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Review of Advances in Heat Transfer

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ABSTRACT: ***Different heat transfer enhancers are reviewed. They are (a) fins and microfins, (b) porous media, (c) large particles suspensions, (d) Nano fluids, (e) phase-change devices, (f) flexible seals, (g) flexible complex seals, (h) vortex generators, (i) protrusions, and (j) ultrahigh thermal conductivity composite materials. Most of heat transfer augmentation methods presented in the literature that assists fins and microfins in enhancing heat transfer are reviewed. Among these are using joint-fins, fin roots, fin networks, biconvections, permeable fins, porous fins, capsulated liquid metal fins, and helical micro fins. It is found that not much agreement exists between works of the different authors regarding single phase heat transfer augmented with micro fins. However, too many works having sufficient agreements have been done in the case of two phase heat transfer augmented with microfins. With respect to nanofluids, there are still many conflicts among the published works about both heat transfer enhancement levels and the corresponding mechanisms of augmentations. The reasons beyond these conflicts are reviewed. In addition, this paper describes flow and heat transfer in porous media as a well-modeled passive enhancement method. It is found that there are very few works which dealt with heat transfer enhancements using systems supported with flexible/flexible-complex seals. Eventually, many recent works related to passive augmentations of heat transfer using vortex generators, protrusions, and ultra-high thermal conductivity composite material are reviewed. Finally, theoretical enhancement factors along with many heat transfer correlations are presented in this paper for each enhancer..***

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1. INTRODUCTION

The way to improve heat transfer performance is referred to as heat transfer enhancement (or augmentation or intensification). Nowadays, a significant number of thermal engineering researchers are seeking for new enhancing heat transfer methods between surfaces and the surrounding fluid. Due to this fact, Bergles [1, 2] classified the mech- anisms of enhancing heat transfer as active or passive methods. Those which require external power to maintain the enhancement mechanism are named active methods. Examples of active enhancement methods are well stirring the fluid or vibrating the surface [3]. Hagge and Junkhan [4] described various active mechanical enhancing methods that can be used to enhance heat transfer. On the other hand, the passive enhancement methods are those which do not require external power to sustain the enhancements' characteristics. Examples of passive enhancing methods are: (a) treated surfaces, (b) rough surfaces, (c) extended surfaces, (d) displaced enhancement devices, (e) swirl flow devices, (f) coiled tubes, (g) surface tension devices, (h) additives for fluids, and many others.

2. MECHANISMS OF AUGMENTATION OF HEAT TRANSFER

To the best knowledge of the authors, the mechanisms of heat transfer enhancement can be at least one of the following.

(1) Use of a secondary heat transfer surface.

(2) Disruption of the unenhanced fluid velocity.

(3) Disruption of the laminar sublayer in the turbulent boundary layer.

(4) Introducing secondary flows.

(5) Promoting boundary-layer separation.

(6) Promoting flow attachment/reattachment.

(7) Enhancing effective thermal conductivity of the fluid under static conditions.

(8) Enhancing effective thermal conductivity of the fluid under dynamic conditions.

(9) Delaying the boundary layer development. (10) Thermal dispersion.

(11) Increasing the order of the fluid molecules. (12) Redistribution of the flow.

(13) Modification of radiative property of the convective medium.

(14) Increasing the difference between the surface and fluid temperatures.

(15) Increasing fluid flow rate passively.

(16) Increasing the thermal conductivity of the solid phase using special nanotechnology fabrications.

Methods using mechanisms no. (1) and no. (2) include increasing the surface area in contact with the fluid to be heated or cooled by using fins, intentionally promoting turbulence in the wall zone employing surface roughness and tall/short fins, and inducing secondary flows by creating swirl flow through the use of helical/spiral fin geometry and twisted tapes. This tends to increase the effective flow length of the fluid through the tube, which increases heat transfer but also the pressure drop. For internal helical fins however, the effect of swirl tends to decrease or vanish all together at higher helix angles since the fluid flow then simply passes axially over the fins [5]. On the other hand, for twisted tape inserts, the main contribution to the heat transfer augmentation is due to the effect of the induced swirl. Due to the form drag and increased turbulence caused by the disruption, the pressure drop with flow inside an enhanced tube always exceeds that obtained with a plain tube for the same length, flow rate, and diameter.

Turbulent flow in a tube exhibits a low-velocity flow region immediately adjacent to the wall, known as the laminar sublayer, with velocity approaching zero at the wall. Most of the thermal resistance occurs in this low-velocity region. Any roughness or enhancement technique that disturbs the laminar sublayer will enhance the heat transfer [6]. For example, in a smooth tube of 25.4 mm inside diameter, at Re = 30, 000, the laminar sublayer thickness is only 0.0762 mm under fully developed flow conditions.

The internal roughness of the tube surface is well-known to increase the turbulent heat transfer coefficient. Therefore, for the example at hand, an enhancement technique employing a roughness or fin element of height ∼ 0.07 mm will disrupt the laminar sublayer and will thus enhance the heat transfer.

Accordingly, mechanism no. (3) is a particularly important heat transfer mechanism for augmenting heat transfer.

