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A CRITICAL REVIEW OF THERMAL AUGMENTATION IN CIRCULAR TUBE WITH TWISTED TAPE AND WIRE COIL TURBULATORS

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Abstract: *In the first part of this paper have been presented the influences of insertion of wire coils in conjunction with twisted tapes on heat transfer in a uniform heat-flux, circular tube using air as the test fluid. The wire coil used as a turbulator is placed inside the test tube while the twisted tape is inserted into the wire coil to create a continuous impinging swirl flow along the tube wall. The effects of insertion of the two turbulators with different coil pitch and twist ratios on heat transfer and friction loss in the tube are examined for Reynolds number ranging from 3000 to 18,000. In this paper we use six different types of tubes with the coil pitch ratio (CR) and the twist ratio (Y) of twisted tape.The results indicate that the presence of wire coils together with twisted tapes leads to increase in heat transfer over the use of wire coil/twisted tape alone. The combined twisted tape and wire coil with smaller twist and coil pitch ratios provides higher heat transfer rate than those with larger twist and coil pitch ratios under the same conditions. Also, performance evaluation criteria to assess the real benefits in using both the wire coil and the twisted tape of the enhanced tube are presented.*

Key words*: Heat transfer enhancement, twisted tape, wire-coil insert, turbulator, performance evaluation criteria.*

INTRODUCTION

In many engineering applications the high-performance thermal systems are needed and thus, various methods to enhance heat transfer in the system have been developed extensively. The conventional heat exchangers can be generally improved by means of various augmentation techniques with emphasis on several types of surface enhancements. Enhanced surfaces can create one or more combinations of the following conditions that are favorable for the increase in heat transfer rate with an undesirable increase in friction: (1) interruption of boundary layer development and rising degree of turbulence, (2) increase in heat transfer area, and (3) generating of swirling and/or secondary flows. To date, several studies have been focused on passive heat-transfer enhancement methods. Reverse/swirl flow devices (rib, groove, wire coil, conical ring, snail entry, twisted tape, winglet, etc.) form an important group of the passive augmentation technique. The reverse flow device or the turbulator is widely employed in heat transfer engineering applications. The turbulators are inserted into the flow to provide an interruption of boundary layer development, to increase the heat transfer surface area and to cause enhancement of heat transfer by increasing turbulence intensity or fast fluid mixing.

Therefore, more compact and economic heat exchanger with lower operation cost can be obtained.

Bergles et al. (1974a) proposed nine kinds of PEC for practical applications, denoted as $R – R_9$. These criteria are the most common PEC, used by many researchers for fast and easy evaluation of the benefit of heat transfer enhancement. Four of them are directed to the use of promoters for improvement of existing heat exchangers (fixed basic geometry constraint). The objectives are increased heat flow, R_1 - R_3, or reduced pumping power R_4, with various process restrictions.

The other four criteria, R_5 - R_8, evaluate the advantages of using promoters in the design of a new heat exchanger (variable geometry with length and number of tubes free) where the objective is to reduce the exchanger size (reduced heat transfer area). The last criterion R_9 is employed when the cost considerations are the deciding factor and is presented for the economic evaluation of enhanced tubes.

However, as Bergles et al. (1974b) pointed out, at least two major assumptions must be removed from the analysis, namely, the assumption of constant driving temperature difference between hot and cold fluids must be removed, and the effect of the thermal resistance external to the enhanced surfaces must be included.

In the case of two- or multifluid heat exchangers, overall performance depends on the performance of the other side surface and its thermal resistance, the thermal resistance across the surface wall, flow arrangement, the possibility of enhancement on the other side, and other considerations not necessarily related to the surface characteristics. In this regard, the performance evaluation by the surface comparison methods may not be optimum when the above parameters are considered. In this case, the performance evaluation of two- or multifluid heat exchangers must be performed by the use of overall performance comparison methods.

