Thermal benefits from the use of single and compound heat transfer enhancement by twisted tape and wire-coil inserts

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Abstract:

This study presents the results of the experimental investigation related to a single and compound heat transfer enhancement by tube inserts as twisted tape, wire-coil, and their combination. The study has been conducted with water (Pr = 4-10) as a working fluid in the range Re = 250-10000. The geometrical parameters of the inserts are as follows: twisted tape with pitch ratios $p / D_i = 3.14, 3.66$, and wire-coil with pitch-to-height ratios $p / e = 10.0,\ 12.0$. Hydraulic and thermal characteristics are presented as the Fanning friction factor ratio $f_* = f(\mathrm{Re})$, and the Nusselt number ratio $Nu_* = f(\mathrm{Re})$. The thermal benefit has been evaluated by the use of two criteria: FG-1a (increase in the heat flow) and FN-1 (decrease in the heat transfer surface area). An additional constraint is the augmentation entropy generation number less than unity, $N_{sa} < 1$.

These criteria can be used in their two modifications: (a) when the surface thermal performance is defined in a two-fluid heat exchanger where outside thermal resistance is available (shell-and-tube heat exchanger); (b) when the surface thermal performance is defined in the case of lack of thermal resistance outside of the wall (electrical heating of the tube wall or solar collectors). The criterion PEC (fixed pumping power) is also considered to show the many erroneous results and misunderstood conclusions that could arise. The use of these criteria permits a real evaluation of the benefits that can be achieved. The results revealed that the greatest thermal energy benefits could be obtained in the transient fluid flow regime between laminar and turbulent flow.

Keywords:

Heat transfer enhancement; Twisted tape; Wire-Coil; Compound Heat transfer; Benefits.

1. Introduction

Among the many existing techniques that are used to enhance the heat transfer for internal single-phase flows in channels and tubes, the most popular are the classical tube inserts as wire-coil and twisted tape. They are easy to install and operate, and their usage is to retrofit existing heat exchangers to increase the heat transfer capacity or reduce the heat transfer surface area when a new heat exchanger is designed [1-3].

The geometrical parameter that defines the insertion of a twisted tape (TT) into a horizontal tube is the twist ratio $y = p / D_i$, where *p* is the 180-degree twist pitch, and D_i is the tube inner diameter. Due to the potential to enhance heat transfer, many investigations have been conducted to study tape inserts in different conditions. Mousa et al. [4], focusing on changing the geometry of TTs either by introducing notches, perforations, serrations, or breaks to obtain a higher heat transfer enhancement.

Like TTs, the wire-coil insert generates swirl flow but mainly near the tube wall. The enhanced heat transfer is due to the swirl juxtaposed with the mainstream flow that generates centrifugal forces, causing high-temperature fluid in the boundary layer to move towards the center of the tube. Furthermore, the wire-coil acts as a turbulence promoter in the laminar sublayer of turbulent flows and has the same effect as artificial roughness as that of internally corrugated and ribbed tubes [1-3]. The geometrical parameters of the wire-coil are the ratios e/D_i and p/e, where p is the wire-coil pitch, whereas e is the wire diameter.

Detailed studies relative to the enhanced heat transfer with wire-coil inserts implemented in smooth tubes can be found in [5-9]. Promvonge [10] investigated a compound heat transfer enhancement by a combination of both wire-coil and TT inserts, Fig. 1, in an attempt to connect the benefits of both effects acting simultaneously. The study reported that the Nusselt number has increased tremendously by 300%–685% depending on p/e (wire-coil pitch ratio) and y (tape twist ratio) for Reynolds numbers ranging from 3000 to 18000.

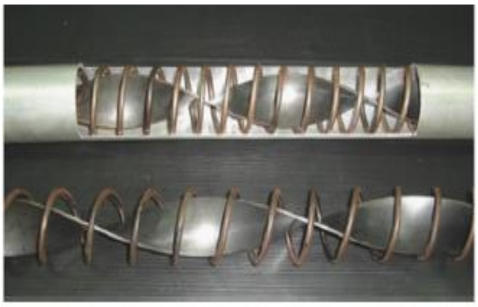

Figure 1. The combination of twisted tape and wire-coil inserts in tube (from Promvonge et al. 2008).