Li et al. [5] described the flow structure in helically finned tubes using flow visualization by means of high-speed photography employing the hydrogen bubble technique. They used four tubes with rounded ribs having helix angles between 38◦ and 80◦ and one or three fin starts, in their investigation. Photographs taken by them showed that in laminar flow, bubbles follow parabolic patterns whereas in the turbulent flow, these patterns break down because of random separation vortices. Also, for tubes with helical ridges, transition to turbulent flow was observed at lower Reynolds numbers compared to smooth tube values. Although swirl flow was observed for all tubes in the turbulent flow regime, the effect of the swirl was observed to decrease at higher helix angles. Li et al. [5] concluded that spiral flow and boundary-layer separation flow both occurred in helical- ridging tubes, but with different intensities in tubes having different configurations. As such, mechanisms no. (4) and no. (5) are also important heat transfer mechanisms for augmenting heat transfer.

Arman and Rabas [7] discussed the turbulent flow structure as the flow passes over a two-dimensional transverse rib. They identified the various flow separation and reattachment/redevelopment regions as: (a) a small recirculation region in front of the rib, (b) a recirculation region after the rib, (c) a boundary layer reattachment/redevelopment region on the downstream surface, and finally (d) flow up and over the subsequent rib. The authors noting that recirculation eddies are formed above these flow regions, identified two peaks that occur in the local heat transfer-one at the top of the rib and the other in the downstream recirculation zone just before the reattachment point. They also stated that heat transfer enhancement increases substantially with increasing Prandtl number. Therefore, the mechanism no. (6) plays an important role in heat transfer enhancements.

Heat transfer enhancements associated with fully/partially filing the fluidic volume by the porous medium take place by the following mechanisms [8, 9].

(7) Enhancing effective thermal conductivity of the fluid under static conditions.

(8) Enhancing effective thermal conductivity of the fluid under dynamic conditions.

(9) Delaying the boundary layer development. (10) Thermal dispersion.

(11) Increasing the order of the fluid molecules. (12) Redistribution of the flow.

(13) Modification of radiative property of the convective medium.

Ding et al. [10] showed that fluids containing 0.5 wt.% of carbon nanotubes (CNT) can produce heat transfer enhancements over 250% at Re = 800, and the maximum enhancement occurs at an axial distance of approximately 110 times the tube diameter. These types of mixtures are named in the literature as “nanofluids” and it will discussed later on in this report. The increases in heat transfer due to presence of nanofluids are thought to be associated with the following mechanisms. (7) Enhancing effective thermal conductivity of the fluid under static conditions. (8) Enhancing effective thermal conductivity of the fluid under dynamic conditions. (9) Delaying the boundary layer development. (10) Thermal dispersion. (11) Increasing the order of the fluid molecules. (12) Redistribution of the flow.

Flexible fluidic thin films were introduced in the work of Khaled and Vafai [11, 12] and Khaled [13]. In their works, they describe a new passive method for enhancing the cooling capability of fluidic thin films. In summary, flexible thin films utilize soft seals to separate between their plates instead of having rigid thin film construction. Khaled and Vafai [11] have demonstrated that more cooling is achievable when flexible fluidic thin films are utilized. The expansion of the flexible thin film including flexible microchannel heat sink is directly related to the average internal pressure inside the microchannel. Additional increase in the pressure drop across the flexible microchannl not only increases the average velocity but also, it expands the microchannel causing an apparent increase in the coolant flow rate which, in turn, increases the cooling capacity of the thin film. Khaled and Vafai [12] and Khaled [13] have demonstrated that the cooling effect of flexible thin films can be further enhanced if the supporting soft seals contain closed cavities filled with a gas which is in contact with the heated plate boundary of the thin film. They referred to this kind of sealing assembly as “flexible complex seals”. The resulting fluidic thin film device is expandable according to an increase in the working internal pressure or an increase in the heated plate temperature. Therefore, mechanism no. (15) can have an impact in enhancing heat transfer inside thermal systems. Finally, the mechanism no. (14) finds its applications when rooted fins are utilized as an enhancer of heat transfer [14]. Finally, the last mechanism will be discussed when the topic of ultra-high thermal conductivity composite materials is discussed.

3. HEAT TRANSFER ENHANCERS

From the concise summary about mechanisms of enhancing heat transfer described in last section, it can be concluded that these mechanisms cannot be achieved without the presence of the enhancing elements. These elements will be called as “heat transfer enhancers”. In this report, the following heat transfer enhancers will be explained: 3.1. Extended surfaces (Fins); 3.2. Porous media; 3.3. Large particles suspensions; 3.4. Nanofluids; 3.5. Phase change devices; 3.6. Flexible seals; 3.7. Flexible complex seals; 3.8. Vortex generators; 3.9. Protrusions; and 3.10. Ultra high thermal conductivity composite materials.

3.1. Heat Transfer Enhancement Using Extended Surfaces (Fins)

3.1.1. Introduction.

Fins are quite often found in industry, especially in heat exchanger industry as in finned tubes of double-pipe, shell-and-tube and compact heat exchangers [15–20]. As an example, fins are used in air cooled finned tube heat exchangers like car radiators and heat rejection devices. Also, they are used in refrigeration systems and in condensing central heating exchangers. Moreover, fins are also utilized in cooling of large heat flux electronic devices as well as in cooling of gas turbine blades [21]. Fins are also used in thermal storage heat exchanger systems including phase change materials [22–25]. To the best knowledge of the authors, fins as passive elements for enhancing heat transfer rates are classified according to the following criteria.

(1) Geometrical design of the fin. (2) Fins arrangements. (3) Number of fluidic reservoirs interacting with the fin. (4) Location of the fin base with respect to the solid boundary. (5) Composition of the fin.