Webb (1981) defined PEC as an overall performance comparison method to obtain the performance merits of enhanced heat transfer tubular surfaces primarily in single-phase flow. Webb's PEC are an extension and supplement of the PEC defined by Webb and Eckert (1972), and particularly Bergles et al. (1974a,b). These PEC were developed for the general category of surface roughness and internally finned tubes for three different geometrical constraints: fixed geometry (FG), fixed flow area (FN), and variable geometry (VG). With each of the geometrical constraint groupings, PEC are established for three performance objectives: reduced surface area, increased heat flow (or UA), and reduced pumping power. The PEC, as suggested by Bergles et al. (1974a,b) and Webb and Eckert (1972), with the modification of Webb and Scott (1980) for internally finned tubes, as well as Webb's PEC (Webb, 1981, 1994; Webb and Kim, 2005) characterized most of the PEC for enhanced heat transfer surfaces.

While the origins of their development may have been such, their successful usage is not confined to this segment of enhancement alone. They have been shown to provide optimized designs for plate-and-frame heat exchangers with chevron plates (Muley and Manglik, 2000), tube bundles with twisted tape inserts (Yerra et al., 2006), rectangular twisted duct exchangers (Manglik et al., 2012), oval cross-section twisted tube exchangers (Bishara et al., 2013), and compact plate-fin and tube-fin-type compact heat exchangers (Manglik and Jog, 2016), among others.

The performance evaluation criteria for heat exchangers based on second law analysis (Yilmaz et al., 2001) can be grouped into two classes: (i) evaluation techniques using entropy as an evaluation parameter, and (ii) evaluation techniques using exergy as an evaluation parameter. In many investigations reported in the literature, the entropy has been used as an evaluation parameter in different forms. The most popular is the number of entropy production units, N_s (Bejan, 1978). For heat exchangers using enhanced heat transfer techniques, the augmentation entropy generation number $N_{s,a}$ is the most useful as a criterion. This criterion compares the rate of entropy generated of the heat exchanger before and after the implementation of the augmentation technique (Bejan, 1982). It has been implemented for the first time by Bejan and Pfister (1980) and Ouellette and Bejan (1980). This new criterion has been connected with Webb's PEC by Zimparov and Vulchanov (1994) to merge the 1st and 2nd law analysis. Extended PEC based on the augmentation entropy generation number for ducts with constant wall temperature and constant heat flux were developed by Zimparov (2000, 2001a). These PEC have been used to obtain the real merit from the use of compound heat transfer enhancement techniques (Zimparov, 2001b, 2002; Zimparov et al., 2006, 2012).

The objective of this paper is to present a critical review of the performance criteria used by many researchers in evaluation of their experimental or numerical results concerning the use of different enhanced heat transfer techniques and to reveal how real are the profits they claim.

The wire coils of different spring pitch ratios (CR = H/d = 4, 6 and 8) were inserted into the tube by wall-attached position while the twisted tapes for two twist ratios (Y = P/w = 4 and 6) were fitted tightly into the wire coil placed in the test tube.

EXPOSITION

For rapid evaluation, it is a criterion for turbulent flow widely used by many researchers in the form:

$$i_E = \frac{Nu / Nu_s}{(f / f_s)^{0.291}} \tag{1}$$

where the values of Nusselt numbers and friction factors are defined at one and the same Reynolds number, that of the augmented channel, Re_a. Sano and Usui (1982) suggested an evaluation of the heat transfer promoters by fluid dissipation energy, developing a criterion i_E based on the correlation of the heat transfer coefficient as a function of the energy dissipation per unit mass of fluid. For turbulent flow, this criterion takes the same form as Eq. (1).

Fig.1 shows the variation of the criterion i_E with Reynolds number for all wire-coils and twisted tapes. For all, the data obtained by the measured Nusselt number and friction factor values are compared at the same exchanger flow rate. It can be seen in the figure that the criterion i_E generally are above unity for combined wire coil and twisted tape insert at at all Raynolds, indicating that the use of combined both turbulators is advantageous over the smooth tube. The criterion i_E tends to decrease with the increase of Reynolds number values for all combined wire coil and twisted tape inserts. This can be expressed that all turbulators are not advantageous in terms of energy saving at high Reynolds number values. The criterion i_E of combined the wire coil with *CR* = 4 and the twisted tape with *Y* = 4 is found to be the best, especially for lower Reynolds number. The maximum i_E is about 1.65 at the lowest *Re* for the lowest *Y* and *CR* values used in the present work.

