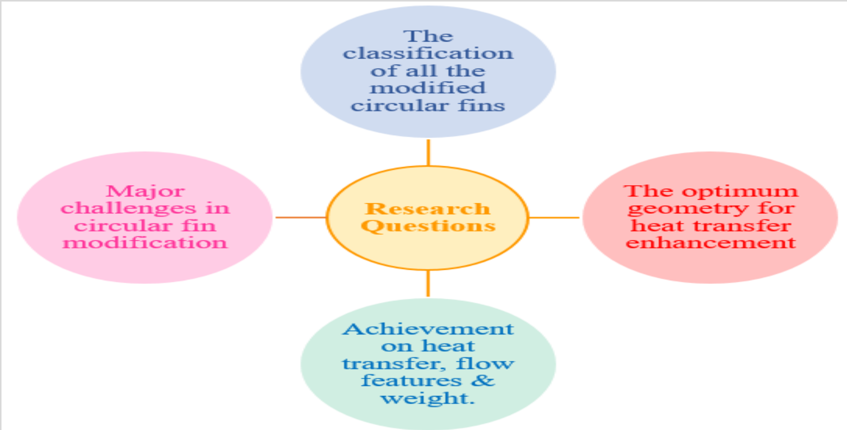

Figure 3 Mapping research questions

Table 1 Reference categorization

Reference type	Author	Year	Reference type	Author	Year	Reference type	Author	Year
J	[1]	1996	J	[40]	2015	J	[79]	2012
J	[2]	2017	J	[41]	2003	J	[80]	2016
J	[3]	1979	J	[42]	2012	J	[81]	2020
B	[4]	1987	J	[43]	2011	J	[82]	2017
J	[5]	1994	J	[44]	2012	J	[83]	2021
J	[6]	2015	J	[45]	2009	J	[84]	2021
J	[7]	2012	B	[46]	2020	J	[85]	2018
J	[8]	2014	J	[47]	1976	J	[86]	2016
J	[9]	2022	J	[48]	2005	J	[87]	2022
J	[10]	2011	J	[49]	2006	J	[88]	2022
J	[11]	2020	J	[50]	2015	J	[89]	2021
J	[12]	2022	J	[51]	2012	J	[90]	2017
J	[13]	2020	J	[52]	2010	J	[91]	2022
J	[14]	2021	J	[53]	1976	J	[92]	2020
J	[15]	2020	B	[54]	2002	C	[93]	2018
J	[16]	2020	J	[55]	2007	J	[94]	2018
J	[17]	2015	C	[56]	2000	J	[95]	2019
J	[18]	2017	J	[57]	2007	J	[96]	2019
J	[19]	2018	J	[58]	2009	J	[97]	2021
J	[20]	2017	J	[59]	2010	J	[98]	2020
T	[21]	2021	J	[60]	2012	J	[99]	2014
J	[22]	2017	J	[61]	2013	J	[100]	2018
B	[23]	2013	J	[62]	2022	J	[101]	2017
B	[24]	2015	J	[63]	2022	J	[102]	2015
B	[25]	2005	J	[64]	2022	J	[103]	2015
J	[26]	2015	J	[65]	2020	J	[104]	2021
J	[27]	2019	J	[66]	2015	J	[105]	2019
J	[28]	2019	J	[67]	2015	C	[106]	2005
J	[29]	2012	J	[68]	2019	J	[107]	2011
B	[30]	2013	J	[69]	2020	J	[108]	1969
B	[31]	2003	J	[70]	2016	J	[109]	2006
J	[32]	2015	C	[71]	1962	J	[110]	1979
J	[33]	2005	J	[72]	2004	J	[111]	2013

Reference type	Author	Year	Reference type	Author	Year	Reference type	Author	Year
J	[34]	2020	J	[73]	2014	J	[112]	2010
J	[35]	2020	J	[74]	2022	J	[113]	2020
J	[36]	2021	J	[75]	2016	J	[114]	2020
J	[37]	2022	J	[76]	2020			
J	[38]	2021	J	[77]	2020			
J	[39]	2018	J	[78]	2020			

B = Book; C = Conference; J = Journal; T= Thesis.