According to design aspects, fins can have simple designs, such as rectangular, triangular, parabolic, annular, and pin rod fins. On the other hand, fin design can be complicated such as spiral fins [26, 27]. In addition, fins can have simple network as in finned tubes heat exchangers [20]. In contrast, they can be arranged in a complex network as can be seen in the works of [28–30]. Moreover, fins can be further classified based on the fact whether they interact thermally with a single fluid reservoir or with two different fluid reservoirs.

Example of works based on the last classification is the works of Khaled [31, 32]. In addition, fins can be attached to the surface as in the works [15–25] or they may have roots in the heated/cooled walls [13]. Finally, fins can be solid [3, 4, 15–25] or they can be porous [33] or permeable [34].

3.1.2. Laminar Single-Phase Heat Transfer in Finned Tubes.

Laminar flow generally results in low heat-transfer coefficients and the fluid velocity and temperature vary across the entire flow channel width so that the thermal resistance is not just in the region near the wall as in turbulent flow. Hence, small-scale surface roughness is not effective in enhancing heat transfer in laminar flow; the enhancement techniques employ some method of swirling the flow or creating turbulence [6]. Laminar flow heat transfer and pressure drop in “microfin” tubes (discussed later) was experimentally measured by [35]. Their data showed that the heat transfer and pressure drop in microfin tubes were just slightly higher than in plain tubes and they recommended that microfin tubes not be used for laminar flow conditions. This outcome has also been confirmed in the investigations of Shome and Jensen [36] who concluded that “microfinned tube and tubes with fewer numbers of tall fins are ineffective in laminar flows with moderate free convection, variable viscosity, and entrance effects as they result in little or no heat transfer enhancement at the expense of fairly large pressure drop penalty”.

3.1.3. Turbulent Single-Phase Heat Transfer in Finned Tubes.

Turbulent flow and heat transfer in finned tubes has been widely studied in the past and the literature available on the experimental investigations of turbulent flow and heat transfer in finned tubes is quite extensive. One of the earliest experimental work on the heat transfer and pressure drop characteristics of single-phase flows in internally finned tubes dates back to 1964 when Hilding and Coogan [37] presented their data for ten different internal fin geometries for a 0.55 in. (14 mm) inner diameter copper tube with 0.01 in. (0.254 mm) straight brass fins using air as the test fluid. Hilding and Coogan [37] observed that the heat transfer is enhanced by around 100–200% over that of the smooth tube and the enhancement is accompanied by a similar increase in the pressure drop. The Reynolds number in this study ranged from 15, 00 to 50, 000.

Kalinin and Yarkho [38] used different fluids in the range $1500 \le Re \le 400,000$ and $7 \le Pr \le 50$ to investigate the effect of Reynolds and Prandtl numbers on the effectiveness of heat transfer enhancement in smoothly outlined internally grooved tubes. The ranges of the transverse groove heights tested were $0.983 \le d/D \le 0.875$, (where d is the fin tip diameter and D is the inner diameter) with maximum groove spacing equal to the pipe nominal diameter. They reported that the critical Reynolds number at which transition to turbulent flow occurs decreased from 2400 for a smooth tube to 1580 at e/D = 0.875 and fin spacing equal to half the pipe diameter for a grooved tube, with a maximum increase in the heat transfer coefficient of up to 2.2 times the minimum measured value. The authors also observed that the behavior of the Nusselt number for the tested range of Prandtl numbers is independent of the Prandtl number.

In their two papers, Vasilchenko and Barbaritskaya [39, 40] published their results for the heat transfer and pressure drop of turbulent oil flow in straight finned tubes with 4 ≤ N ≤ 8 and 0.13 ≤ e/D ≤ 0.3, for an operating condition range of 103 ≤ Re ≤ 104 and 70 ≤ Pr ≤ 140.

Their results showed that the heat transfer is enhanced by 30% to 70% over that of smooth tubes for the finned tubes tested. Correlations for predicting the friction factors and the Nusselt numbers were also presented.

In the work of Bergles et al. [41], heat transfer and pressure drop data for straight and spiral finned tubes of fin heights from 0.77 to 3.3 mm with water as the working fluid was investigated. The Reynolds number based on the hydraulic diameter ranged from around 1, 500 to 50, 000. They found an earlier transition from laminar to turbulent flow and their friction factor data indicated that the smooth tube friction factor correlations could also be used for the tested finned tubes in the turbulent region. The heat transfer coefficients were found to be up to twice that of comparable smooth tubes. From their heat transfer data, they concluded that the hydraulic diameter approach is effective for correlation only in the case of straight fins of moderate heights.

Watkinson et al. [42, 43] in their two separate works, performed experiments for water and air flows, respectively, in a tube-in-tube heat exchanger under isothermal heating conditions to study the turbulent heat transfer and pressure drop characteristics of straight and helically finned tubes. A total of eighteen tubes, 5 with straight fins and 13 with spiral fins, having internal fin geometries with fin starts from 6 to 50, 0.026 ≤ e/D ≤ 0.158, helix angles from 0◦ to 15◦, and inside diameters of 0.420 to 1.196 inch were examined for 7 × 103 ≤ Re ≤ 3 × 105 and 0.7 ≤ Pr ≤ 3.4. A commercial smooth copper tube was also tested for comparison. The air results presented show that for most tubes, from Re = 50, 000 up to Re = 300, 000 the heat transfer is enhanced up to 95% over that of a smooth tube. On the other hand, in water flow tests, at Re = 50, 000 heat transfer is enhanced up to 87% over that of a smooth tube, but at higher Reynolds number, the finned tubes approached smooth tube heat transfer performance. A maximum increase of 100% in the pressure drop over smooth tube was observed for tubes with tall helical fins. Separate empirical nondimensional heat transfer correlations were presented for water and air, for both straight and spiral fin tubes, having a form similar to the smooth tube turbulent Sieder-Tate correlation with additional parameters consisting of the ratios of the inter fin spacing to the tube diameter and the fin pitch to the tube diameter. For straight fin tubes, the inter fin spacing- to-diameter ratio and for spiral finned tubes the pitch-to-diameter ratio were incorporated to form a modified Blasius- type correlation for predicting the friction factor. These correlations predicted their data to within a maximum error of 13%.