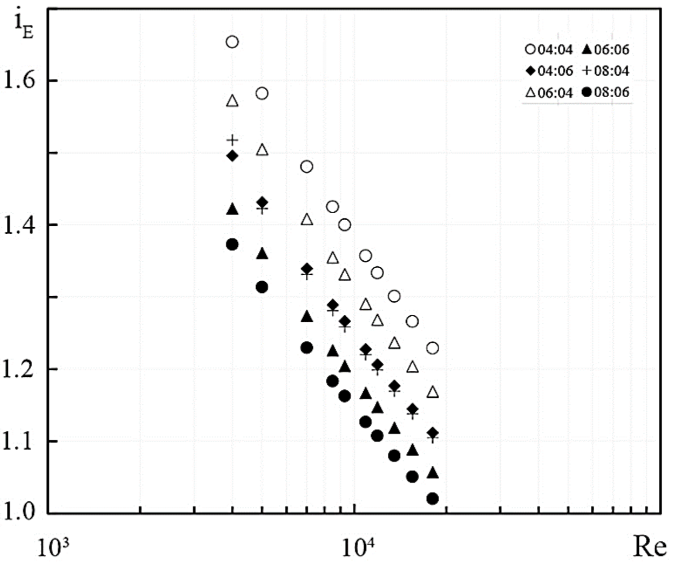

Fig. 1 Variation of i_E with Reynolds number

Performance evaluation equations

The quantitative formulation of the PEC requires algebraic relations which:

(1) quantify the objective function; (2) define the heat transfer and friction characteristics relative to the reference exchanger. For simplicity, the equations are developed for tubes of constant diameter, and heat transfer and friction factors are based on the nominal diameter:

(a) The relative friction power equation for the tube side fluid is

$$P_* = f_* A_* G_*^3, \tag{2}$$

where $P_* = P/P_s$, $f_* = f/f_s$, $A_* = A/A_s$ and $G_* = G/G_s$ are ratios of pumping powers, friction factors, heat transfer areas and mass flow flux, Webb [1981];

b) The UA equation for the reference (smooth tube) and the augmented exchangers is

$$(UA)_* = \frac{1+\beta_s}{St_*^{-1}\left(f_* P_*^{-1} A_*^{-2}\right)^{1/3} + \beta A_*^{-1}}, \tag{3}$$

where $St_* = St/St_s$ is the ratio of Stanton numbers, β and β_s are thermal resistances defined in Webb [1981];

c) The internal surface area ratio (for roughness) A_* and mass flow ratio W_* are given by Eq. (4) and (5), respectively

$$A_* = N_* L_*, \tag{4}$$

$$W_* = N_* G_*, \tag{5}$$

where $N_* = N/N_s$ is the ratio of number of tubes, Webb [1981];

d) The augmented and smooth exchangers may not operate at the same effectiveness (ε). For these cases when the objective is increased heat duty, the $\varepsilon - NTU$ design method gives

$$Q_* = W_* \varepsilon_* \Delta T_i^*, \tag{6}$$

where ΔT_i is the temperature difference between the two inlet streams $\Delta T_i^* = \Delta T_i / \Delta T_{i,s}$ and $\varepsilon_* = \varepsilon / \varepsilon_s$ is the ratio of heat effectiveness. For fixed inlet temperatures, $\Delta T_i^* = 1$, Eq. (6) yields

$$Q_* = W_* \varepsilon_*. \tag{7}$$

Since the operating conditions of the smooth tube exchanger are known, its $\left(NTU\right)_s = U_s A_s / W_s c_p$ is known and ε_s is calculable. Once $(UA)_*$ and W_* for the augmented exchanger are known, the NTU is calculated by

$$NTU = \left(NTU\right)_s (UA)_* W_*^{-1}. \tag{8}$$

Then ε of the augmented exchanger may be calculated and Q_* obtained from Eq. (7).

Performance evaluation criteria

The benefit from the use of this heat transfer enhancement technique has been assessed through three different performance evaluation criteria (PEC), FG-1a, FG-2a and VG-2a, Webb [1981]. The common objective for the three cases was increased heat duty, $Q_* > 1$.

