PERFORMANCE EVALUATION OF TUBE-IN-TUBE HEAT EXCHANGERS WITH HEAT TRANSFER ENHANCEMENT IN THE ANNULUS

by

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Different techniques as angled spiraling tape inserts, a round tube inside a twisted square tube and spiraled tube inside the annulus have been used to enhance heat transfer in the annulus of tube-in-tube heat exchangers. The heat transfer enhancement in the shell can be supplemented by heat transfer augmentation in tubes using twisted tape inserts or micro-finned tubes. The effect of the thermal resistance of the condensing refrigerant could also be taken into consideration. To assess the benefit of using these techniques extended performance evaluation criteria have been implemented at different constraints. The decrease of the entropy generation can be combined with the relative increase of the heat transfer rate or the relative reduction of the heat transfer area to find out the geometrical parameters of the tubes for optimal thermodynamics performance. The results show that in most of the cases considered, the angled spiraling tube insert technique is the most efficient.

Key words: *enhanced heat transfer in annulus, spiraling tape insert, spiraled tube, performance evaluation criteria*

Introduction

The performance of conventional heat exchangers can be substantially improved by many augmentation techniques applied to design systems. Heat transfer enhancement devices are commonly employed to improve the performance of an existing heat exchanger or to reduce the size and cost of a proposed heat exchanger. An alternative goal is to use such techniques to increase the system thermodynamic efficiency, which allows to reduce the operating cost. A classification of enhancement techniques can be found in [1-3].

The surface methods include any technique which directly involves the heat exchanger surface. They are used on the side of the surface that comes into contact with a fluid of low heat transfer coefficient in order to reduce the thickness of the boundary layer and to introduce better fluid mixing. Some of the existing methods for enhancing heat transfer in a single-phase, fully developed turbulent flow are one of the two types: (a) methods in which the surface is roughened, *e. g.* with repeated or helical ribbing, by sanding, or with fins, and (b) methods in which a heat transfer promoter, *e. g.*, a twisted tape, disk or streamlined shape is inserted into the channel.

In many cases heat transfer enhancement in tubes can be supplemented by heat transfer enhancement on the outside wall of tubes, as for tube-in-tube heat exchangers. An application is in vapor compression hot-water heat pumps. The condensing refrigerant may typically flow in the inner tube and the water to be heated in a counter flow direction in the annulus. In this case, heat transfer enhancement on the outer wall is also important. Heat transfer enhancement on the inner or outer tube decreases the temperature difference between the condensing refrigerant and water to be heated, which is an advantage as higher temperature can be reached.

Recently, several studies have been done to improve the heat transfer coefficient in annuli [4-6]. The purpose of these studies was to investigate the potentials of some very simple and inexpensive methods of heat transfer augmentation that could be used by small manufacturing companies. Van der Vyver and Meyer [4] used a round tube inside a twisted square tube to enhance the rotation component in the annulus which increased the heat transfer coefficient by up to 50%. At the same time the friction factor increased by up to 9%.

Another method used by Herman and Meyer [5] was to use a spiraling tube inside the annulus of a tube-in-tube heat exchanger. Three tubes were thus used. Except for the inner and outer tube, the third tube was spiraled in the annulus of the other two and also formed a flow passage. The aim was not only to increase the heat transfer in the annulus by swirl flow but also to increase the cross flow area, since some of the flow will not only be through the annulus but also through the spiraled tube. The advantage of this method when compared to spiraled thin wires is that no flow can occur through a wire since it blocks the flow.

The third method that makes use of this principle was patented by Meyer and Coetzee [7]. An angled spiraling tape is used in the annulus to induce swirl, fig. 1. Three tube-in-tube heat exchangers were tested with angled spiraling tape in the annulus with different pitches. It was determined that the heat exchanger with the smallest pitch of the angled spiraling tape and with flow against the curvature of the tape resulted in the highest increase in the Nusselt number of 206%. As penalty this heat exchanger also had the highest increase of the pressure drop of 203%.

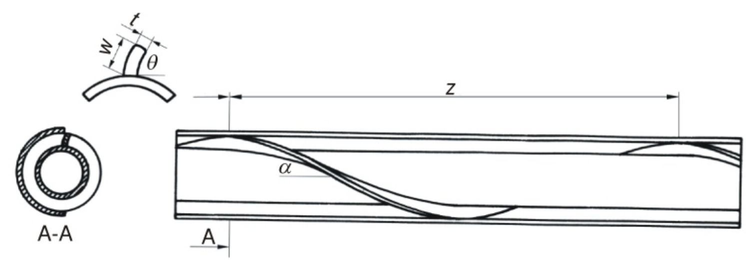

Figure 1. Schematic representation of angled spiraling tape heat exchanger [6]

Being familiar with these investigations [4-6] one faces a question, "Which surface is the best?" The designer may feel frustrated, since the performance characteristic of each of the surfaces tested has not been evaluated on any common basis.