Carnavos [44] tested eight finned tubes (both straight and helically finned) to obtain heat transfer and pressure drop data for cooling of air in turbulent flow employing a double tube heat exchanger. The tubes tested had fin starts from 6 to 16 and helix angles from 2.5◦ to 20◦. The results were presented on the hydraulic diameter basis for the range 104 ≤ Reh ≤ 105 and correlations were proposed to predict the heat transfer and the pressure drop. The reported heat transfer correlation was in the form of a modified Dittus-Boelter single-phase correlation having an additional correction factor "F", consisting of the ratios of the nominal heat transfer area to the actual heat transfer area and the actual flow area to the core flow area, respectively. The secant of the helix angle raised to the third power was also included in "F". The friction factor was also in the form of a modified Blasius type equation with a correction factor "F ∗" comprising of the ratio of the actual flow area-to-the nominal flow area and the secant of the helix angle. In a later investigation, Carnavos [45] used the same apparatus and three more tubes with number of fin starts and helix angles up to 38◦ and 30◦, respectively, to extend his air results by including experimental data conducted for heating of water and a 50% w/w ethylene glycol-water solution.

The data for air obtained earlier [44] was reexamined and a set of correlations that predicted their entire data obtained with air,

water, and ethylene glycol-water solution to within ±10%, were proposed in the ranges 104 ≤ Reh ≤ 105 , 0.7 ≤ Pr ≤ 30, and 0 ≤ α ≤ 30◦. The Nusselt number correlation was allowed to retain its original form whereas the helix angle dependency in the friction factor correlation was slightly changed. Armstrong and Bergles [46] conducted experiments for electric heating of air in the range 9, 000 ≤ Re ≤ 120, 000 and Pr = 0.71, using seven different silicon carbide finned tubes all having straight fins. The tubes tested had fin starts from 8 to 24 and e/D from 0.06 to 0.15. The results indicated that the heat transfer is enhanced by around 30–100% over that of a smooth silicon carbide tube. Their heat transfer data was predicted to within ±20% by the Carnavos [45] heat transfer correlation, but a large disagreement was observed between the measured friction factors and those predicted by the Carnavos [45] friction factor correlation.

Since their introduction more than 20 years ago, “microfin” tubes have received a lot of attention playing a very significant role in modern, high-efficiency heat transfer systems. Microfin enhancements are of special interest because the amount of extra material required for microfin tubing is much less than that required for other types of internally finned tubes [47]. Of the many enhancement techniques which have been proposed, passively enhanced tubes are relatively easy to manufacture, cost-effective for many applications, and can be used for retrofitting existing units, whereas active methods, such as vibrating tubes, are costly and complex [48]. Moreover, these tubes ensure a large heat transfer enhancement with a relatively small increase in the pressure drop penalty. Microfin tubes are typically made of copper and have an outside diameter between 4 and 15 mm. The principal geometric parameters that characterize these tubes [49] are: the external diameter, fin height (from 0.075 mm to 0.4 mm), helix angle (from 10◦ to 35◦), and the number of fin starts (from 50◦ to 60◦). These dimensions are in contrast to other types of internal finning that seldom exceed 30 fins per inch and fin heights that range from several factors higher than the microfin tube height. Currently, tubes with axial and helical fins, in rectangular, triangular, trape-zoidal, crosshatched, and herringbone patterns are available. Important dimensionless geometric variables of an internally microfinned tubes include the dimensionless fin height (ε/D, fin height/internal diameter) and the dimensionless fin pitch (p/ε, fin spacing/fin height). A microfin tube typically has 0.02 ≤ ε/D ≤ 0.04 and 1.5 ≤ p/D ≤ 2.5, [50]. As microfinnned tubes are typically used in evaporators and condensers, thus most of the extensive existing research literature on microfinned tube performance characteristics is devoted to two-phase refrigerant flows. Schlager et al. [51] and Khanpara et al. [52] are typical examples of such investigations, showing a 50% to 100% increase in boiling and condensation heat transfer coefficients with only a 20% to 50% increase in pressure drop. However, single-phase performance of microfinned tubes is also an important consideration in the design of refrigeration condensers as a substantial proportion of the heat transfer area of these condensers is taken up in the desuperheating and later subcooling of the refrigerant. Consequently, accurate correlations for predicting the single-phase heat transfer and pressure drop inside microfinned tubes are necessary in order to predict the performance of these condensers and to optimize the design of the system.