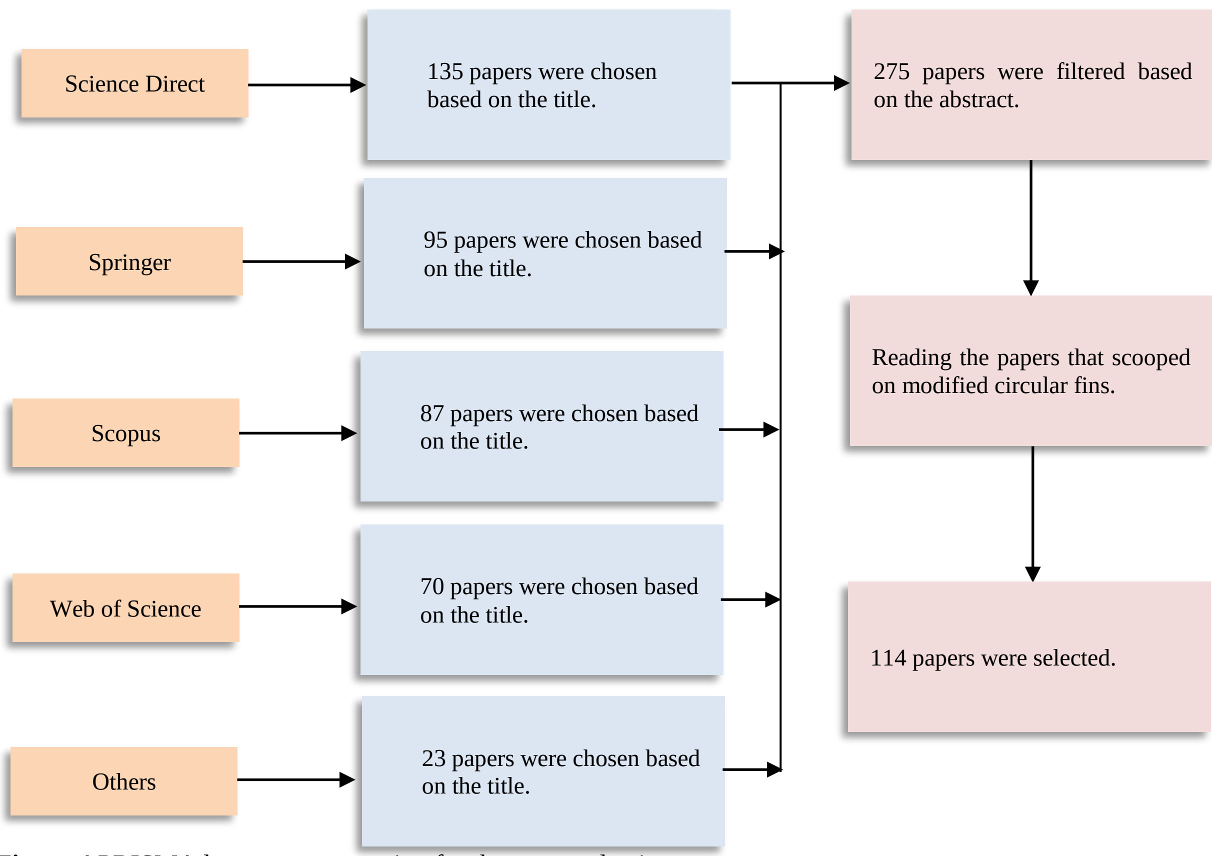

Figure 4 PRISMA layout representation for the paper selection strategy

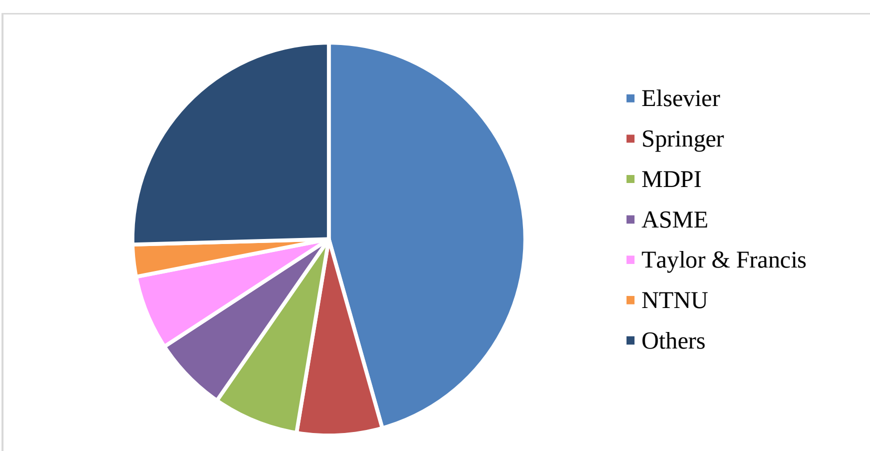

Figure 5 Publisher selection distribution

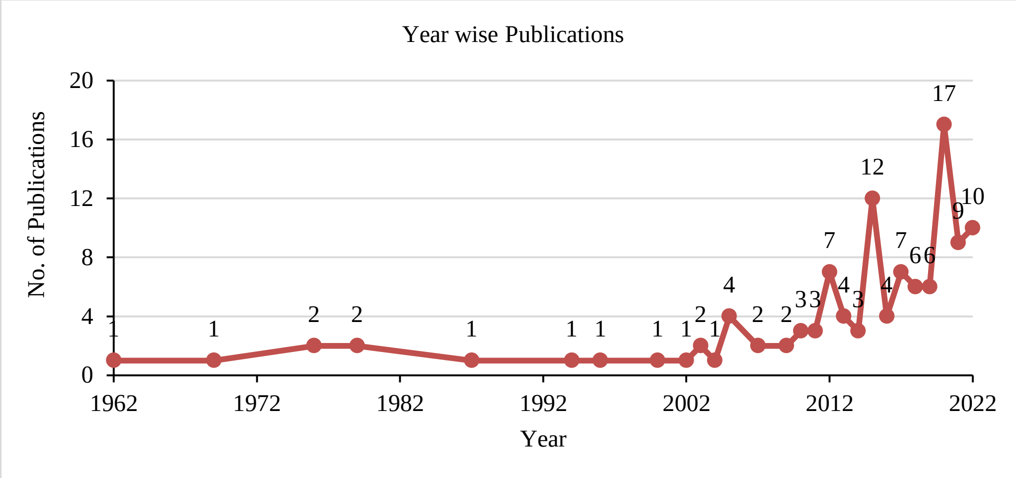

Figure 6 Year-wise distribution of the selected papers

3.The motivation for the study

Various modifications of fin geometries and their impacts on material and energy savings inspired the suggested review. It is observed that most of the existing literatures' review related to plate fins and other fin kinds. Therefore, a comprehensive study is crucial to cover the influence of the modified circular fin geometry, velocity, and material utilized on the thermo-hydraulic parameters (Nu, h, Q, j, Eu, f, and pressure drop), as well as on weight reduction.

4.Inclusions and exclusions

The criteria for inclusion/exclusion in this review are presented in *Table 2.*

Table 2 Reference categorization

Inclusions	Exclusions
Individually (Separately) finned tubes	Continuous finned tubes
Externally fins	Internally fins
Modified circular fins	Plate and longitudinal fins
Round tube shape	Varied tube shapes
Heat transfer	Mass transfer

5.Approaches to the analysis

Scientists have developed diverse approaches to employ the examinations for heat exchanger and pressure collapse where fin functioning could be explored analytically, experimentally, or numerically, for example, using CFD. Analytical approaches focus on accurately regulating equations, whereas experimental methods focus on describing physical events. Recently in fluid dynamics CFD, algorithmic numerical techniques are most encouraged for usage to solve the Navier–Stokes equation and related equations. In addition, CFD enables the assessment of design criteria without influencing the actual plant. Moreover, CFD was acknowledged as a reliable asset for research purposes. Variable geometries may be properly controlled but need a large number of grid points or a decent mesh. The process takes a long time on the computer, and the turbulence model must be chosen carefully, as clarified by [22]. As a consequence, plant construction costs are lowered. Furthermore, as compared to analytical and experimental fluid dynamics, CFD gives complete, graphical, and exhaustive data as described in many studies [23−28]. CFD is a helpful approach for forecasting the behaviour and performance of a broad range of heat exchangers since the accuracy of the outputs from these simulations is generally within the acceptable limits [23, 29]. Due to the time and cost consumption. Many strategies, including the number of heat transfer units (NTU) and the log mean temperature difference (LMTD), have several downsides, such as their iterative approach and the requirement for a model in order to put the idea into practice. In contrast, CFD has made it possible to be employed easily and developed rapidly with accessibility to superior computer processors.

The number of transferred units ($NTU = UA/(\dot{m}\,Cp)$) depends on the total heat transferred (U), transfer area (A), fluid flow rate ($\dot{m}$), and heat capacity (Cp). It condenses all of these dimensional parameters into a single dimensionless parameter. This dimensionless parameter becomes a monotone function of performance. These dimensionless correlations are used to estimate geometric and design characteristics [30, 31]. CFD is the most used approach to investigate heat transferred and flow features due to the aforementioned reasons and for different applications [32]. As a result, it was employed by many applications as [33−39].

Researchers have developed numerical algorithms to utilize this tool to model challenging flows and heat transfer issues for decades, including conventional and accelerated methods. The conventional approaches are most broadly applied, fairly accurate and generally popularly employed in commercial software platforms, which include finite volume, finite element, finite difference, and spectral. On the other hand, the standard (conventional) approaches are exceedingly slower in regard to simulation time. Therefore, accelerated approaches have developed that consist of elevated numerical methods and hardware approaches, as explained in detail by [40]. CFD results are susceptible to boundary conditions and need intensive calculation. Current CFD software is less complex where. In CFD simulations, users may analyse just a few development or functional situations by adopting a fixed set of boundary conditions, such as fixed speed rate, fixed wall temperature, or heat flux (Q) through the simulation [41]. The aforementioned approaches are summarized in *Table 3.*

Table 3 Approach analysis

Approach of analysis	Advantage	Disadvantage	Limitation
CFD	• Cost reduction. • Less time compared to the experimental method. • Environment friendly • Model complex application • Visualized the flow features easily.	• Issues might arise from basic flow models or boundary conditions. • Too few computations per cell might produce interpolation issues. • Large models may need more time.	• Required elevated computers. • Demanded a fine mesh. • Sensitive to boundary conduction.
NTU	Predicting outlet temperatures without iteratively solving nonlinear equations.	Less ability to work with complex heat exchanger design.	Heat transfer coefficient and input temperatures favoured
LMTD	LMTD technique determines U based on experimental inlet and exit temperatures and fluid flow rates.	This approach is inconvenient for predicting output temperatures if intake temperatures and U are known.	LMTD can be enlarged to utilise complicated heat-exchanger design features

6.Experimental studies

6.1Needle or star shaped fins

Bošnjaković et al. [13] experimentally studied the effect of stainless steel star-shaped finned tubes on heat transport with Reynold values (Re) ranging from 2000 to 13000 and potential mass reduction. Heat flow through star-shaped finned tubes rose from 35.5 to 55.8% compared to round finned tubes, and mass decreased by roughly 23.5 %. The heat transfer coefficient (h) rose to 18.5% from 16.5% earlier.

Two correlations for heat exchange and pressure fall have been proposed by Bošnjaković et al. [14] for star-shaped fins as shown in the Equations 1 and 2 and tubes organized in cross-flow heat exchangers through utilizing CFD. Parametric simulations have been created for twenty-one heat exchangers in alternative configurations, alternating fin thickness, fin pitch, and external speed (cooled air). It was determined that the original correlations for Nu and Euler number (Eu) both were justified in Re ranges less than 16,000 and bigger than 2000, fin thickness larger than 0.3 and less than 1.0, and the ratio of the gap between fins to the outer diameter (s_f/d_o) 0.15< (s_f/d_o) <0.4.

Finally, the correlation coefficients were adjusted to reflect the laboratory findings. The final attained linkage Nu equations diverged by 10% for Re less than 3500 from the experimental findings, but less than 2% for higher values of Re, and by 19% in the region of Re<4000 for Eu. The variance was less than 1% for Re > 5600.

$$Nu = 0.15\,.\,Re^{0.71}.\,Pr^{\frac{1}{3}}\left(\frac{t_f}{s_f}\right)^{0.0254}.\,(\frac{s_f}{d_0})^{0.07} \quad (1)$$

$$Eu = 2.6.\,Re^{-0.2}.\left(\frac{t_f}{s_f}\right)^{0.071}.\,(\frac{s_f}{d_o})^{-0.273} \quad (2)$$

6.2Crimped fins

Diverse fin pitches, including 2.4 mm, 3.2 mm, and 4.2mm, were tested by [42] for frictional characteristics and heat transfer performance using L-footed spiral fins, where Re ranged 4000 - 15,000 and tube arrangements in parallel and counter cross-flow.

Fin pitch was shown to have a substantial impact on the heat exchange rate and pressure decrease. Heat transfer seems to be self-governing not influenced by distances between fins. Fin pitch, on the other hand, has negligible influence on friction factor (f) for Re values less than 6000. However, when the Re approaches 6000, the f rises. At higher Re, a 4.29 % departure from the Colburn factor (j) and a 2.12 % divergence from the f is indicated.

Pongsoi et al.[43] constructed six crimped finned tubes to evaluate the air-side performance of diverse gaps between fins (f_p) like 3.2 mm, 4.2 mm, and 6.2 mm, and fin materials (copper and aluminium) using two rows of tubes, each having 0.4 mm fin thickness and an outer diameter of 34.8 mm. A suggested formula for mean effectiveness that may clearly describe the usefulness of the NTU connection based on the present z-shaped structure.