The objective of this study is to investigate the compound heat transfer enhancement by wire-coil insert in combination with twisted tape with water as a working fluid in the range of Reynolds number $\mathrm{Re} = 250 - 10000$. The benefits will be assessed by two different criteria, FG-1a and FN-1, according to the objectives pursued and the constrains imposed.

2. Experimental results

Details from the experimental program, setup design (counter-current water flow double-pipe heat exchanger), and verification of the results have been presented in Zimparov et al. [11], Zimparov et al. [12], and Zimparov et al. [13]. The geometrical parameters of the smooth tube, twisted tapes and wire coils under study are presented in Table 1, together with the ratios e/D_i, p/e and p/D_i.

Figure 2 presents the variation of the average values of f_a with Re for twisted tapes TT01 and TT02, together with the curve of f_s for the smooth pipe in the range $\mathrm{Re} = 250 - 10^4$. The results revealed that the twisted tapes studied almost coincide. The second important observation is that the transition between laminar and turbulent flow is smooth, contrary to that in the smooth tube.

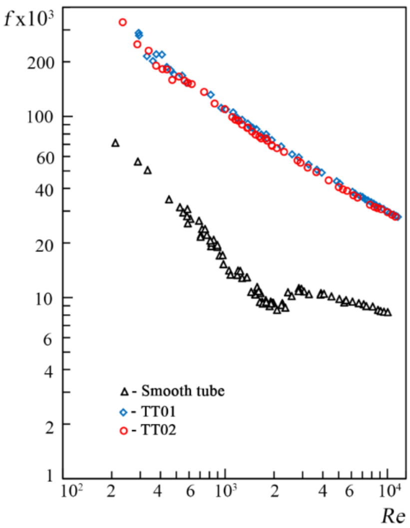

Figure 2. The variation of the friction factor f with Re

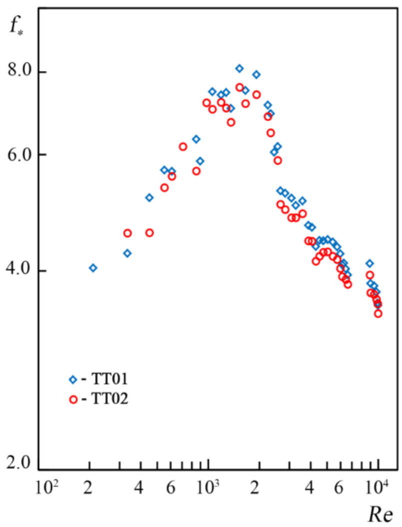

Figure 3. The variation of the friction ratio f_* with Re

Figure 3 presents the ratio $f_* = f_a / f_s$ according to the experimental results with Reynolds number. As seen, for the low Reynolds numbers the ratio f_* gradually increases up to $f_* \approx 8.0$ ($\mathrm{Re} \approx 1800$), and after that decreases reaching a value 3.5, $\mathrm{Re} = 10^4$.

Table 1. Geometrical parameters of a smooth tube, twisted tape and wire-coil inserts.

Tube	D_i mm	p mm	e mm	p/e -	e/D_i -	e/D_i -
TT01	14.0	44.0	-	-	-	3.14
TT02	14.0	51.3	-	-	-	3.66
TT03	14.0	37.4	-	-	-	2.67
TT04	14.0	41.9	-	-	-	2.99
TT05	14.0	56.0	-	-	-	4.00
WC01	14.0	10.0	1.0	10.0	0.071	0.71
WC02	14.0	12.5	1.0	12.5	0.071	0.89

The variation of the ratio $f_* = f_a / f_s$ with Re, for the different insert configurations, is depicted in Fig. 3. As seen the maximum increase in f_* is at $\mathrm{Re} \approx 1800$ for all geometrical configurations. It has to be noted that the increased in the friction ratio, when compound enhanced heat transfer technique is used, reaches almost 3 times compared to the single technique implemented.

Figure 4 depicts the variation of f_a vs. Re for wire-coil inserts WC01 and WC02. It is obvious that the curves have different behavior compared with the twisted tape inserts. These curves are similar to the curve f_s vs. Re for the smooth pipe. The difference is only for the start of the transition from laminar to turbulent flow, first critical number, $\mathrm{Re}_{cr,1} \approx 800$ (wire-coil insert) vs. $\mathrm{Re}_{cr,2} \approx 2000$ (smooth pipe). The variation of f_* vs. Re is shown in Fig. 5. As seen, in the range $200 < \mathrm{Re} < 1800$ f_* does not depend on p/e and it depends only on Re However, this dependence is different for $200 < \mathrm{Re} < 800$ and $800 < \mathrm{Re} < 1800$. When $\mathrm{Re} > 1800$, f_* depends simultaneously on p/e and Re, and it increases when p/e decreases.

