© Journal of the Technical University - Sofia
Plovdiv branch, Bulgaria
"Fundamental Sciences and Applications"
Vol. 26, 2020

A critical report of heat transfer and pressure drop in a spirally grooved tube with twisted tape insert

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Abstract. In this paper are presented performance enhancement parameters of flow of water in a spirally grooved tube with twisted tape insert. It takes into account Reynolds numbers for fully turbulent ranges 3000-7000. The twisted tape inserts are divided into two types - clockwise and anticlockwise having twist ratios $Y = 10.16, 7.95\ and\ 3.4$ Compared to smooth tube, the heat transfer enhancement is further augmented by inserting twisted tapes. In spirally grooved tube with and without twisted tape, heat transfer increases in turbulent range of Reynolds numbers at constant pumping power. Among the three twist ratios (Y = 10.16, 7.95 and 3.4) tested, heat transfer performance of clockwise twisted tape with Y = 10.16 is found to be the highest at Pr =5.4 in turbulent ranges of Reynolds numbers.

1. Introduction

To obtain compact heat exchangers and reasonable energy costs, passive heat transfer enhancement techniques are preferred over active ones. We are going to consider compound heat transfer enhancement with insertion of a twisted tape in an internally grooved tube [1].

The groove geometry is characterised by groove height e, groove base width t_b and tip width t_t, helix angle α and the number of starts N_s.

The axial pitch p_a of the grooves is given by $(\pi D_t / N_s)\ x\ tan\ \alpha$ where D_i is the nominal internal diameter of the tube. Webb et al. [1] reported experimental data for $0.024 \le e/D_i \le 0.041$ $18 \le N_s \le 45$ $25 \le \alpha \le 45$ and $t_t/D_i \le 0.015$. They developed following correlations for single phase flows $(5.15 \le \Pr \le 6.29)$

$$f = 0.108 \times \mathrm{Re}^{-0.283} N_s^{0.221} \alpha^{0.78} \left(\frac{e}{D_i} \right)^{0.785} \tag{1}$$

$$Nu = 0.00933 \times \mathrm{Re}^{0.819} \Pr^{0.33} N_s^{0.285} \alpha^{0.505} \left(\frac{e}{D_i} \right)^{0.323} \tag{2}$$

The correlations are used only for turbulent range of Reynolds numbers. The increase in Nu will exceed that in f in respect of number of starts N_s.Compared to a smooth tube Usui et al. [3],

$$i_E = \frac{Nu / Nu_s}{(f / f_s)^{0.291}} = f(\mathrm{Re}) \tag{3}$$

 Online ISSN 2603-459X

There are many other related contributions of twisted tapescorrugated [4,5] or spirally fluted tubes [6] that are not reviewed here in which both single phase as well as flow-boiling and flow-condensation are considered.

2. Exposition

2.1. Performance evaluation and discussion

Many performance evaluation criteria (PEC) have been developed for evaluating the performance of heat exchangers [9]. They may be categorized as criteria based on the first law of thermodynamics and criteria based on the second law of thermodynamics.

A widely used method to evaluate the benefit of an enhanced heat transfer surface is to compare the performance of the enhanced surface with that of the corresponding plain (smooth) surface. The benefit depends on the goal to be achieved and the constraints imposed of the comparison. In general, the performance evaluation includes three considerations: the performance objective, operation conditions and constraints. The potential objectives could be: increased heat transfer rate, reduced pumping power, or reduced size of the heat exchanger. Possible constraints are: fixed mass flow rate, heat flow, pumping power, size of the heat exchanger, etc. The major operational variables include the heat transfer rate, fluid pumping power or pressure drop, flow rate, and fluid velocity. A PEC is established by selecting one of the operational variables for the performance objective subject to design constraints on the remaining variables.

If the objective is more heat flow to be transferred, this criterion is known as

$$R_3 = \left(\frac{h}{h_s}\right)_{D,L,N,P,T_{in},\Delta T} = \frac{\dot{Q}}{\dot{Q}_s} \quad . \tag{4}$$

In this case, the process constraints are: fixed pumping power P, inlet fluid temperature T_{in}, and driving tempera-ture difference ΔT. The constraint of equal pumping power requires different Reynolds numbers for the working fluid in reference and augmented channels, $\mathrm{Re} < \mathrm{Re}_s$, and

$$f\left(\mathrm{Re}\right)\mathrm{Re}^3 = f_s(\mathrm{Re}_s)\mathrm{Re}_s^3 \ . \tag{5}$$

Consequently, the corresponding heat transfer coefficients in Eq. (4) should be calculated at these Reynolds numbers and Eq. (4) yields

$$R_3 = \frac{Nu(\mathrm{Re})}{Nu_s(\mathrm{Re}_s)} = \frac{\dot{Q}}{\dot{Q}_s} . \tag{6}$$

It must be emphasized that the criterion R_3 has been developed with the assumptions of negligible external thermal resistance, $R_{ext} = 0$, and equal temperature difference ΔT in the comparative heat exchangers. In general, however, the ΔT will decrease due to the increased rate of heat transfer.