The outcomes exposed that the gap between fins has no influence on the heat exchanged quality and j since a high Re promotes proper mixing, which increases heat exchanged function independently on space between fins. The difference between 3.2 and 4.2mm fin pitch may be minor for the same reason. When the distance between fins expanded to 6.2 mm, the f increased significantly. Furthermore, the material of the fin has little influence on air-side performance.

Researched the consequence of tube row number on the gas-side functionality of crimped spiralled finned tube with multi-pass parallel and counter cross-flow at Re from 3000 to 13,000, dependent on outer tube diameter[44]. At higher Re (3000–13,000), the density of tube rows has essentially little influence on the j. Moreover, the pressure loss rises with forwarding air velocity, and it is twice in 5-row tubes compared to 2-row tubes. However, the number of tube lines increases the mean heat exchange rate and pressure fall. Both the j and f diminished as the Re climbed.

Tang et al. [45] explored (crimped, basic or simple, trimmed, vortex generators along tubes (VGs) wing shaped) fins. Lastly, a combination design of fins with frontally located 6-lines VGs fin and back 6-row trimmed fin with Re ranging more than 4000 and less than 10000, N=12 tube rows, and D_o =18 mm. Crimped fins showed the strongest score for heat exchanged and pressure loss when compared to other fin designs, but mixed fins outperformed delta-wing longitudinal VG fins. The algorithm improvement findings revealed that increasing the attack angle or extent or reducing the VG elevation would augment the usefulness of the VG fin. In terms of heat transfer, it can outperform a slit fin. On the other hand, VG doesn't boost fin performance since there is no longitudinal horseshoe vortex [46].

6.3Spiral fins

Spiral fins might be segmented, serrated, or helical. Serrated fins are used to improve the performance of boilers, HVAC, power plant cooling, and economizers. Weiermann [47] evaluated the performance of inline and staggered steel tubes with serrated finned tube bundles. A waste heat recovery device weight-reduction test was also conducted. Outcomes found that segmented finned tube banks may increase heat transmission by disrupting the flow. Moreover, fin sections reduce thermal stress, allowing for increased fin height. As a result, reducing the number of finned tubes reduces expenses. Kawaguchi et al. [48] performed a comparison between spiral (solid or plain) and serrated segmented finned tubes to predict Nu and develop heat exchanger equations. Fin pitch differs between 5mm solid and 3.3mm serrated fin tubes. Additional geometrical characteristics, including fin kinds, tube arrangement, stream wise tube pitch (S_L), and row number (N_L), were also explored. The study found that narrower fin spacing improves h regardless of tube arrangement or fin-type (spiral or serrated); the increase is 30% for spiral fins and 10% for serrated fins. It also tends to grow when tube pitch drops, although this impact is minor. Same tube line number enhanced spiral and serrated fin h. Rising the number of tube lines, from 3 to 6, slowed the pace growth. Finally, serrated fins provide greater heat transmission with less area. Nu correlation equations were reported for both spiral solid (without serration), as presented in Equations 3 and 4.

For spiral fins:

$$Nu = A_1. Re_v^{0.787}. Pr^{\frac{1}{3}} (\frac{s_f}{d_v})^{-0.264} \qquad (3)$$

For serrated fins:

$$Nu = A_2. Re_v^{0.784} Pr^{\frac{1}{3}} (\frac{s_f}{d_v})^{-0.0620} \qquad (4)$$

According to Kawaguchi et al. [49], the water flow rate is 0.075 kg/s and the airflow rate is 0.14 kg/s to 0.80 kg/s. The influence of tube layout was also evaluated and found to be minimal for spiral and segmented serrated fins. Similar to prior work, f correlation equations were provided for spiral solid (without serration) in the range $2000 \leq Re_h \leq 27{,}000$ and serrated segmented in the range $3000 \leq Re_h \leq 30{,}000$, as presented in Equations 5 and 6. Higher fin

heights resulted in more pressure drop. Also, shorter fin tubes had an equal or larger h compared to the longer fin tubes.
For spiral fins:

$$f = 2.60Re_h^{-0.24}\left(\frac{h_f}{d_h}\right)^{0.004}\left(\frac{s_f}{p_f}\right)^{-4.13} \quad (5)$$

For serrated fins:

$$f = 4.99Re_h^{-0.23}\left(\frac{h_f}{d_h}\right)^{0.13}\left(\frac{s_f}{p_f}\right)^{-1.9} \quad (6)$$

As a consequence of the smaller fin area, finned tube banks with low fin height have a lower heat exchange rate than those with higher fin height. Serrated spiral fins were tested at varying water vapour concentrations [16]. Each row has eight active heat transfer tubes, with Re varying from 6000 to 12000. Nu increases with water vapour concentration, according to the results. Twisting a finned tube's serrated fin at 250°C and 350°C increases Eu by 13%.

Kiatpachai et al. [50] examined the impact of fin pitches (3.6, 4.2, 6.2) mm on serrated spiral finned tubes with Z-shaped flow arrangement in Re ranging from 4000 to 15000 and two rows of steel finned tubes. On the gas side, space between fins has a substantial influence. Under observed conditions, h has the same behaviour for the fin gaps 3.6mm to 6.2mm. When fins depart as the gaps climb to 6.2 mm, there is a noticeable increase in f. As the distance between the fins diminishes, the pressure and f decrease. Correlation equations have been proposed and presented in Equations 7 and 8.

$$j_{corr} = 0.0924Re_{do}^{-00.22583} \quad (7)$$

$$f_{corr} = 0.76833Re_{do}^{-0.11243}\left(\frac{f_p}{d_o}\right)^{0.33662} \quad (8)$$

Ma et al. [51] adopted serrated fins with a staggered tube configuration and water characteristics of $2\times10^4<Re<4\times10^4$, $4.0<Pr<7.0$ using Gnielinski correlations, as presented in Equations 9 and 10, to explore their thermohydraulic features. Heat exchange and pressure drop characteristics were affected by fin density, longitudinal, and transverse tube spacing. Increasing fin density increased Eu, and fin height lowered Nu. Eu reduced when transversal tube departing grew, while Nu stayed roughly constant, and the optimal transversal-to-longitudinal tube departing percentage increased. Transverse tube departure increased from 2.3mm to 3.2mm, lowering pressure drop by 20%. The longitudinal tube spacing has a negligible effect on Nu and Eu, pressure drop, and j to f proportion. The Nu was unaffected by increasing the tube separation distance, while the Eu was. Nu ranged from 0% to 11%, whereas Eu rose to 8% as the fin elevation-distance fraction increased to 5.5mm from 5.0mm. Increasing fin separation and transverse tube spacing increased j and f by 28% and 25%, respectively.

$$Nu = 0.117Re^{0.717}Pr^{0.33}\left(0.6 + 0.4e^{-\frac{250h_f/s_f}{Re}}\right)\left(\frac{S_T}{S_L}\right)^{0.06} \quad (9)$$

$$Eu = 1.773Re^{-0.184}\left(\frac{h_f}{s_f}\right)^{0.556}\left(\frac{s_1}{d_o}\right)^{-0.673}\left(\frac{s_2}{d_o}\right)^{-0.133} \quad (10)$$

Martinez et al. [52] industrially assessed the functioning of helical segmented fins and compared it to experimental results using a 10,000 Re flue gas turbine. A mix of Kawaguchi et al. [48] and Gnielinski [53] theories predicted and measured heat transport with 95% accuracy. Ganapathy [54] and Kawaguchi et al. [48] models predicted lower Re values than the experiment. Backpressure poses operational risks for certain models. Reynolds Re values over 8000 have an exponential slope; therefore, forecasts fluctuate significantly. The Nire-Gnielinski hypothesis showed worse heat transfer estimates, with accuracy ranging from 57.42 to 73.87 % and one estimate of 95.7%. Temperature forecasts are 96% accurate; hence the results are credible. The ESCOA-Gnielinski hypothesis is 69.94% to 94.41 % accurate in total heat exchanged (U), and 97 % accurate in temperature estimates. Weierman-Gnielinski model has 83.8% to 97.66% U accuracy. The Kawaguchi-Gnielinski hypothesis has the highest accuracy (93.75 to 99.59%) and single-precision (77.74%).

Hofmann et al. [55] studied cross-flow heat exchange and pressure decrease with varying transversal serrated fins and full finned tubes with alternate fin separation space, fin thickness, fin height, and fin width. Experimental data and existing formulae confirmed the Nu and pressure drop coefficient calculations. Heat transfer reduced as fin thickness increased. Second law analysis can compute effective fin height. When paired with the Lévêque equation, the Nu yields accurate pressure drop predictions [56]. Using a single-phase performance criterion, finned-tube packs' efficiency was estimated. Three Re ranges were developed while comparing serrated finned tubes: Re <10000, 10000<Re<20000, and Re > 20000. Changing fin height and pitch in two designs had little influence on these calculations. Full and segmental fins boosted air-side heat transfer across the Re range. Full and serrated I-shaped finned tubes functioned well. Fin height was the most essential aspect in assessing finned tube designs.