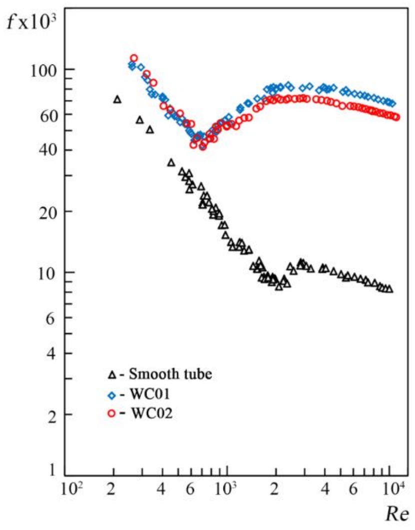

Figure 4. The variation of friction facto f with Re

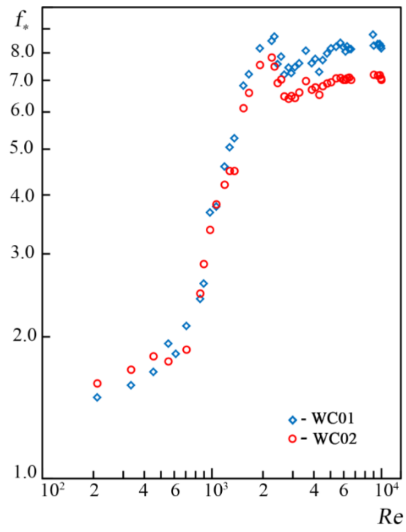

Figure 5. The variation of friction ratio f_* with Re

Figure 6 represents the results for the friction factor, f with Re when a combined enhanced heat transfer technique is implemented, namely, the combinations WC01+TT01 and WC01+TT03, Table 1. As expected, in the case where the values of f are greater compared with those from the single heat transfer enhancement. As seen, in the range $\mathrm{Re} < 2000$, the greatest values of f are intrinsic to the combination WC01+TT05, greater than the combination WC01+TT03. If the single enhanced heat transfer technique is applied, the twisted tape TT01 possesses the bigger f values than the wire-coil insert WC01. However, when $\mathrm{Re} > 2000$ it is observed, the opposite tendency. The combination WC01+TT03 has values of f greater than those of the combination WC01+TT05. Similarly, the values of f for the wire-coil insert WC01 are greater than those of the twisted tape TT01.

The variation of the ratio f_* with Re is depicted in Fig. 7, and as seen the maximum increase in f_* is at $\mathrm{Re} \approx 1800$ for all geometrical configurations. It has to be noted that the increased in the friction ratio when compound enhanced heat transfer technique is used reaches almost 3 times compared to the single technique used.