In this paper we attempt to identify the preferred enhancement geometry on the basis of its thermodynamic performance. The evaluation of the performance will be made on the basis of the first and the second law analysis, taking into account some design or operational constraints.

Performance evaluation criteria

The extended performance evaluation criteria (PEC) equations have been used to assess the thermodynamic efficiency of the tubes investigated. The equations are developed for tubes of different diameters and heat transfer and friction factors based on the presentation format of performance data for enhanced tubes [8]. The relative equations for single-phase flow inside enhanced tubes or channels are [9]:

$$A_* = N_* L_* D_* \tag{1}$$

$$P_* \quad W_* \Delta p_* \quad \frac{f_R}{f_S D_* L_* N_* u_{m*}^3} \quad \frac{W_*^3 L_*}{N_*^2 D_*^5} \frac{f_R}{f_S} \tag{2}$$

$$Q_* \quad W_* \varepsilon_* \Delta T_i^* \tag{3}$$

$$W_* \quad u_{m*} D_*^2 N_* \quad \frac{\mathrm{Re}_R}{\mathrm{Re}_S} D_* N_* \tag{4}$$

$$\Delta p_* \quad \frac{f_R}{f_S} \frac{L_*}{D_*} u_{m*}^2 \quad \frac{f_R}{f_S} \frac{L_*}{D_*^3} \frac{\mathrm{Re}_R^2}{\mathrm{Re}_S^2} \tag{5}$$

$$\frac{(UA)_{iR}}{(UA)_{iS}} \quad \frac{1 \quad R_S}{\dfrac{\mathrm{St}_S}{\mathrm{St}_R} \sqrt[3]{\dfrac{f_R}{f_S} \dfrac{1}{P_*} \dfrac{1}{A_*^2}} \quad R_R \dfrac{1}{A_*}} \tag{6}$$

where R is a sum of resistances (defined in [9]).

The effect of the thermal resistance external to the surface under consideration could also be taken into account by including the external heat transfer coefficient h_{oS}. The analysis includes the possibility that the enhanced heat exchanger may have an enhanced outer tube surface $E_o = h_{oR}/h_{oS}$. The fouling resistances on both sides of the tube wall could also be put into consideration. In this study all the tubes have been evaluated at boundary condition constant wall temperature. It should be pointed out that before this the correlated friction factors and heat transfer coefficients have been recalculated with respect to the hydraulic diameter of the smooth tube-in-tube heat exchanger as a common basis for comparison.

The evaluation and comparison of the heat transfer augmentation techniques [4-6] have been made on the basis of both first and second law analysis. Thus it is possible to determine the thermodynamic optimum in a heat exchanger by minimizing the augmentation entropy generation number compared with the relative increase of heat transfer rate $Q_* > 1$, or relative reduction of heat transfer area $A_* < 1$ or pumping power $P_* < 1$. Consequently, a ratio N_S/Q_* and a group $N_S A_* = \mathrm{f}(\mathrm{Re_R})$ might be defined to connect the two objectives pursued by the first and second law analysis and as a basis for thermodynamic optimization. The heat transfer efficiency of the tubes investigated has been evaluated for the following cases.

Fixed geometry criteria (FG)

These criteria involve a one-for-one replacement of the inner smooth tubes of shell-and-tube heat exchanger by augmented ones of the same basic geometry, *e. g.*, shell diameter, tube length, and number of tubes. The FG-1 cases [9] seek increased heat duty or overall conductance *UA* for constant exchanger flow rate. The pumping power of the enhanced shell-and-tube heat exchanger will increase due to the increased fluid friction characteristics of the augmented surface. For these cases the constraints , ΔT_i^* 1, $W_* = 1$, $N_* = 1$, and $L_* = 1$ require $P_* > 1$. When the objective is increased heat duty $Q_* > 1$, this corresponds to the case FG-1a, [9]. The augmentation entropy generation number N_S, [10] is:

$$N_S \quad \frac{1}{1 \quad \phi_\mathrm{o}} \quad Q_* \exp B \;\; 1 \quad \frac{\mathrm{St_R}}{\mathrm{St_S}} D_*^{\;0.8} \quad \frac{T_\mathrm{iS}}{T_\mathrm{oS}} \quad Q_* \;\; 1 \quad \frac{T_\mathrm{iS}}{T_\mathrm{oS}} \quad ^{1} \quad \phi_\mathrm{o} \frac{\dfrac{f_\mathrm{R}}{f_\mathrm{S}}}{D_*^{4.75}} \quad \mathrm{f}(\mathrm{Re_R}) \tag{7}$$