Khanpara et al. [52] investigated the heat transfer characteristics of R-113, testing eight microfinned tubes in the range 60 ≤ N ≤ 70, 0.005 ≤ e/D ≤ 0.02, and helix angles from 8◦ to 25◦, for 5 × 103 ≤ Re ≤ 11 × 103 . The results presented for the single phase heating of R-113 indicated that the heat transfer is enhanced by around 30%–80%. The authors concluded that a major part of the enhancement is due to the increase in the area available for heat transfer and a part of the enhancement is due to flow separation and flow swirling effects induced by the helical fins. This is because the corresponding increase in the heat transfer area over that in a smooth tube is around 10%–50% for the tubes tested. In a subsequent paper, Khanpara et al. [53] also reported the local heat transfer coefficients for single-phase liquid R-22 and R-113 flowing through a smooth and an internally finned tube of 9.52 mm outer diameter, in their paper on in-tube evaporation and condensation characteristics of microfinned tubes. The single-phase experiments were performed by direct electrical heating of the tube walls in the Reynolds number range of 5, 000 to 11, 000 for R-113 and 21, 000 to 41, 000 for R-22. The microfinned tube had of 60 fin starts, a fin height of 0.22 mm, and a helix angle of approximately 17◦. The heat transfer coefficients for the internally finned tube were found to be 50% to a 150% higher than the smooth tube values.

Al-Fahed et al. [54] experimentally tested a single microfinned tube with a 15.9 mm outside diameter having 70 helical fins with fin height of 0.3 mm and a helix angle of 18◦ using water as the test fluid in a tube-in-tube heat exchanger. Results were presented for isothermal heating conditions in the range 10, 000 < Re < 30, 000. Under the same conditions, comparative experiments with an internally smooth tube were also conducted. They noted that the heat transfer is enhanced by 20%–80% and the pressure drop is increased by around 30%–80% as compared to the smooth tube values. The experimentally obtained friction and heat transfer data were correlated as a Blasius and a Sieder-Tate type correlation, respectively. The heat transfer correlation predicted their data to within ±25%, showing a large error band while no error band was reported for the friction factor results. The authors reasoned that at Re > 25, 000 the heat transfer enhancement ratio is moderate plausibly because at higher Re numbers the turbulence effect in microfinned tubes becomes similar to that in a plain tube.

Chiou et al. [55] conducted an experimental study with water using two internally finned tubes having the same outer diameter equal to 0.375 in. (9.52 mm). The two tubes had 60 and 65 fins, fin heights of 0.008 in (0.20 mm) and 0.01 in. (0.25 mm), and helix angles of 18 and 25◦, respectively. The Reynolds number in this study ranged from about 4, 000 to 30, 000. Modified Dittus-Boelter type correlations were formulated to predict the value of the heat transfer coefficient for flow Reynolds number greater than about 15000 and 13000, respectively, for each tube.

An available heat-momentum analogy based correlation for rough tubes along with a set of constitutive equations for calculating related roughness parameters was utilized to propose correlations for predicting the friction factor and the Nussult number for the entire range of the Reynolds number tested.

Brognaux et al. [50] obtained experimental single-phase heat transfer coefficients and friction factors for three single- grooved and three cross-grooved microfin tubes all having an equivalent diameter of 14.57 mm,a fin height of 0.35 mm, and 78 helical fins; only the fin helix angle was allowed to vary between 17.5◦ and 27◦ Using liquid water and air as the test fluid the experiments were carried out in a double-pipe heat exchanger. Results were presented for cooling conditions in the range 2500 < Re < 50, 000 and 0.7 < Pr < 7.85. Validation experiments with an internally smooth tube were also conducted using water and air. Compared to a smooth tube, the maximum heat transfer enhancement reported was 95% with a pressure drop increase of 80% for water at Pr = 6.8. They also found that the friction factors in microfin tubes do not reach a constant value at high Reynolds numbers as is usually observed in rough pipes. The authors also used their data in the range 0.7 < Pr < 7.85 (using only 2 of the tested tubes) to analyze the dependence of the heat transfer on the Prandtl number exponent. Using the heat- momentum transfer analogy as applied for rough surfaces, they presented their experimental heat transfer and friction factor data as functions of the “roughness Reynolds number” and from cross-plots deduced the Prandtl number exponent to be between 0.56-0.57. The Prandtl number exponent between 0.55–0.57 was also determined for the power law formulation $Nu = C Re^m Pr^n$. The authors also defined an “efficiency index” (which gives the ratio of the increase in heat transfer to the increase in friction factor for a finned and plain tube, resp.) and presented its value for the different tubes tested. The higher the efficiency index, the better the enhancement geometry.

Huq et al. [56] presented experimental heat transfer and friction data for turbulent air flow in a tube having internal fins in the entrance region as well as in the fully-developed region. The tube/fin assembly was cast from aluminum to avoid any thermal contact resistance. The uniformly heated test section was 15.2 m in length and the inner diameter of the tube was 70 mm which contained six equally spaced fins of height 15 mm. The Reynolds number based on hydraulic diameter ranged from 2.6×10^4 to 7.9×10^4 .

The results presented by the authors exhibited high pressure gradients and high heat transfer coefficients in the entrance region, approaching the fully developed asymptotic values away from the entrance section. The enhancement of heat transfer rate due to integral fins was reported to be very significant over the entire range of flow rates studied in this experiment. Heat transfer coefficient, based on inside diameter and nominal area of finned tube exceeded unfinned tube values by 97% to 112% for the tested Reynolds number range. When compared at constant pumping power, an improvement as high as 52% was also observed for the overall heat transfer rate.