Heat exchanged and pressure fall characteristics for helical I and U-formed fins, along with solid I-fins in staggered tube arrangement, were experimentally examined in a cross-flow by Hofmann and Walter [57]. The effect of tube number lines ordered in stream path is also investigated. Based on the results, the impact of fin design on I- and U-shaped helical heat transfer might be neglected. For pressure drop, two original formulae applicable to helical sectioned U-formed fins tube bundles in a staggered pattern exhibited a mean variance of ±13% for 90% of the data gathered. The heat transmission and pressure drop formulae were deviated by ±20%.

Martinez et al.[58] utilized LMTD to produce helical toothed. Fins Semi-empirical correlations, as presented in Equations 11 and 12, produced convection heat transfer coefficients with interior and external staggered tubes. With a 0.4–4 C mismatch between theoretical and real data, accuracy fluctuates from 94.41% to 69.94% for the ESCOA–Gnielinski hypothesis and by 97.66% to 83.80% for the Weierman–Gnielinski hypothesis. For Re values below 10,000, pressure decrease estimates varied substantially.

$$Nu = \frac{\alpha_i d_i}{\lambda_w} = \frac{\left(\frac{f_i}{8}\right)(Re-1000)Pr}{1+12.7\left(\frac{f_i}{8}\right)^{\frac{1}{2}}\left(Pr^{\frac{2}{3}}-1\right)} \quad (11)$$

$$f_i = \frac{1}{(1.82 log_{10} Re - 1.64)^2} \quad (12)$$

Huisseune et al. [59] explored experimentally the helically finned tubes in a single row heat exchanger. The transversal tube's pitch was changed in this study. Lower transversal tube pitch and larger pressure drop resulted in increased h. The minimum Correlation between Nu with f with a mean deviation of 4.49 %, the heat transfer correlation may reflect 95% of the data. With a mean deviation of 6.84 %, the f correlation properly predicts 95% of the data.

Heat transmission of air passing across spiral finned tube banks with various Re from 5×10^3 to 5.5×10^4 and geometric characteristics (fin distance t/d=0.220.5, fin length h/d=0.220.5, longitudinal tube pitch (s_2/d =23.3125), transverse tube pitch (s_1/d =23.3125) was studied by [60]. Tests showed Nu and Eu correlated with Re, dense, and fin height with transversal tube dense s_1 and horizontal tube dense s_2. Pongsoi et al. [61] studied the effects of fin pitches (2.4, 3.2, and 4.2 mm) on heat transmission and frictional characteristics for an L-footed Spiral finned tube throughout a large Re range (4000–15,000). Fin density had no effect on j or heat transfer. Fin pitch affects heat exchange, pressure decline, and f. Below 6000 Re, it has no impact. When Re reaches 6000, the fin pitch increases f. The samples were organized using parallel and counter cross-flow. The mean variances for these correlations were 4.29 % and 2.12 %, respectively. Correlations were proposed as presented in Equations 13 and 14.

$$j_{corr} = 0.2150 Re_{dc}^{-0.4059} \quad (13)$$

$$f_{corr} = 0.4852 Re_{dc}^{-0.2156}\left(\frac{f_p}{d_c}\right) \quad (14)$$

Serrated spiral fins were tested experimentally. Segmented fin height (h_s) and fin pitch (f_p) alter air-side performance, according to Keawkamrop et al. [62]. At the same f_p, segmented serrated finned tubes have a higher air-side heat transfer coefficient. Nu and (j) are more affected by h_s than f and Eu. Nu, j, f, and Eu correlate with plain and serrated fins.

Kiatpachai et al. [63] studied how fin morphologies affect louvre spiral fins' air-side performance. Round, curved, and combined-louvre spiral fins were compared to plain spiral fins. Heat exchangers had 8.45 mm fin pitch, crimped aluminium fin bases, and two rows of steel tubes. Air velocity affects Q_{ave}, h_o, and P. Louver-spiral fin increased Q_{ave} and h_o 9.7–15.6% and 13.4–27.1%, respectively. Louvre-spiral fins have higher j factors than plain fins by 10.4–13.1% for combined louvre, 7.7–8.8% for circular, and 2.1–5.1% for a curved fin. Both louvre- and plain-spiral fins have the same pressure and f. h_o and j are boosted by louvre fins.

A serrated twisted fin is suggested to improve heat transmission. The windward side of the partially serrated twisted finned tube is grooved to enhance heat transfer. Leeward side non-grooving may boost heat transmission and flow resistance. Experiments carried out by Wei et al. [64] on a partly serrated twisted finned tube validate the modelling findings. End with Nu and f connections. Compared to circular fins with Re number ranging 5000 - 17,000, partly serrated twisted fins raise Nu by 14.9% and $j/f^{1/3}$ by 9.2%.

Chien et al. [65] studied film evaporation at 10°C and 20°C saturation temperatures utilizing 60 annular fins per inch, each with 40 axial cuts. Heat transfer coefficients increased for smooth and serrated fin tubes. The flow rate also increased before dry-out.

The film flow rate didn't impede smooth tube heat transfer. Lower film Re reduces serrated fin tube evaporation heat transfer. Droplet spreading shows axial incisions improved fluid dispersion on the serrated fin tube. Serrated fin tube increased R-134a falling film evaporation by 10.18 and 6.02 at 10°C and 20°C.

Experiments were conducted utilizing several fin dense (2.4, 3.2, and 4.2) mm with Re ranging (5,000–15,000) [66]. Parallel and counter cross-flow test sections featured 2 tube rows. Both 2.4 and 3.2 mm fin pitches were found to be the optimized shape for L-footed spiralled fin.

6.4Perforated fins

The influence of perforated fins on heat transmission was tested experimentally [19]. Hot water inside the finned tube warms the fin air side. Two fin sets are mounted on 22- and 26-mm tubes. Two hot water flow rates and six air Re were tested. 60 tests were performed on four finned holes, with R1.25 and R1.85, and one un-finned tube. Internal or external flow rates enhance a larger cross-influence. Perforated fins reduce weight and increase heat transmission by 8.78% and 9.23% at low Re. High external Re reduces heat transmission by 8.4% and 10.6% at 0.05 and 0.1 kg/s.

Perforated finned tubes with widths of 6 mm were studied experimentally by [67] to determine the best location that influenced the thermal properties and, therefore, performance increased. Perforation fins may diminish the boundary layer breadth created on ring-shaped fins, improving heat transmission. The fins used were 87 mm in diameter and 0.5 mm thick. In addition, gap-separated fins apart varied by 4, 8, 10, 12, and 15 mm apart. For the hot fluid (water), m_w = 1.1kg/s was employed, with temperatures ranging from 40°C to 70°C in 5°C increments. Also, perforated finned tube cases were compared to solid finned. The 60° heated fins were 18% more efficient and lost 1.16 % less pressure than other angles, although their effectiveness dropped with increasing m_{air}. The ideal angle for heat transfer and fin utilisation is 60°. This spot reduced resistance.

Sundar et al. [68] created a model to improve thermal efficiency and reduce the heat sink's mass. He verified the model with experimental testing. The perforated staggered heat sink arrangement lowered thermal resistance by 7% to 12% at varied orientation angles and mass by 9%.

Poor thermal conductivity of phase-transition materials restricts heat storage. Karami and Kamkari [69] sped up buoyancy-driven convection to improve phase transition materials' thermal performance. This research used toothed fins to improve shell-and-tube thermal performance. Clear Plexiglas tubes were used to make heat exchanger shells. Finned and tubed copper. At 0.5 and 1 l/min at 55 and 65 °C, phase transition material melts. Because perforated fins have no impact on convection flow, toothed fins have a 30% higher Nu than plain fins. Compared to solid heat exchangers, toothed ones melt 7% quicker.

7.Numerical studies

7.1Optimized round shapes and needle fins

CFD was used to quantitatively investigate eccentric annular finned tube banks by [6]. The study's purpose is to achieve optimum heat dissipation while minimizing pressure drop. Also, looked at how fin dense and diameter affect heat transmission and stream features for Re ranging from 4500 to 22500. The eccentric annular finned tube outperformed the concentric one at small fin gaps. This location increases heat transmission characteristics and decreases pressure loss when contrasted to the other heat exchangers. The eccentric annular-finned tube exhibited a 7.61 % greater mean heat transfer coefficient than the concentric one with a tube layout where the transversal gap was equal to 12 mm and Re equal to 9923 and followed by a pressure drop decrease of 43.09%.

Modifying fins geometry is one of the keys to enhancing the heat transfer process, considering the heat exchanger's mass and size. [70] Concerned with the validity of gas-side formulas in a cross-flow heat exchanger, where it was examined numerically using high-performance computational fluid dynamics (CFX) and CFD to form a model with various improved fin geometries and varying fin thickness. According to the Briggs and Young [71] formulas, heat transmission diminishes as fin width increases.