Sano and Usui [7] suggested evaluation of the heat transfer promoters by fluid dissipation energy, developing a criterion based on the correlation of the heat transfer coefficient as a function of the energy dissipation per unit mass of fluid (ε). For turbulent flow, this criterion takes the form

$$i_E = \frac{Nu / Nu_s}{(f / f_s)^{0.291}} = f(\mathrm{Re}) . \tag{7}$$

It is important to note that the criterion i_E is identical to R_3 criterion of Bergles et al. [8], but is more convenient to use since the Nusselt numbers Nu, Nu_s, and friction factors f, f_s are defined at one and the same Reynolds number Re. It should also be emphasized that the two heat exchangers must work in turbulent regime, $\mathrm{Re} > 3\times10^3$, and $\mathrm{Re}_s > \mathrm{Re}$.

3. Discussion

Fig. 1 shows friction factor augmentation $f_* = f/f_s$ produced by grooved tube. The tendency of friction factor behaviour commented in the foregoing discussion is very clean expressed. As seen, for Reynolds number in the range 3 000 – 7 100 the friction factor ratio f_* decreases from 0.374 to 0.362 and reaching its minimum.

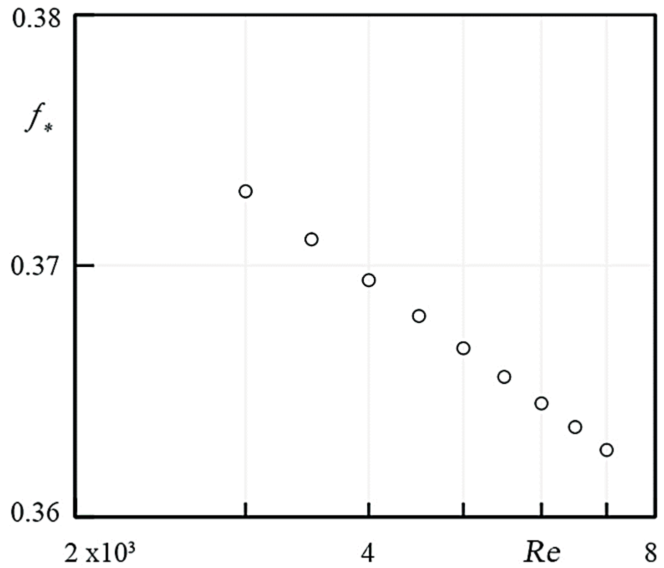

Figure 1. The variation of f_* with Reynolds number. Grooved tube without twisted tape.

Nusselt number augmentation is defined by the ratio $Nu_* = Nu/Nu_s$ at the same Reynolds and Prandtl numbers. Fig. 2 shows the ratio Nu_* for the grooved tube without wire-coil insert studied. In the region (3 000 < Re < 7 000) Nu_* increases from 0.87 to almost 1.22

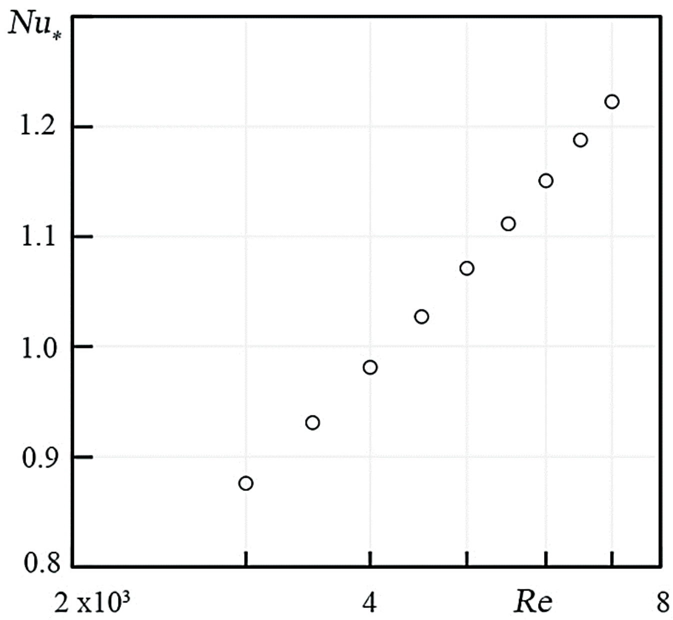

Figure 2. The variation of Nu* with Reynolds number. Grooved tube without twisted tape.