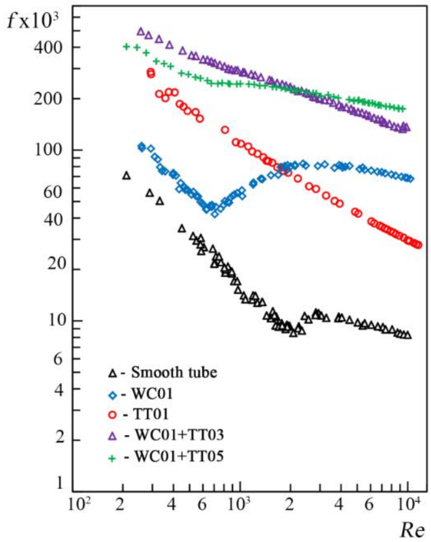

Figure 6. The variation of the f with Re

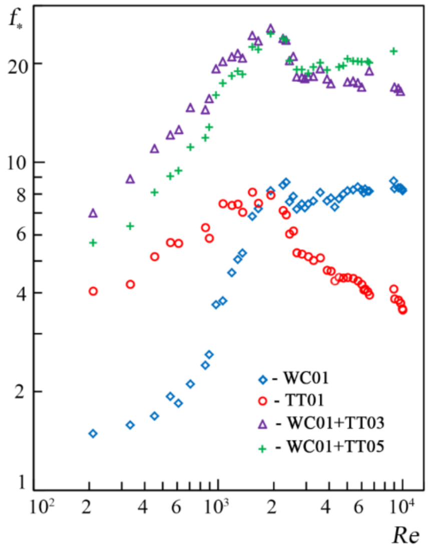

Figure 7. The variation of the f_* with Re

The variation of Nu vs. Re is depicted in Figs. 8-10, in the form $Nu\,\mathrm{Pr}^{-0.42}$ vs. Re, and the ratio Nu_* vs. Re in Fig. 11. An interesting feature (see Fig. 9) is that in the range $\mathrm{Re} < 800$, the wire-coil inserts possess negative enhancement, $Nu_a < Nu_s$.

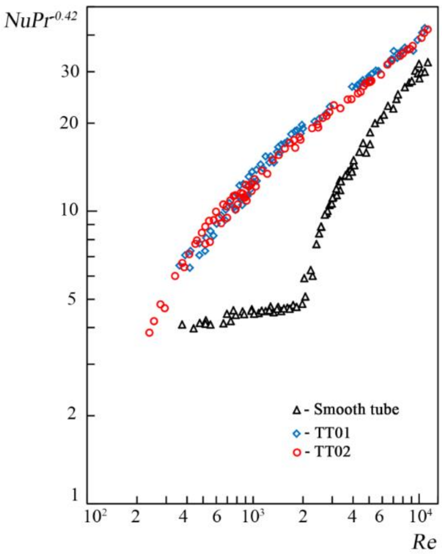

Figure 8. The variation of $Nu\,\mathrm{Pr}^{-0.42}$ vs. Re

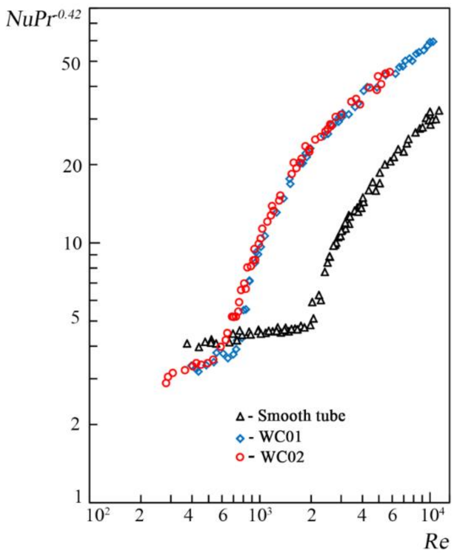

Fig. 9. The variation of $Nu\,\mathrm{Pr}^{-0.42}$ vs. Re

Figure 11 presents a comparison between the thermal characteristics of all inserts studied and the value of the increase in $Nu_* = Nu_a / Nu_s$ with Re. As seen, if the insert combination WC01+TT03 is implemented, it will bring about the greatest increase in the heat transfer coefficient. As expected, the maximum benefit can be obtained at the end of the transition region, and this benefit is as follows: $Nu_* \sim 4$ for the twisted tapes, $Nu_* \sim 6$ for the wire-coils, and $Nu_* \sim 7$ for the combination of both of them. Another feature is that in the turbulent region $\mathrm{Re} > 2000$, Nu_* decreases gradually. The greatest value of Nu_* (for $\mathrm{Re} \sim 10^4$) is again when the combined enhanced heat transfer technique is applied, $Nu_* \sim 3$.