With the expressed objective of developing physically based generally applicable correlations for Nusselt number and friction factor for the finned tube geometry, Jensen and Vlakancic [57] carried out a detailed experimental investigation of turbulent fluid flow in internally finned tubes covering a wide range of fin geometric and operating conditions. Two geometrically identical double pipe heat exchangers were used. The test fluid (water and ethylene glycol were used) flowed through the tube side of each of the heat exchangers in counter-flow with hot water in one test section and cold water in the other. Friction factor tests were also conducted under isothermal conditions. A total of sixteen pairs of tubes (15 finned and one smooth tube) with a wide range of geometric variations (inside diameter 24.64–21.18 mm, helix angles 0◦–45◦, fin height 0.18mm–2.06 mm, and number of fins 8–54) were tested. In the reported results, the authors first described the parametric effects of different fin geometries on turbulent friction factors and Nusselt numbers in internally finned tubes and then go on to prescribe a criterion for labeling a tube as a "high" fin tube ($2e/D > 0.06$) or a "micro" fin tube. They stated that a microfin tube is characterized by its peculiar pressure drop behavior with long lasting transitional flow up to Re = 20, 000. Trends in the reported data are different depending on whether the tube is a high-fin or a microfin tube. High fin tubes show friction factors curves similar to those of a smooth tube, only displaced higher with the friction factor increasing as the number of fins increases. For microfin tubes, in general, the friction factor is insensitive to the fin height and the Reynolds number up to Re = 20, 000, but beyond this value the friction factor showed a decreasing trend with increasing Re as in smooth tubes and the effect of number of fins, fin height, and the helix angle also comes into play, whenever anyone of these parameter is increased the friction factor increased (exceptions may occur due to difference in fin profile).

Overall the reported increase in friction factor for the high finned tubes ranged from 40%–170% and in microfin tubes from 40%–140%, over smooth tubes. For both types of tubes the reported trends of the slope of the Nu curves generally followed that of the smooth tube; however, the trends revealed a different slope at lower Re for the two categories of tubes. This characteristic was attributed by the authors to the greater capacity of swirling flow for higher finned tubes. However, the trends with geometry were similar to those noted for the friction factors. Overall, the reported increase in Nu for the high finned tubes ranged from 50%–150% and in microfin tubes from 20%–220% over smooth tubes. They reported that the correlations from the literature poorly predict their data and based on the findings from the trends observed went on to develop new correlations for friction factors and Nusselt numbers separately for the two categories of tube categories (high and microfin) identified by them. These correlations are applicable to a wide range of geometric and flow conditions for both categories of tubes and estimated well their data as well as the data from the literature.

Webb et al. [58] investigated the heat transfer and fluid flow characteristics of internally helical ribbed tubes. Using liquid water as the test fluid the experiments were carried out in a double-pipe heat exchanger. Results were presented for cooling conditions in the range 20, 000 < Re < 80, 000 and 5.08 < Pr < 6.29. A total of eight tubes (7 ribbed and one smooth tube) all having an inside diameter of 15.54 mm but a wide range of geometric variations (helix angles 25◦–45◦, rib height 0.327◦mm–0.554 mm, and number of fin starts 10–45) were tested. The authors presented power law based empirical correlations using their experimental data for the Colburn j-factor and the fanning friction factor, which predicted their data reasonably well. The finned tube performance efficiency index as defined by Brognaux et al.

[50] was also determined for the tubes tested, from which the authors concluded that the two key factors that affect the increase of the heat transfer coefficient in helically-ribbed tubes are the area increase and fluid mixing in the inter fin region caused by flow separation and reattachment, and the combination of the two determines the level of the heat transfer enhancement.

Copetti et al. [49] tested a single internally microfinned tube of 9.52 mm diameter using water as the test fluid.

Microfin height was 0.20 mm, fin helix angle was 18◦, and number of fin starts was 60. Results were presented for uniform heating conditions in the range $2,000 < Re < 20,000$. Under the same conditions, comparative experiments with an internally smooth tube were also conducted. They noted that the microfin tube provides higher heat transfer performance than the smooth tube although the pressure drop increase is also substantial (in turbulent flow $h_{microfin} / h_{smooth} = 2.9$ and $\Delta p_{microfin} / \Delta p_{smooth} = 1.7$ at the maximum Reynolds number tested). The finned tube performance efficiency index as defined by Brognaux et al. [50] were also determined which showed that the heat transfer increase was always superior to the pressure drop penalty. The experimentally obtained Nusselt numbers were empirically correlated separately as a Dittus-Boelter, a Sieder- Tate, and a Gnielinski type correlation. These correlations predicted their data reasonably well.

Wang and Rose [59] compiled an experimental database of twenty-one microfin tubes, covering a wide range of tube and fin geometric dimensions, Reynolds number and including data for water, R11, and ethylene glycol for friction factor for single-phase flow in spirally grooved, horizontal microfin tubes. The tubes had inside diameter at the fin root between 6.46mm and 24.13 mm, fin height between 0.13mm and 0.47 mm, fin pitch between 0.32mm and 1.15 mm, and helix angle between 17◦ and 45◦. The Reynolds number ranged from 2.0×10^3 to 1.63×10^5. Six earlier friction factor correlations, each based on restricted data sets, were compared with the database as a whole. They reported that none was found to be in good agreement with all of the data and indicated that the Jensen and Vlakancic [57] correlation was found to be the best and represented their database within ±21%.