Mon and Gross [72] studied four-line circular fins with staggered and inline configurations. Re and the fin dense to elevation fraction affected the occurrence of a boundary layer and horseshoe eddies between the fins.

Compared nine different turbulence models for four rows of round finned tube bundles using CFD software. Due to cost, geometric complexity, flow behaviour, and fin thickness, computational procedures may replace experimental methods [73].

The two techniques, K-Kl-ω and transition shear stress transport (SST), are more compatible with experimental correlations because of the low Re. Moreover, although all models worked effectively, the k-outcomes groups were different. The flow through the tube separates at 90°, whereas the flow on the fin edge separates at θ>120°.

One of the most used applications for circular finned tubes is in air conditioning and other devices that demand exchanger heat with the best possible methods; as [15], who studied an air cooler with four rows of finned tube bundles in the preliminary, the potential of independently modifying fins has been explored by altering the shape of the annular finned tubes locally, leading to non-uniform tube bundles with four objectives to accomplish weight reduction, pressure drop reduction, fan power reduction, and entropy decrease. In a reference case, elliptical geometry of 57.15 mm diameter is enlarged to 36 mm, reflecting 63 % of the main diameter. The results of the first scenario on weight loss indicated a 35% reduction in weight. Also, the total heat transmission rate decreased by 22% compared to the reference case (circular fins). The finned tubes were made up of circular fins at the flow intake and enlarged elliptical fins with the lowest minor diameter. The circular flow intake fins cause turbulence while the elliptical fins save weight. Entropy is reduced by 22%, and pressure drops are reduced by 50% by a homogenous bundle of elliptical-shaped finned tubes with modest vertical diameters. In another case, the heat transfer was improved by 14%, the pressure dropped by 30%, and the weight was reduced by 23% compared to base fins, with the finned tube layout consisting of circular fins at the inlet and outlet and elliptical fins in between.

Bošnjaković et al. [2] numerically examined, by utilizing CFD, and the SST k-ω turbulence model, a novel design of finned tube bundles called needle fins in Re range 2300>Re>12000. A needle fin is a segmented serrated fin arranged in a staggered layout in cross-flow heat exchangers. Also, a comparison is held between the needle and round fins. Air is flowing at a velocity range from 1 m/s to 5 m/s. Findings for heat transmission and pressure drop were confirmed by previous studies in the literature. Conclusion revealed that Nu increased range from 20 % to 30 % with a reduction in mass of heat exchanger about 23,8 %. In addition to that, fan power has increased by 10%.

Tahrour et al. [6] estimated the optimum tube location in a circular fin for heat dissipation and pressure drop. Fin spacing and tube diameter were studied for Re range 4500<Re<22500. At 12 mm tube shift and Re equal 9923, the eccentric annular-finned tube has 7.61% better h. 43.9% less pressure with gain.

Concentric, eccentric, perforated, and star-shaped finned tubes were compared by Tahrour et al. [74]. Fin design and spacing altered thermal flow from 4,300 to 15,000 Re. Increasing fin spacing from 2mm to 7mm increased j but decreased f and fin performance. Offshore energy systems favour star-shaped fins heat exchangers.

Wais [75] numerically evaluated the performance of a particular heat exchanger for alternative fin thickness and winglet shapes, including air flow and streamline deviations generated by the plate and tube arrangement. Adding winglets increases pressure drop by improving heat transmission.

The thermal performance of circular, axial, and triangular fins was estimated and optimised using finite element analysis [76]. ANSYS finite element study compares circular fins to axial and triangular fins. Moreover, thermal stress distribution has also been measured.

Numerical analysis of 16 fin-tube heat exchangers was performed by [77]. When Rayleigh numbers (Ra) equal 100, a circular fin-tube heat exchanger reaches its functional limit because the fin density is rarely affected by natural air flow or the fluid's boundary to heat. Fin performs effectively at 100 Ra. At Ra numbers 15, shorter fins or a greater s/D ratio improve functionality. In the second part of their study [78], they focused on the correlation of various circular fins. The circular fin-tube heat exchanger was categorised by Do/D = 1.2, where D and Do represent the tube and fin diameters, respectively. With CFD, the range of parameters was enlarged for the case with Do/D is ranged 1.2< Do/D≤10.

Flow and temperature fields for a cylinder with different radial fin directions, quantities, and elevations were studied [79]. The fins' impact on secondary frequencies, the effects of fin length and number on Strouhal number and drag coefficient were explored. By increasing fin elevation, the Strouhal number was lowered, and the drag coefficient was raised. Utilizing two fins acts as breaker plates, which diminishes the drag. However, the usage of extra fins. Adding fins changes this tendency. The investigation found that 4-fin cylinder cases perform best.

An annular elliptical fin tube row was compared to circular fins [80]. Where h and Δp are regarded as a function of diameter ratio, i.e., they are affected by both diameter ratio and elliptical orientation. With the same air velocity, vertical fin heat transfer is higher than circular. Lower pressure drop allows greater arrival velocity, which increases heat transfer. When space or pressure drop is limited, annular elliptical fin tubes may be a viable choice.

Elliptical and circular fins were analysed for their thermal and hydraulic properties [81]. The turbulent flow was simulated using transition SST. Air speed, fin diameter fractions, and fin densities were studied. Simulations showed elliptical fins work better. Pressure drop of elliptical fin reduced by 50% compared to a circular fin with equivalent heat transfer. Vertical elliptical annular fin transfer heat better than a circular fin. Normalized entropy production was explored to better explain the elliptical layout's second law advantage. Circular fins always generate more entropy than elliptical fins.

7.2Crimped fins

The thermal and hydraulic characteristics of circular fins (plain, crimped, trimmed) and plate fins with punched delta winglet pair (DWP) were studied [82]. Circular finned tubes had an efficiency of 77–83%, whereas plate fins had an efficiency of 55–66 %. The friction coefficients for both crimped and serrated finned tubes in hydraulics were more than one in plate fins, suggesting vigorous flow circulation. Circular finned tubes, on the other hand, outperformed plate fins in regards to h and the concept of heat transmission for every unit produced by the pump, requiring 140–170% more power than plate fins. The performance of crimped fins with a tube diameter (9.53 mm) at Re ranging 1500-6400 is experimentally studied [83]. Heat flux and pressure drop rose as fin dense and fin diameter fell. Fin pitch affects f significantly. However, Nu and j are affected insignificantly.

7.3Spiral fins

To lower the weight of finned tube heat exchangers, multiple trim-off angles included (30°, 45°, 60°, and 90°) were numerically studied at the back portion of helical sectioned fins by Martinez-Espinosa et al. [18]. Re-Normalization in order to represent the turbulence impact, the cut-cell technique was adopted. The findings showed that the thermal and hydraulic characteristics are identical when the cut-off angle is lower than 60°, where the stream split-up appears virtually. Moreover, at 45° and 60° cut-off angles, finned tube weight is reduced by 25% to 33%. The major impacts are considered part of modern finned tubes, which allow for less fin material and lighter heat exchangers. Compared to the benchmark case, it exhibited a 34.3 % reduction in transport energy but a reduction of 50% in fin material. Finally, fins with 60° or 45° as trimmed-off angles are proposed for lighter heat exchangers. The flow around segmented helical finned tubes was studied numerically by Salinas-Vázquez et al. [84] using Large Eddy Simulation. A numerical model using immersed boundary and pressure extrapolation in the tube's solid area. In the simulation, the flow is completely developed away from the tube bundle limits. The findings allow viewing and analyzing different turbulence forms. Two counter-rotating horseshoe eddies were found between the fins that may modify heat transmission. The mean velocity data generated two symmetric vortices behind the tubes. Their sizes and intensities were almost the same in each normal plane, regardless of fin placement.

Lindqvist and Næss [85] utilized the steady reynolds averaged navier-stokes (RANS) based turbulence simulations to create a CFD model for helical wound finned tubes. Two cases were used: solid (plain) aluminium fins and serrated fins. For the stated geometry, the heat transport values are equivalent to the older Holfeld [86] correlation. Pressure drop correlations are inconsistent, and no correlation captures numerical predictions for all geometries. Using a modelled fin temperature distribution, three efficiency coefficients are derived. For a number of semi-empirical correlations between Nu and Eu, a number of the geometries studied had non-conservative predictions. The findings support reduction domain modelling, which takes advantage of the geometric periodicity of the heat exchanger to decrease computational cost.

Three helically-wound finned tubes in a cross-flow heat exchanger, including whole (continuous) fins, incompletely serrated fins, and completely serrated fins were analyzed numerically by Cléirigh and Smith [8] using CFD with Re varying from 5,000 to 30,000. The exterior Nu and total pressure drop are anticipated by the CFD model in comparison to those expected by existing empirical correlations. According to the CFD results, the Nu raised to 23%

between partly and completely serrated fins. In addition, the actual correlations found in the literature do not accurately represent this susceptibility, but it is easily distinguishable utilizing the CFD model.