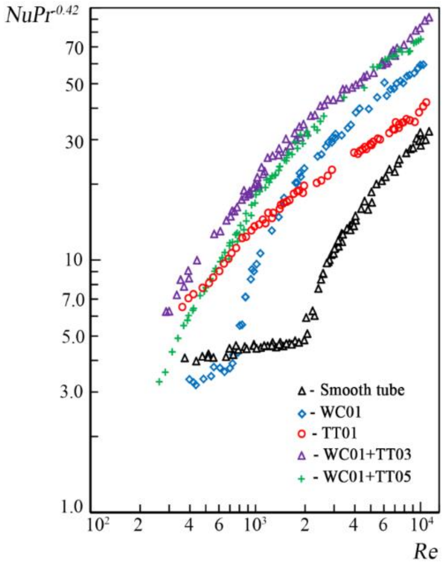

Figure 10. The variation of $Nu\,\mathrm{Pr}^{-0.42}$ vs Re

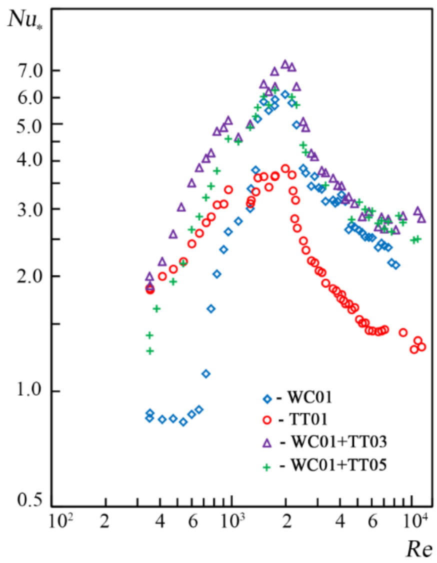

Figure 11. The variation of Nu_* vs. Re

3. Performance evaluation criteria

In a recent paper, Zimparov et all. [14] suggested that the imposed constraint for fixed pumping power to be removed from the list of constraints and replaced with the constraint of the augmentation entropy generation number $N_{\mathrm{sa}} \leq 1$, defined as

$$N_{sa} = \frac{1}{1+\phi_{\mathrm{s}}}\left(N_T + \phi_{\mathrm{s}} N_P\right), \tag{1}$$

where N_T and N_P are the values of N_{sa} in the limits $\phi_s \to 0$ and $\phi_s \to \infty$, whereas ϕ_s is the *irreversibility distribution ratio* [15].

The benefit equations are discussed and documented in detail in [2,9,12]. The fixed geometry criteria FG-1a require a replacement of smooth tubes in the heat exchanger with augmented tubes of equal diameter and length, and may be regarded as "retrofit" applications. The criterion FN-1-new is used to assess the decreased in the heat transfer surface area in new heat exchanger design.

3.1. Case FG-1a

The objective of the case FG-1a is an increase in the heat flow, $Q_* > 1$ with the constraints $W_* = 1$ $A_* = 1$ (with $D_* = 1$, $N_* = 1$ and $L_* = 1$), $\Delta T_{\mathrm{i}}^* = 1$, and additional constraint $N_{\mathrm{sa}} \leq 1$ [9]. The relative equations for heat flow and augmentation entropy generation number are [15],

$$Q_* = \varepsilon_* \tag{2}$$

$$N_{sa} = \frac{1}{1+\phi_s}\left\{\frac{Q_*^2}{Nu_*} + \phi_s f_*\right\}. \tag{3}$$

where the ratio, ϕ_s, has to be calculated following Bejan [15, Chapter 6, p. 120].

3.2. Case FN-1-new

The objective of this case is to reduce the heat transfer surface area by reducing the tubing length, $L_* < 1$. The constraints are: $N_* = 1$, $Q_* = 1$, $W_* = 1$, $\Delta T_i^* = 1$ and $N_{sa} \le 1$. Then,

$$L_* = \frac{Nu_*^{-1} + r_s}{1 + r_s} \tag{4}$$

and

$$N_{sa} = \frac{1}{1+\phi_s}\left(\frac{1}{Nu_* L_*} + \phi_s f_* L_*\right). \tag{5}$$

4. Results and discussion

4.1. Case FG-1a

When the experimental data for the heat transfer coefficient and friction factor are collected in a two-fluid heat exchanger, the existence of the outside fluid thermal resistance impacts the increase in the heat flow $Q_* > 1$, and it has to be taken into consideration. Figures 12 and 13 show the variation of Q_* and N_{sa} for single and compound heat transfer enhancement, when the tube outside thermal resistance is negligible or absent. As seen, the increase in the heat flow, $Q_* > 1$, is greater when the compound heat enhancement technique is used than the single enhancement technique. This effect is more visible in the laminar flow and transitional flow, $\mathrm{Re} < 2\times10^3$, where the increase in the heat flow reaches $Q_* = 2.3$. However for $\mathrm{Re} > 2\times10^3$, $N_{sa} > 1$, and the compound heat transfer technique is also inefficient.