Han and Lee [60] obtained experimental single-phase heat transfer coefficients and friction factors for four micro finned tubes all with 60 helical fins using liquid water as the test fluid in a double-pipe heat exchanger. The tubes tested had fin helix angles between 9.2 and 25.2◦ and fin height between 0.12 mm and 0.15 mm. Results were presented for cooling conditions in the range $3000 < Re < 40,000$ and $4 < Pr < 7$. Validation experiments with an internally smooth tube were also conducted. Using the heat- momentum transfer analogy, as used by Brognaux et al. [50], they presented their experimentally determined heat transfer and friction factor correlations as functions of the roughness Reynolds number, Re_ε, with a mean deviation and root mean square deviation of less than 6.4%. They noted that the microfin tubes show an earlier achievement of the fully rough region which starts at $Re_\varepsilon = 70$ for rough pipes and also validated the finding of Brognaux et al. [50] that the friction factors in microfin tubes do not reach a constant value at high Reynolds numbers as is usually observed in rough pipes surface. No attempt was made to present a direct comparison of heat transfer enhancement between a smooth and a microfinned tube, but an efficiency index was defined. Smaller value of efficiency index means increased friction penalty to establish a given enhancement level. Using this index, the authors noted that the tubes with higher relative roughness and smaller spiral angle show a better heat transfer performance than tubes with larger spiral angle and smaller relative roughness. The authors concluded that the heat transfer area augmentation by higher relative roughness is the main contributor to the efficiency index.

Li et al. [61] experimentally investigated the single-phase pressure drop and heat transfer in a microfin tube with a 19 mm outside diameter having 82 helical fins with a fin height of 0.3 mm and a helix angle of 25.5◦ using oil and water as the test fluid in a tube-in-tube heat exchanger. Results were presented for cooling conditions in the range $2500 < Re < 90,000$ and $3.2 < Pr < 220$. The pressure drop data were collected under adiabatic conditions. Under the same conditions, comparative experiments with an internally smooth tube were also conducted. Their results showed that there is a critical Reynolds number, Re_{cr} for heat transfer enhancement. For $Re < Re_{cr}$, the heat transfer in the microfin tube is the same as that in a smooth tube, but for Reynolds numbers higher than Re_{cr}, the heat transfer in the microfin tube is gradually enhanced compared with a smooth tube. It reaches more than twice that in a smooth tube for Reynolds numbers greater than 30, 000 with water as the working fluid. They attributed this behavior to the decrease in the thickness of the viscous sublayer with increasing Reynolds numbers. When the microfins are inside the viscous sublayer, the heat transfer is not enhanced, while when the microfins are higher than the viscous sublayer, heat transfer is enhanced. They also investigated the Prandtl number dependency of the Nusselt number in the form of NuαPrn and found that the Nusselt number is proportional to Pr0.56 in the enhanced region and is proportional to Pr0.3 in the nonenhanced region. For the high Prandtl number working fluid (oil, 80 < Pr < 220), the critical Reynolds number for heat transfer enhancement is about 6000, while for the low Prandtl number working fluid (water, 3.2 < Pr < 5.8), the critical Reynolds number for heat transfer enhancement is about 10, 000. The reported friction factors in the microfin tube are almost the same as for a smooth tube for Reynolds numbers below 10, 000. They become higher for Re > 10, 000 and reach values 40%–50% greater than that in a smooth tubes for Re > 30, 000. They also concluded that the friction factors in the microfin tube do not behave as in a fully rough tube even at a Reynolds number of 90, 000.

An artificial neural network (ANN) approach was applied by Zdaniuk et al. [62] to correlate experimentally determined Colburn j-factors and Fanning friction factors for flow of liquid water in straight tubes with internal helical fins. Experimental data came from eight enhanced tubes reported later in Zdaniuk et al. [63]. The performance of the neural networks was found to be superior compared to the corresponding power-law regressions. The ANNs were subsequently used to predict data of other researchers but the results were less accurate. The ANN training database was expanded to include experimental data from two independent investigations. The ANNs trained with the combined database showed satisfactory results, and were superior to algebraic power-law correlations developed with the combined database.

Siddique and Alhazmy [64] also tested a single internally microfinned tube with a nominal inside diameter of 7.38 mm. Microfin height was 0.20 mm, helix angle was 18◦, and number of fin starts was 50. Experiments were conducted in a double pipe heat exchanger with water as the cooling as well as the heating fluid for six sets of runs. The pressure drop data were collected under isothermal conditions-

Data were taken for turbulent flow with 3300 ≤ Re ≤ 22, 500 and 2.9 ≤ Pr ≤ 4.7. The heat transfer data were correlated by a Dittus- Boelter type correlation, while the pressure drop data were correlated by a Blasius type correlation. These correlations predicted their data to within 9% and 1%, respectively. The correlation predicted values for both the Nusselt number and the friction factors were compared with other studies. They found that the Nusselt numbers obtained from their correlation fall in the middle region between the Copetti et al. [49] and the Gnielinski [65] smooth tube correlation predicted Nusselt number values. For pressure drop results, they reported the existence of a transition zone for Re < 11, 500 in which the friction factor data exhibited a local maxima. The presented correlation predicted friction factors values were nearly double that of the Blasius smooth tube correlation predicted friction factors. The authors concluded that the rough tube Gnielinski [65] and Haaland [66] correlations can be used as a good approximation to predict the finned tube Nusselt number and friction factor, respectively, in the tested Reynolds number range.