ANSYS analysed plain and serrated circular fins with cross-flow heat exchangers [9]. Six geometrical parameters, including outer tube diameter, elevation, fin dense, and fin width, are numerically examined. Thermal performance is 8.2% greater in the serrated fins compared to the plain fins, while flow performance is 7.5% lower. Nu and Eu are fin-length proportional. Nu had no effect on low Re values for fin pitch. At high Re levels, Nu increases with fin pitch, reversing the f. Serrated finned tubes had a Nu 10.3% higher than the entire finned tubes. Solid (whole) performed better at 10.5 %. The Nu grows with the segment height ratio.

Feng et al. [87] modelled unwelded integrated spiralled fins economizer thermal-hydraulic and fouling performance in 3D and investigated the influence of longitudinal and transverse tube bundle pitches on integrally-moulded spiral finned (IMSF) tube thermal-hydraulic and fouling performance at different Re. To reduce boiler system instability caused by the considerable pressure drop at the short longitudinal pitch, an economizer with a longitudinal pitch greater than 80 mm is suggested. Transverse pitch improved functioning but worsened fouling and pressure loss. Furthermore, new correlations for these innovative fins j and f coefficients were shown. Unwelded integrated spiralled fins were found to increase heat exchanger performance as a consequence.

The research employed large eddy simulation (LES) to describe the turbulent flow of a heat exchanger with spiralled fins, which outperforms prior two-equation solvers in determining flow structure around bluff bodies [88]. Using the Q-criteria, this study focuses on constructing coherent structures around spiral fins and generating flow topologies that improve heat transfer in spiral finned tubes. The research also looks at the turbulent flow on the fin surface with varying Re and Nu values. The correlation between Nu and heat flux throughout the time series at each sample point was 0.81. Diana et al. [89] compared the tube heat transfer process for both serrated and non-serrated fins at two different velocities of 7 and 9 m/s. Serrated fins have 0° and 30° parts per period. The outlet temperature for the notched fin was 350.15 K. Numerical investigations compared entropy generation rates of segmented (sectioned) and non-segmented configurations [90]. The fin was 16% more efficient when segmented. With 6% less area, the average convective coefficient rose 4.5%. Overall heat loss was 1% against the non-segmented design. Plain and serrated fin heat exchangers were explored by [91]. The analysis recommended that the entropy with the two methods of calculations were varied based on the convection process impacts assessed. The serrated fin has a considerable benefit compared to the plain fin (base). These statistics help design heat exchangers for aeroplanes. Lindqvist and Næss [92] examined five elements that impact heat transmission and pressure drop models' thermal-hydraulic performance, including heat flow direction, fin-type (circular or sectioned), fin efficiency, fin dense, tube layout, and entering turbulence degree. They conclude that flow outside the fin diameter may impair efficiency by 20%. A study simulated the flow across three sectioned tubes with high, low, and non-finned tubes were carried out [93]. When normalised by effective diameter, the lift coefficient for identical tubes for both finned and unfinned are similar. These results have an optimised significance for the design of weight. Mansour et al. [94] evaluated the influence of fin segment number (4, 8, 10, 12, and 14) and fin twisting on serrated finned-tube performance. The twisted angles evaluated are between 0° and 25°, and a study of finned tubes with and without fin serration found that Nu with serrated fins is raised by (61.9, 53.41, 27.8, and 12.6%) compared with a smooth tube. 14 serrated fins exhibited strong Nu and skin friction.

Perforated fins

The effect of perforated star-shaped finned tubes with different diameter holes of Ø2, Ø3, and Ø4 mm on performance was numerically analyzed by [11] with 2300<Re<16000 using CFD. At a temperature of 288k, air passes over the fins at speeds varying from 1 to 7m/s. The numerical result shows a mass drop of 17.65 % and an increase in heat transmission from 5.5 to 11.3 % for perforated star-shaped finned tubes with 4 mm hole diameter compared to solid star-shaped fins. Furthermore, perforated fins had 51.8 % less mass than circular fins, despite a 26.5 % increase in Nu. Banerjee et al. [7] aimed to optimize the overall transfer capabilities of a circular finned-tube system in order to augment its performance. Therefore, perforation seemed to be one of the choices to enhance heat transfer by a passive technique where circular cavities were drilled into the regular circular fins. The maximum heat flux and h, as well as the lowest pressure drop, were calculated for five cases with single perforations and three cases with multiple perforations. The fin heat flux and h performance ratios rose by 5.96% and 7.07%, respectively, in the perforated casing (perforations commencing at 60 at intervals of 30). Flow passed from fin 1 to fin 4 for h values between 90° and 180°, Nu grew for each perforated fin owing to improved blending and air disturbances in the top zone from second and third row inner fins. 120° perforations increased heat flux and h by 1.70 and 1.39 %, respectively. Multiple perforations cases (10°,150°,180°) showed the biggest growth in heat flux by 3.99% and pressure by 3.81% among all multiple cases. Multiple perforations cases (10°,150°,180°) showed the biggest growth in heat flux by 3.99% and pressure drop by 3.81% among all multiple cases. Fin pressure drop increased by 11.87 %. Increasing heat flux and h will increase pressure and price.

The general thermal performance of the various passive designs for cross-flow heat exchangers has been documented by Mangrulkar et al. [95]. They covered various tube cross-sections, fins with unique designs, and fins modifying parameters like dense tube ratio, eddy size, fin dense, Re, and by adding VGs. The outcome of fin preference studies indicated that using trimmed and helical fins are restricted to a spherical tube. However, slim basic fins and curving fins may be used with any tube cross-section. For helical and trimmed finned tubes, tube density might be limiting. No limits exist on the tube's pitch ratio. Also, smaller fins are better since they have the same temperature as the tubes, which enhances heat transfer and reduces pressure drop.

Flow parameters and thermal performance have been numerically investigated [96] by utilizing different positions of perforation angle (90º,120º, and 150º). Case 1 was the reference, case 2 had the holes at 90º, case 3 at 120º, case 4 at 150º, case 5 at 90º and 120º, case 6 at 90º and 150º, case 7 at 120º and 150º, case 8 at 90º,150º, and 120º. Perforated fins have a smaller pressure drop and better heat exchange than helical fins. Case 8's j is 11.7% higher than the first case. Due to increased backflow, perforation pressure drops. Cases (2, 5, 6, and 8) had pressure decreases within 2%. Case 6 has 8% less f than the reference case. Li et al. [97] investigated perforated fin construction by varying hole diameter and position. Two hole parameters impact latent heat thermal energy storage charging. Larger holes boost natural convection but reduce heat conduction. More heat conduction and natural convection will occur closer to the fin root. Perforated fins promote natural convection and minimise heat conduction. 3 mm hole diameter, 8 mm hole location, and 6 holes reduce melting time by 5.49 % and increase heat capacity by 0.21%.

8.Other studies

The research was conducted to compare numerical and experimental measurements using helically segmented (serrated) finned tubes [20], which are employed in commercial applications due to their lower weight than plain fins, capacity to induce disturbance, and reduced heat transfer surface. The ultimate goal is to provide periodic boundary condition requirements for discrete finned tube modules. Furthermore, average streamwise qualities are semi-performance, and heat transmission is basically the same in every finned tube in the fully formed flow zone. Both numerical and experimental tests were performed for circular finned tubes with various positions [98], L-footed spiral fin [99], and different fin shapes [100]. Heat transmission and pressure drop between eccentric and concentric circular fins were analysed [101]. The turbulence model (k–ω) is utilised to predict optimum tube placement in both staggered and aligned configurations. Re ranging (5500 - 29,700), layouts, and tube locations were investigated. Eccentric annular-finned tubes are superior for aligned and staggered heat exchangers, and this gain lowers the pressure drop.

Air flow crossing the toothed (perforated) circular fins has been analysed and tested [102]. The investigation targeted to improve the circular fins. The number and form of heat exchanger holes were evaluated. The experiment included a test setup with different heat exchanger types and 7500 to 17500 sensors. The six circular toothed fins model has a higher average Nu by 11.08 %. Considering pressure drop, triangular perforation finned heat exchangers have a greater Nu.

The performance of solid circular fins is impacted by fin elevation, fin gap, fin density, tube diameter, tube gap, row and bundle arrangement, which were highlighted by [103]. Temperature and flow gradients on fin surfaces need more localised activity data. Such impacts are hard to quantify without impacting fin heat transfer. All relevant circular finned-tube research is experimental, and numerical approaches have not proven linked connections. Better relationships need more numerical data. The thermal performance of helical, circular, non-finned, and cut circular fins is evaluated numerically [104]. The impact of fluid temperature and entrance velocity on helical heat transferred features was studied. The effects of fin elevation, width, axial fin dense, and circular fin dense on thermal and hydraulic functionality were explored [105]. The outcomes exhibited that fin elevation has the greatest influence on heat transferred and flow impedance and should be addressed when designing three-dimensional finned tubes.