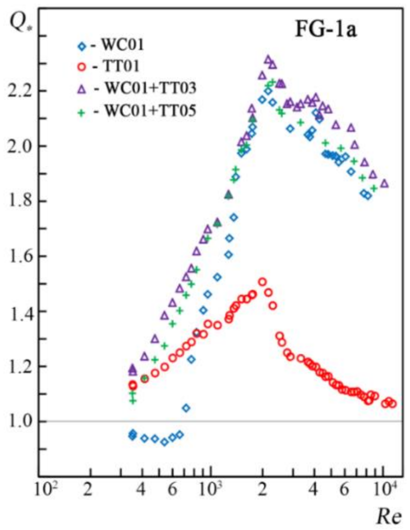

Figure 12. The variations of Q_* with Re

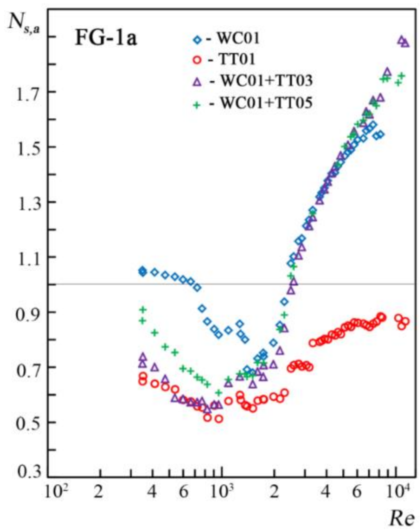

Figure 13. The variations of N_{sa} with Re

In this case, an interesting fact should be noted: the values of N_{sa} for insert TT01 are the smallest ones compared to all others in the range of Reynolds numbers studied. Then, the question is: “Which is the preferable heat transfer enhancement technique?” The answer to this question can be obtained if the next criterion is used

$$N_s^+ = \frac{N_{sa}}{Q_*} < 1. \tag{6}$$

This criterion is used if two objectives are pursued simultaneously: a greater increase in the heat flow and a smaller entropy generation. Fig. 14 presents the variation of N_s^+ with Re, using the data from Figs. 12 and 13. As seen in the range, $\mathrm{Re} < 2\times10^3$ the best choice is the compound technique WC01+TT03.

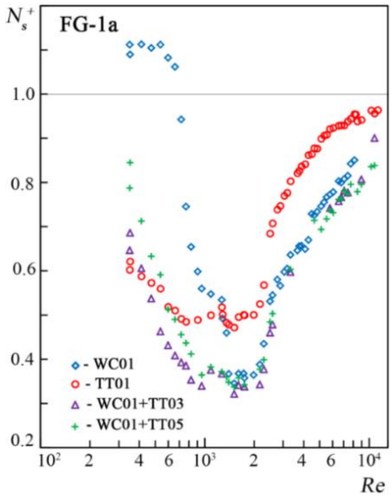

Figure 14. The variations of N_s^+ with Re

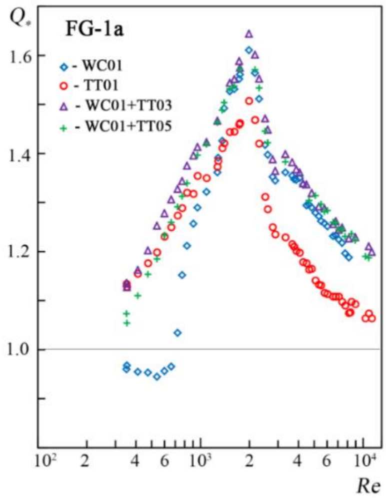

Figure 15. The variations of Q_* with Re

However, for a two-fluid heat exchanger, the real evaluation of the benefit is when the tube outside thermal resistance is taken into consideration. That is why the thermal resistant $1/h_o A_o = 0.011$ K/W from the experiment has been implemented, and the result is depicted in Figs. 15.