Fixed geometry criteria

These criteria involve a one-for-one replacement of smooth tubes by augmented tubes of equal length. These may be regarded as “retrofit” applications.

Case FG-1a

The FG-1 cases seek increased heat duty or UA for constant exchanger flow rate and velocity. The pumping power of the augmented tube will increase due to the higher friction characteristic of the augmented surface.

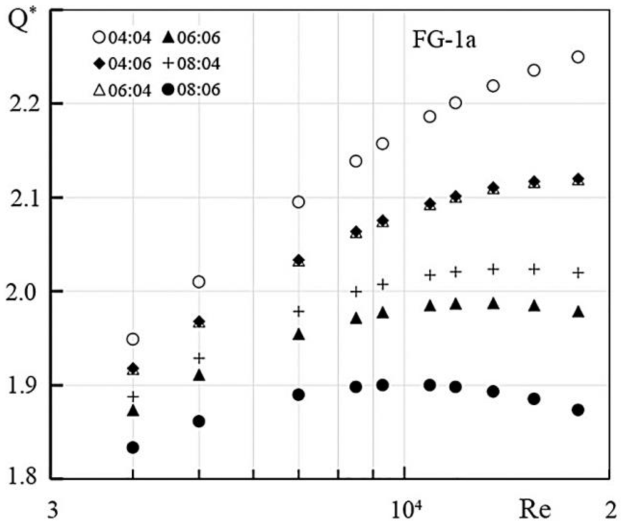

Fig. 2. The variation of Q_* with Reynolds number

The flow rate may be maintained constant with existing equipment if the pressure drop in the system is determined exclusive of the friction loss in the exchanger or if the positive displacement pump is used. If this is not the case, a new, higher head pump will be required to accommodate the higher frictional pressure drop of the promoters.

For the case FG-1a, $W_* = 1$, $N_* = 1$, and $L_* = 1$. Since $G_* = 1$, the f and $St\ (Nu)$ are directly calculable. Eqs. (3) yields ,

$$(UA)_* = \frac{1+\beta_s}{St_*^{-1} + \beta}, \tag{9}$$

$$P_* = f_*. \tag{10}$$

If the entering fluid temperatures are fixed, Eq. (7) gives $Q_* = \varepsilon_*$.

Fig. 2 presents the variation of increased heat duty $Q_* > 1$ with Reynolds number . As seen, one curve 4:4 is depicted in the figure, having maximum at $\mathrm{Re} = 1{,}8.10^4$. In this case, FG-1a, it coincides with the criterion $R_{1,a}$, Bergles et al. [1974a].

$$R_{1,a} = \left(\frac{h}{h_s}\right)_{R_{\exp=0}, D, L, N, T_i, \Delta T} = \frac{\dot{Q}}{\dot{Q}_s}. \tag{11}$$

Several conclusions can be made from these results:

(i) the greatest benefit can be obtained at $\mathrm{Re} = 1{,}8.10^4$ with $CR:Y = 4:4$;

(ii) for tube $CR:Y = 8:6$ at $\mathrm{Re} \approx 4.10^3$ a decrease is observed;

(iii) there is a significant coincidence for tubes $CR:Y = 6:4$ and $CR:Y = 4:6$

CONCLUSION

A critical review has been carried out to investigate air flow friction and heat transfer characteristics in a round tube fitted with both the coiled wire and the twisted tape turbulators for the turbulent regime, $\mathrm{Re} = 3000 - 18000$. The use of the wire coil and the twisted tape inserts causes a high pressure drop increase and also provides considerable heat transfer augmentations, $Nu_a / Nu_0 = 3 - 6$. However, Nusselt number augmentation tends to decrease with the rise of Reynolds number. If the combined wire coil and twisted tape turbulators are compared with a smooth tube at a constant pumping power, a double increase in heat transfer performance is obtained especially at low Reynolds number. Therefore, the combined wire coil and twisted tape should be applied instead of using a single one only to obtain the highest heat transfer and performance of about $350 - 400\%$ leading to more compact heat exchanger. The best operating regime for combined both the tur- bulators is found at lower Reynolds number values for the lowest values of the coil spring pitch and twist ratio.

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