9.Novel fin shapes application

For waste heat recovery applications, Naees [106] investigated the heat transmission and pressure drop in different trimmed fin tube bundles. He discovered that increasing heat transfer through the finned tube reduces the pumping power needed. Walter and Hofmann [107] investigated the impact of modified fins' heat transferred characteristics on a heat recovery steam generator. In other parts of the simulation, the Re was beyond the specified region of the Nu correlations and had to be enlarged. Correlations between Schmidt [108], Verein Deutscher Ingenieure which means the association of German engineers (VDI) [109], and extended surface corporation of America (ESCOATM) [110] (classic and updated) were used to compute stream over gas-side h for sectioned fins. The investigation revealed that varied relationships had a substantial similarity in the boiler's overall attitude. However, the comprehensive analysis of the boiler's performance revealed some major differences.

The impact of different parametric modifications on the total heat transfer and pressure drop of a single line of circumferential round fins in cross-flow was studied [111]. The results indicated that the expected global heat transfer increased with fin height (h_f). The increase in total heat transfer was substantial when fin spacing (s_f) was reduced. The pressure drop increased with h_f, with the increase being greater for smaller s_f; nonetheless, the increase in pressure loss with h_f is negligible for a fin distance per tube diameter fraction greater than one-third. The spacing and height of the fins had a significantly bigger impact on heat transfer and pressure reduction. Thicker fins boosted heat transmission and decreased pressure marginally. The thermal conductivity of material increased when its thermal impedance due to convection became larger than its thermal impedance owing to conduction. When s/h_f is greater than 1.5, the pressure decrease is almost constant. However, when the s/h_f ratio increases from 1.5 to 6, heat transport also increases. For the simulation values, building a finned tube with s/h_f> 1.5 is useless.

Experimental studies of thermal and pressure losses in ten trimmed fins for heat recovery were conducted by Naess [112]. When transverse (s_t) and diagonal (s_d) flow planes are equal, heat transmission is enhanced. It was discovered that fin characteristics had a considerable effect on heat transmission. Raising fin pitch to reduce heat transfer is more effective than increasing fin height. It has a negligible impact on pressure drop for st sd. At $s_t > s_d$, the fin pitch rises, but the fin height has a negligible influence on Eu.

Anoop et al. [17] explored conjugate heat transfer from serrated fins on a sodium-to-air heat exchanger in sodium reactors using k–ε model with Re from 500 to 10,000. Parametric studies were conducted on serration depth, pitch, height, and thickness. In addition to cross flow, flow angle affects heat transfer. Serration depth reduces heat transfer area. Solid fins reduce h by 10% and Nu by 6%. The deeper serration transferred as much heat as the solid fin. Serrations don't affect the finned tube's heat transfer. It decreases air heat exchanger weight. Serrations in the fins lead to a 7.5% weight decrease, reducing material costs. The heat transmission rate increased with increasing fin dense, lowering fin height, and increasing fin thickness. Nu over a serrated finned tube has been correlated. This will help develop sodium-to-air heat exchangers. Finally, a correlation for assessing the Nu for serrated fins has been exhibited, as presented in Equation 15.

$$Nu = 0.192Re^{0.647}\left(\frac{D_o}{h_f}\right)^{0.709}\left(\frac{s}{D_o}\right)^{0.171}\left(\frac{A_f}{A_{sf}}\right)^{0.274}\left(\frac{t}{D_o}\right)^{0.273} \quad (15)$$

Non-uniform fin location and pitch were found to improve melting rate by 84.7% in latent heat thermal energy storage (LHTES). Four cases were examined by Yang et al. [113], including the uniform fin distribution and the other three cases, the non-uniform fin distribution (circular, triangular, and parabolic). The non-uniform fins were studied experimentally and numerically for 2-wheeler heat dissipation [114]. Increased temperature gradient enhances heat transfer rate. In parabolic form, the heat transmission area is maximised. Parabolic fins improve heat dissipation and cooling rate. *Table 4* illustrates the comparative summarization of various modified circular fins.

Table 4 Advantages, limitations, operating conditions, and results for the studied modified circular fins

References	Fin Type	Approach	Advantage	Limitations	Operating conditions	Result
[7, 11, 68, 96, 97]	Perforated (toothed)	Numerical	↑ Nu ↑ Q ↑ j ↓ m	↑ Δp	2300<Re<16000 θ=90°, 120°, 150°, and 180°.	m drops 17.65% and an increase in Q from 5.5 to 11.3 % for toothed star-shaped fins with 4 mm as d. 26.5 % increase in Nu. For single perforations, 120° achieved the best significant results, with a 1.70% rise by 1.39% in fin performance ratio but only gain in ΔP performance ratio. Multiple perforations (10°,150°,180°) had the highest rise in Q (3.99%) with an increase in ΔP (3.81%). Q increased by 5.96% and in h by 7.07% in the perforated case holes at 60° at intervals of 30°. ΔP rose by 11.87%. 8 holes at 90º,150º, and 120º j increased by 11.7%. Holes at 90º and 150º have 8% less f. Toothed fins increase natural convection but reduce heat conduction. A 3 mm hole diameter, 8 mm hole position and 6 holes lower melting duration by 5.49% and boost heat capacity by 0.21%. Lowered thermal resistance by 7% to 12% at varied orientation angles and mass by 9%.
[19, 67]		Experimental	↓ W ↑ h	↓ Δp for holes at 60°	m_w 0.05-0.1 kg/s. θ=60°	W reduced by 6.62% and 9.42 % with four holes with R1.25 and R1.85. An increase of 8.87 % in h. Holes at 60° had 18% higher efficiency while losing 1.16 % less Δp than other angles.
[8, 9, 18, 90, 93, 94]	Spiral	CFD	↓ W ↑ Nu ↑ h ↓ A 50%↓ in fin material. ↑Thermal performance	↑ Cf	Re (5,000-30,000). θ=45° and 60°. N=(4, 8, 10, 12, and 14).	At 45° and 60° cut-off angles, finned tube W is reduced by 25% to 33%. Nu raised to 23% between partly and completely serrated fins. Thermal performance is 8.2% higher than that of the complete fin shape. Serrated fins had a Nu 10.3% higher than the entire fin. Nu grows with the segment height ratio Partition the raised performance by 16% over the non-sectioned fin. Considering sectioned 6% decreased in area, the average convective coefficient climbed by 4.5%. Found that the Nu with a serrated insert is raised by (61.9, 53.41, 27.8, and 12.6%) compared with a smooth tube. Serrated fins exhibited strong Nu and skin friction.
[16, 48-50, 55, 63, 64]		Experimental	↓ A ↑ Nu ↑ h ↑ Q_{ave}	↑ Eu 13% when fin height increased	Re (3000-30000). f_p, t_p,	Small f_p improves h regardless of arrangement or fin-type. Increase Nu is 30% for spiral fins and 10% for serrated fins. It also tends to grow when t_p drops. The influence of the tube layout was minimal. Higher fin heights resulted in more pressure decrease, Eu 13% when fin height increased. As fin thickness rose, fin tubes' heat transmission decreased. Louvre-spiral fin raised Q_{ave} and h_o by 9.7–15.6% and 13.4–27.1%, respectively.
[82]	Crimped	CFD	↑Q	↑ f ↑ fan power	Re (2500-4000). f_p, d.	Heat flux and pressure drop rose as fin pitch and diameter fell.
[42–46, 83]		Experimental	↑ Q	↑Δp ↓ j as Re↑ ↑ f	Re (1500-6400).	f_p and material type have no effect on heat transfer quality, and j. The number of tube lines increases the Q and Δp. Both j and f diminished as the Re climbed.
[15, 80, 81]	Elliptical	CFD	↓ Δp ↓ w	↓ Q (14%)	V=3.5 m/s	Pressure loss decreased by 31% along with heat flux and weight reduction by (23%).
		Experimental				
[2, 74]	Star-shaped	CFD	↑ Nu ↓ w ↑ j	↑ fan power	Re (2300-12000). Increasing f_p from 2mm to 7mm	Nu increased by (20- 30) % Weight reduced (23.8%) ↑ fan power (10%) enhanced the j and lowered f and fin performance
[13, 14]		Experimental		↑ fan power	Re (2000-	↑Q (35.5- 55.8) %

References	Fin Type	Approach	Advantage	Limitations	Operating conditions	Result
					13000)	↑ h (16.5-18.5) % ↓ w (23.5 %)
[17, 101, 111, 114]	Other modified shapes	Numerical	↓ Δp ↑ Q	↑ Δp with ↑ L	Re (5500-29,700). Eccentric, concentric, f_p, L, t, parabolic form.	Circular fins with eccentric and concentric radii. Eccentric annular fins have better thermal properties than concentric ones. This gain lowers Δp. The increase in Q was considerable at narrower f_p. The Δp rose with increased fin L. Thicker fins resulted in slightly increased heat transfer and Δp. decrease. In parabolic form, the heat transmission area is maximized. Parabolic fins improve heat dissipation and cooling rate.
[17, 112, 113]		Experimental	↑ Q ↓ material and costs.	↓ 10% in the heat transmission area	Re (500-10,000). S_t=S_l Serrations. L	When flow regions transverse and diagonal planes are equal, heat transfer is improved. The solid fin configuration features a decrease of 10% in heat transmission area and a diminution of 6% in Nu. Serrations in the fins lead to a 7.5% w decrease, reducing material costs. The Q increased with increasing fin dense, lowering L, and increasing fin thickness. Non-uniform fin location and f_p were found to improve melting rate uniformity by 84.7% in (LHTES).