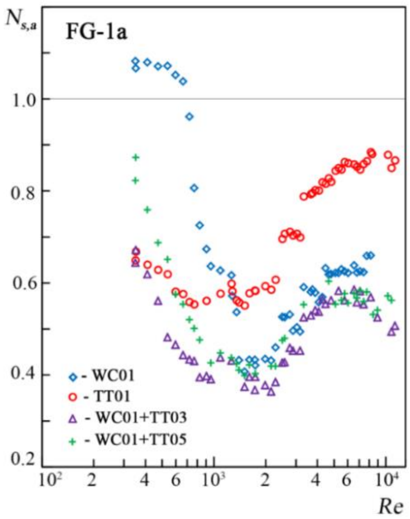

Figure 16. The variations of N_{sa} with Re

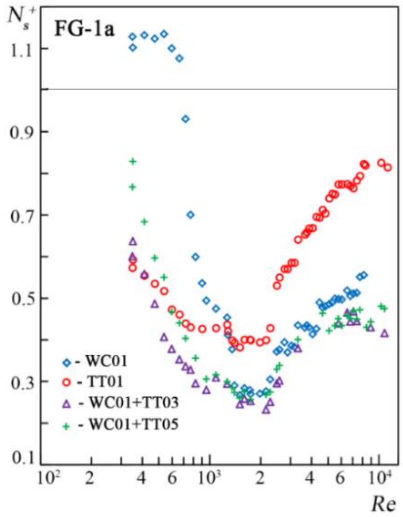

Figure 17. The variations of N_s^+ with Re

Several conclusions can be made from these results: (a) all tube inserts implemented obey the requirements $Q_* > 1$ and $N_{sa} < 1$ for the region of Re studied; (b) the use of compound heat transfer enhancement technique is more beneficial than that of single one; (c) the maximum increase in the heat flow is $Q_* = 1.65$; (d) the compound heat transfer enhancement technique generates less entropy. The results depicted in Figs. 16 and 17 confirm this.

4.2. Case FN-1

Figures 18 and 19 present the results of the decrease in the length of heat exchanger tube, $L_* < 1$, and the decrease in augmentation entropy generation number, $N_{sa} < 1$. As seen in the figures, in the range $\mathrm{Re} < 2\times10^3$ the insert combination WC01+TT03 brings about the greatest reduction of the tube length with maximum $L_* = 0.4$ for $\mathrm{Re} = 2\times10^3$. However, when $\mathrm{Re} > 2\times10^3$, the greatest benefit can be achieved by single twisted tape insert. If the objective is to decrease the entropy generated as much as possible, then the right choice is the insert combination WC01+TT03, Fig. 19.