Fig. 3 shows the variation of the criterion i_E with the Reynolds number for the experimental results for a grooved tube with twisted tape. The grooved tube performed with $i_E > 1$ which shows a significant enhancement for heat transfer. The comparison of the performance enhancement parameter i_E between grooved tube with twisted tapes and the grooved tube without twisted tapes shows that the first one is more sophisticated than second.

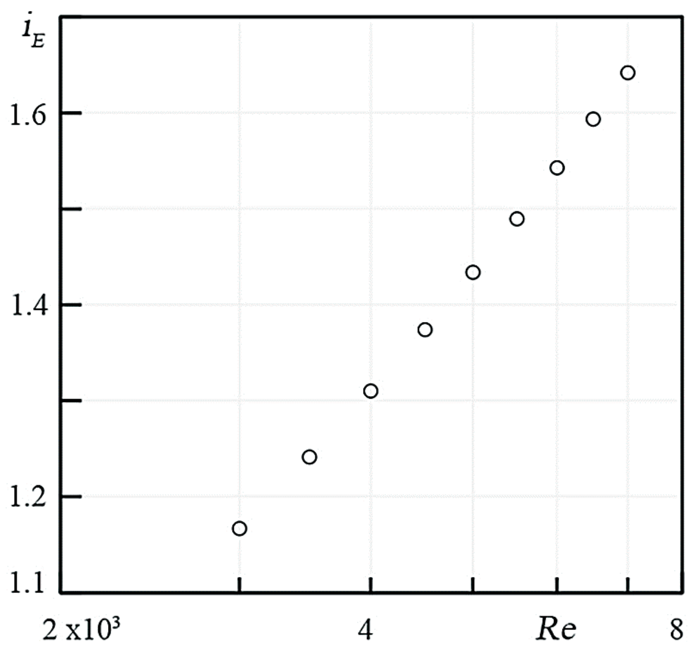

Figure 3. The variation of i_E with Reynolds number. Grooved tube without twisted tape.

Fig.4 presents the variation of i_E with the Reynolds number in the augmented channel for the experimental results of. As seen the grooved tube with twisted ratios Y=10.16, 7.95 and 3.4 performed with $i_E > 1$ and the variation with the Reynolds number is large enough. The benefit of some of them is very similar as tape with Y=7.95 anticlock, Y=3.4 clockwise and anticlock. The greatest profit can be obtained by using the tapes with Y=10.16 and 7.95 clockwise.

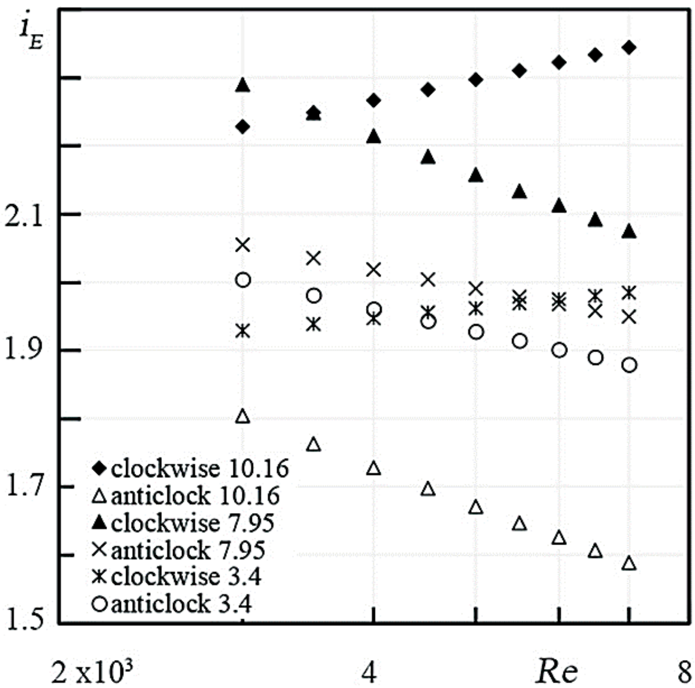

Figure 4. The variation of i_E with Reynolds number. Grooved tube with twisted tape.

4. Conclusion

Pressure drop and constant wall heat flux heat transfer measurements have yielded highly non-linear behaviour of f and Nu with Reynolds number .

Similar comparison for spirally grooved tube with twisted tape shows maximum enhancement of 140% in the turbulent range. However, deterioration in heat transfer is observed at Y=10.16 (clockwise) for 6000 < Re_{sm} <7 000

It was found that among three twist ratios (Y = 10.16, 7.95 and 3.4) tested, Y = 10.16 is found to be the highest at Pr =5.4 in turbulent ranges . There was extremely complex interactions between momentum and heat transfer in the vicinity of the grooved wall resulting in highly non-monotonic and non-linear behaviour at Pr =5.4. Further experimentation and analysis are required to obtain optimum groove geometry and twist ratio Y for a given fluid.

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