10.Challenges to be addressed

These challenges have been identified by means of our complete survey and study of the modified circular fins, which have been explained in this review:

1. Circular fin was regarded as a gold standard for heat transfer. However, it has limited heat transfer features and a heavier heat exchanger weight. So, it cannot be applied widely. An alternative way to meet the problem is by modifying the circular fins' geometry (spiral, perforated, crimped, elliptical, and needle).
2. This review study found a significant heat transfer enhancement in the spiral serrated, perforated, and crimped fin geometries with a noticeable increase in the pressure drop and fan power, consequently.
3. More focus is required to identify how to sort the limitation of the increased pressure generated with the aforementioned fin geometries.
4. Elliptical and needle fin geometry is found to increase the model's performance by enhancing the heat transferred, along with minimizing the pressure drop compared to other modified geometries. Therefore, more efforts are demanded in exploring these geometries.

11.Discussion

This study reviews heat exchanger improvements by utilizing modified circular fins. Geometry-based and flow characteristics-based augmentation methodologies are presented in depth. These are the answers to the research questions addressed in this study.

Q1. What are the classifications of all the modified circular fins?

The classifications of modified circular fins have been presented in the present review as examination approaches-based (numerical and experimental) and geometry-based (spiralled, crimped, elliptical, toothed, and star-formed fins). All the novel fin shapes explored better heat exchanged features and a noticeable weight reduction. In spite of this, the impact of fin geometry on flow characteristics is incredibly complicated and relies on fin type and air velocity. The flow parameters of these modified forms vary, with some generating more pressure than others.

Q2. What are the current achievements in heat transfer, flow characteristics, and weight reduction with respect to the recent circular fin modifications?

In general, there are three major aims in optimizing the heat transmission and fluid flow in fin-and-tube.

The primary objectives are to maximize heat transmission, minimize pressure drop, and reduce weight. Five modified circular fins geometries that influence the thermo-hydraulic performance of finned heat exchangers were investigated. The parameters that reflect heat transfer enhancement include (Nu, h, j, Q) were considered, as well as the flow features include (Δp and f). The optimization results indicated that the Nu number for spiralled, elliptical, toothed, and star-formed fins are enhanced by 14.9%, 25%, 30%, and 35%, respectively, compared to the solid circular fin. Similarly, h and heat flux have the same behaviour. In contrast, the pressure drop is reduced by 13.9% for star-formed fin and increased in the other fin geometries (spiralled, crimped, and toothed fins). The demand for a lighter-finned heat exchanger has evolved. The outcomes from the aforementioned review revealed that modified fins reduced the weight by 30%, 23%, 9%, and 23.8% for spiralled, elliptical, toothed, and star-formed fins, respectively, compared to a solid fin.

Q3. There are so many scenarios of heat transfer modification. What would be the optimum configurations to accomplish that?

All the modified fin shapes explored better heat exchanged features and a noticeable weight reduction, as illustrated in *Figure 7*. Nevertheless, the effect of fin geometry on flow characteristics seems to be quite intricate and depends on the type of fin (spiralled, crimped, elliptical, toothed (perforated), and star formed or needle fins) and air velocity. By interrupting the main flow, the modified circular fins enhance the thermal-hydraulic performance of the finned tube. Star-formed fin enhances heat transferred compared to a circular fin due to the whirling vortex downstream. The outcomes exhibited that the optimal geometry is a star-shaped fin which led to a rise in heat transmission developments than the other modified circular fins reported in the literature. A star shape or needle fins showed the highest increase for heat flow by 35.5-55.8%, and the heat transferred coefficient (h) rose by 18.5% - 16.5%.

Q4. What are the major challenges to be addressed in circular fin modification?

The modified circular fin designs bring new problems. Increased pressure loss requires greater fan power to push gas over fins. Literature discovered that breaking the boundary reduced pressure loss and improved heat transfer compared to simple (circular) fins.

Limitations

All we discussed in this study was how the improved circular fins performed in thermohydraulic terms and how much lighter they were as a result. Other fin types, such as internal, longitudinal, and plate fins, were not addressed.

A complete list of abbreviations is shown in *Appendix I.*

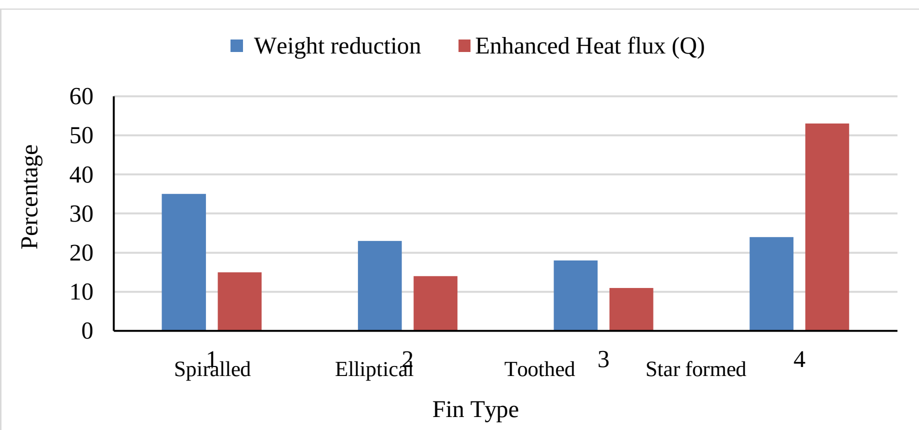

Figure 7 Modified circular fin achievements

12.Conclusion

According to the review, the impact of modified fin types on heat transmission is challenging. They improve the thermal and hydraulic performance of the finned tube by disrupting the main flow. Because of the swirling vortex downstream, a star-shaped fin improves heat transmission over a circular fin. Previous works examined numerous characteristics impacting heat transfer processes, such as fin height, number, material, and spacing between fins. Considering the fin's main role in transmitting heat, several articles depart from prior specifications to decrease the pressure drop for modern devices. Although several articles have advocated studying modified fins, reducing material's weight while boosting heat transmission and minimizing the pressure that affects fan power is currently being explored. Heat transfer needs further research. This involves optimizing the modified fins' architecture to increase heat transmission and minimize pressure drops. In other words, designing with requirements requires balance.

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Conflicts of interest
The authors have no conflicts of interest to declare.

Author's contribution statement
Rand Nabil: Conceptualization, investigation, design of the study, performed the statistical analysis, writing the original draft and editing. **Ali Sabri:** Supervision, revised the article for important intellectual content and approved the final version of the manuscript.

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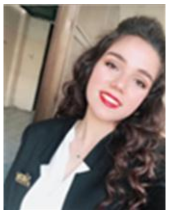

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Appendix I

S. No.	Abbreviation	Description
1	C_p	Specific Heat
2	CFD	Computational Fluid Dynamics
3	CFX	High Performance Computational Fluid Dynamics
4	DWP	Delta Winglet Pair
5	ESCOATM	Extended Surface Corporation Of America
6	Eu	Euler Number
7	f	Friction Factor
8	h	Heat Transfer Coefficient
9	h_f	Fin Height
10	HVAC	Heating, ventilation and air conditioning
11	IMSF	Integrally-Moulded Spiral Finned
12	j	Colburn Factor
13	LES	large eddy simulation
14	LHTES	Latent Heat Thermal Energy Storage
15	LMTD	Log Mean Temperature Difference
16	NTU	Number of Transfer Units
17	Nu	Nusselt Number
18	ΔP	Pressure Drop
19	PRISMA	Preferred Reporting Items for Systematic Reviews and Meta-Analyses
20	Q	Heat Flux
21	Ra	Rayleigh Number
22	Re	Reynold Number
23	RANS	Steady Reynolds Averaged Navier-Stokes
24	S_t	Transverse Tube Diameter
25	S_l	Longitudinal Tube Diameter
26	SST	Shear Stress Transport
27	S_f	Fin Pitch
28	t_f	Fin Thickness
29	T_{wall}	Wall Temperature
30	U	total heat exchanged
31	VDI	Association for German Engineers
32	V_{in}	Inlet Velocity
33	VG	Vortex Generation