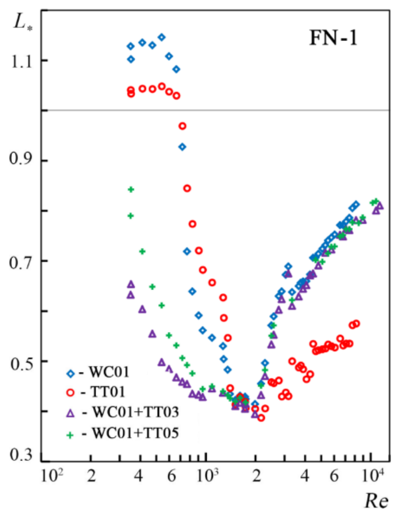

Figure 18. The variation of L_* *with* Re

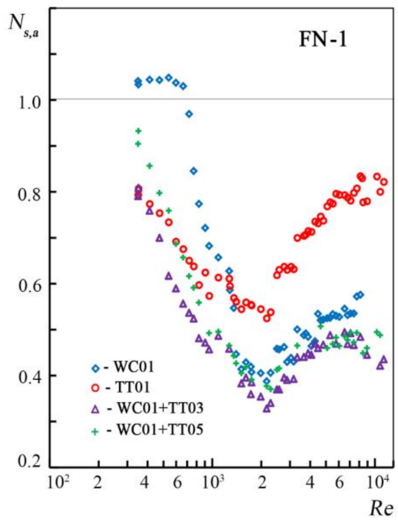

Figure 19. The variations of N_{sa} *with* Re

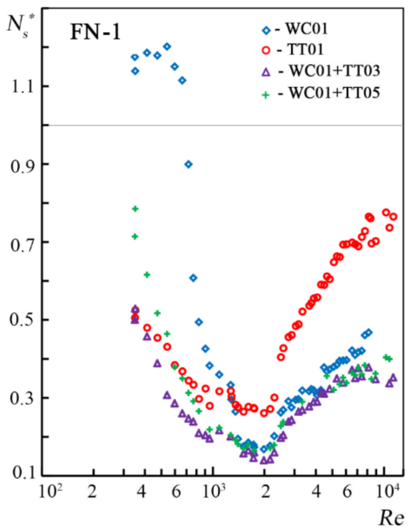

Figure 20. The variations of N_s^* *with* Re

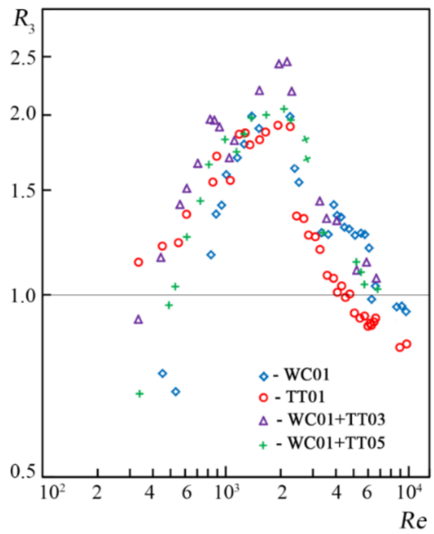

Figure 21. The variations of R_3 *with* Re

In this case, the preferable heat transfer enhancement technique is selected by the criterion

$$N_s^* = L_* N_{sa} < 1. \tag{7}$$

Figure 20 present the results from the use of the criterion N_s^*. Undoubtedly, the choice of the insert combination WC01+TT03 is the most beneficial one.

Many researchers continue to assess the benefits from the use of different heat transfer enhancement techniques by the criterion PEC, Eq. (8),

$$PEC \equiv R_3 = \frac{Nu_a / Nu_s}{\left(f_a / f_s\right)^{1/3}}. \tag{8}$$

The objective of this criterion is to assess the increase in heat duty with a major constraint of a fixed pumping power. Figure 21 shows the benefits evaluated by the use of criterion R_3, Eq. (8). These results can be compared with the results from Fig. 12. There is a big difference. According to Fig. 21, for $\mathrm{Re} > 6 \times 10^3$, all types of inserts used in this study are inefficient and must be discarded.

Conclusion

In this study, the thermal benefit that could be achieved by the use of some tube inserts as a heat transfer enhancement technique has been assessed. The result is that, if a compound heat transfer enhancement by a combination of wire-coil and twisted tape insert will bring about the greatest increase in the heat flow, $Q_* = 1.65$, or the greatest reduction of tube length, $L_* = 0.4$, for $\mathrm{Re} = 2\times 10^3$ with a minimum entropy generation.

Acknowledgments

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Nomenclature

A heat transfer surface area, m^2

A_* ratio of heat transfer surfaces, $A_* = A_a / A_s$

D_* ratio of tube diameters, $D_* = D_a / D_s$

f Fanning friction factor

f_* ratio of Fanning friction factors, $f_* = f_a / f_s$

h heat transfer coefficient, $W\,m^{-2}\,K^{-1}$

L tube length, m

L_* ratio of tube lengths, $L_* = L_a / L_s$

$\dot{m}$ mass flow rate in tube, $kg\,s^{-1}$

N_t number of tubes

N_* ratio of number of tubes, $N_* = N_{t,a} / N_{t,s}$

Nu Nusselt number

Nu_* ratio of Nusselt numbers, $Nu_* = Nu_a / Nu_s$

N_s number of entropy production units

N_{sa} augmentation entropy generation number

NTU number of heat transfer units

P pumping power, W

P_* ratio of pumping powers, $P_* = P_a / P_s$

Pr Prandtl number

$\dot{Q}$ heat transfer rate, W

Q_* ratio of heat transfer rates, $Q_* = \dot{Q}_a / \dot{Q}_s$

Re Reynolds number

$\dot{S}_{gen}$ entropy generation flow, $W\,K^{-1}$

St Stanton number

St_* ratio of Stanton numbers, St_a / St_s

T temperature, K

ΔT temperature difference, K

ΔT_i^* ratio of inlet temperature difference between hot and cold streams, $\Delta T_i^* = \Delta T_{i,a} / \Delta T_{i,s}$

U overall heat transfer coefficient, $W\,m^{-2}\,K^{-1}$

W mass flow rate in heat exchanger, $kg\,s^{-1}$

W_* ratio of mass flow rates, $W_* = W_a / W_s$

Greek symbols

r sum of thermal resistances, $m^2 K\,W^{-1}$

ε heat exchanger thermal effectiveness

ε_* ratio of heat exchanger effectiveness, $\varepsilon_a / \varepsilon_s$

ϕ_s irreversibility distribution ratio

Subscripts

a augmentation

i inlet

m mean

o outlet

s smooth

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