Zdaniuk et al. [63] experimentally determined the heat transfer coefficients and friction factors for eight helically finned tubes and one smooth tube using liquid water at Reynolds numbers ranging between 12, 000 and 60, 000. The helically-finned tested tubes had helix angles between 25◦ and 48◦, number of fin starts between 10 and 45, and fin height-to-diameter ratios between 0.0199 and 0.0327. Power-law correlations for Fanning friction and Colburn j-factors were developed using a least-squares regression using five simple groups of parameters identified by Webb et al. [58]. The performance of the correlations was evaluated with independent data of Jensen and Vlakancic [57] and Webb et al. [58] with average prediction errors in the 30% to 40% range. The authors also gave recommendations about the use of some specific tubes used in their experimentations and concluded that disagreements in the experimental results of Webb et al. [58], Jensen and Vlakancic [57], and their own study imply that a broader database of heat transfer and friction characteristics of flow in helically ribbed tubes is desirable. The authors further recommended that more research should be performed on the influence of geometric parameters on flow patterns, especially in the inter fin region using modern flow visualization techniques or proven computational fluid dynamics (CFD) tools.

In a subsequent analysis, Zdaniuk et al. [67] using genetic programming extended their earlier work [63] presenting a linear regression approach to correlate experimentally-determined Colburn j-factors and Fanning friction factors for flow of liquid water in helically finned tubes. Experimental data came from the eight enhanced tubes used in their previous study [63] discussed above. This new study revealed that, in helically finned tubes, logarithms of both friction and Colburn j-factors can be correlated with linear combinations of the same five simple groups of parameters identified in their earlier work [63] and a constant. The proposed functional relationship was tested with independent experimental data yielding excellent results. The authors concluded that the performance of their proposed correlations is much better than that of the power law correlations and only slightly worse than that of the artificial neural networks.

More recently, Webb [68] investigated the heat transfer and friction characteristics of three tubes (19.05 mm O.D.,

17.32 mm I.D.) including one developed by the author (designated as tube T3) having a conical, three-dimensional roughness on the inner tube surface with water flow in the tube. Experiments were conducted in a double tube heat exchanger with water as the cooling as well as the heating fluid. The pressure drop data were collected under adiabatic conditions. The data were taken at a tube side Reynolds number range of 4000–24, 000 and the Prandtl number varied from 6.6 to 5.9. The heat transfer data were correlated by a Dittus Boelter type correlation, while the pressure drop data were correlated by a Blasius type correlation. The measured maximum uncertainty in the friction factor was reported to be 5.96%, while for most data points, the uncertainty in the measured value of the inside heat transfer coefficient was stated to be 8%. The experimentally obtained values for the Nusselt number were compared with an independent study and were found to be 9%–12% higher. The author reported that the TC3 truncated cone tube provides a Nusselt number 3.74 times that of a plain tube, but it has it has nearly 60% higher pressure drop and concluded that the three-dimensional roughness offers potential for considerably higher heat transfer enhancement (e.g., 50% higher) than is given by helical ridged tubes. Accelerated particulate fouling data were also provided for TC3 tube, and for five different helical-ribbed tubes for 1300 ppm foulant concentration at 1.07 m/s water velocity (Re = 16, 000).

The fouling rate was compared with helical-rib geometries reported earlier by Li and Webb [69]. The author noted that the TC3 tube shows a very high accelerated particulate fouling rate, which is higher than that of the helical-ribbed tubes tested by Li and Webb [69] and recommended that the 3-D roughness tubes should experience minimal and acceptable low fouling, if used with relatively clean or treated water.

Power law-based correlation for the friction factors and Nusselt numbers were presented in the ranges of 300 < Re < 3000 (laminar), 3000 < Re < 7000 (transition), and Re > 7000 (turbulent). They noted that in the laminar and turbulent ranges of Re, Nu is almost doubled compared to its value in a smooth tube as predicted by the Dittus Boelter correlation. But, for the transition range 3000 < Re < 7000, the Nu-data almost coincide with the same smooth- tube correlation predicted value, indicating no enhancement in heat transfer in this range. Constant pumping power comparison with smooth tube showed that the spirally grooved tube without twisted tape yields maximum heat transferdata do exist for predicting pressure drop and heat transfer in turbulent flow in finned tubes, but it also seems that there is a substantial disagreement between the results predicted by these different correlations; therefore, a need exists for further research in this area

4. CONCLUSIONS

In this paper, the following heat transfer enhancers are described and reviewed: (a) extended surfaces including fins and microfins, (b) porous media, (c) large particles suspensions, (d) nanofluids, (e) phase-change devices, (f) flexible seals, (g) flexible complex seals, (h) vortex generators, (i) protrusions, and (j) ultra-high thermal conductivity composite materials. Different research works about each one have been reviewed and many methods that assist their enhancement effects have been extracted from the literature.

Among of these methods presented in the literature are using joint-fins, fin roots, fin networks, bi convections, permeable fins, porous fins, helical microfins, and using complicated designs of twisted tapes. It was concluded that more attention should be made towards single phase heat transfer augmented with microfins in order to alleviate the disagreements between the works of the different authors.

Also, it was found that additional attention should be made towards uncovering the main mechanisms of heat transfer enhancements due to the presence of nanofluids. Moreover, we concluded that perhaps the successful modeling of flow and heat transfer inside porous media, which is a well- recognized passive enhancement method, could help in well discovering the mechanism of heat transfer enhancements due to nanofluids. This is due to some similarities between both media. In addition, it is concluded that noticeable attentions from researchers are required towards further modeling flow and heat transfer inside convective media supported by flexible/flexible-complex seals in order too compute their levels of heat transfer enhancements. Eventu- ally, many recent works related to passive augmentations of heat transfer using vortex generators, protrusions, and ultra-high thermal conductivity composite material have been reviewed. Finally,

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