The ratio N_S/Q_* *vs*. Reynolds number, for the channels investigated in [4-6] is shown in figs. 2-4. As can be seen, in the range $\mathrm{Re} < 1.4 \cdot 10^4$, the best characteristics have the channels with tubes 1, 2 [6] ($y = 0.731$, the smallest pitch of the angled spiraling tape), whereas for $\mathrm{Re} > 2.5 \cdot 10^4$, the tube N5 [6] have a tendency to be the best. Furthermore, all the tubes in [4, 5] and tubes 1, 2 [6] continuously have a degradation in their performance, whereas for $\mathrm{Re} > 1.3 \cdot 10^4$, the tubes 3-6 [6] improve their characteristics diminishing the entropy generation.

The FG-2 [9] criteria have the same objectives as FG-1, but require that the augmented tube unit should operate at the same pumping power as the reference smooth tube unit. The pumping power is maintained constant by reducing the shell-side velocity and

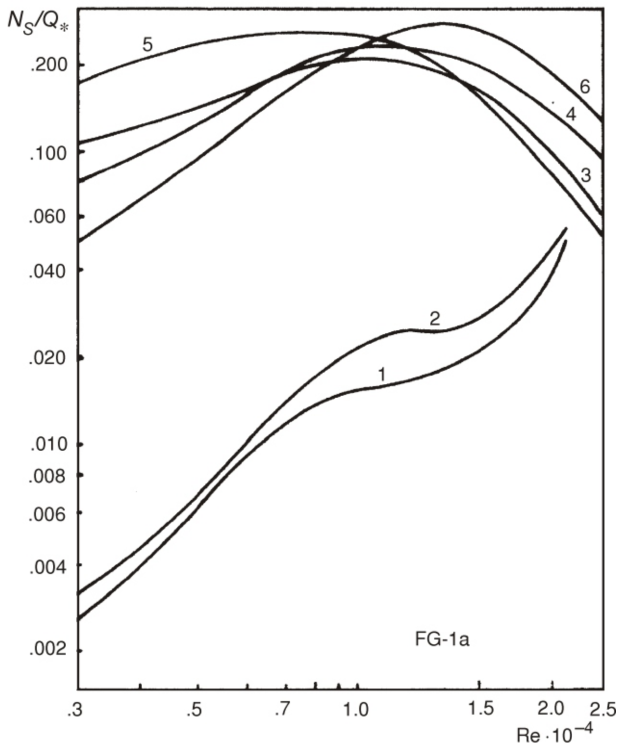

Figure 2. The performance ratio N_S/Q_* *vs.* Reynolds number, [6]
1 – y = 0.731 – against; 2 – y = 0.731 – along; 3 – y = 1.799 – against; 4 – y = 1.799 – along; 5 – y = 2.878 – against; 6 – y = 2.878 – along

thus the exchanger flow rate. The constraints are: $N_* = 1$, $L_* = 1$, and $P_* = 1$ requiring $W_* < 1$, and $\mathrm{Re_R} < \mathrm{Re_S}$. In the case FG-2a the goal is increased heat transfer rate $Q_* > 1$. The augmentation entropy generation number N_S [10] is:

$$N_S \quad \frac{1}{1 \quad \phi_o} \quad Q_* \exp \; B \;\; 1 \quad \frac{\mathrm{St_R}}{\mathrm{St_S}} D_*^{\;1.145} \quad \frac{f_\mathrm{R}}{f_\mathrm{S}}^{\;0.073}$$

$$\frac{T_\mathrm{iS}}{T_\mathrm{oS}} \quad Q_* \quad \frac{f_\mathrm{R}}{f_\mathrm{S}}^{\;0.364} \quad D_*^{\;1.727} \;\; 1 \quad \frac{T_\mathrm{iS}}{T_\mathrm{oS}}^{\;1} \quad \phi_\mathrm{o} \quad \mathrm{f}(\mathrm{Re_R}) \tag{8}$$

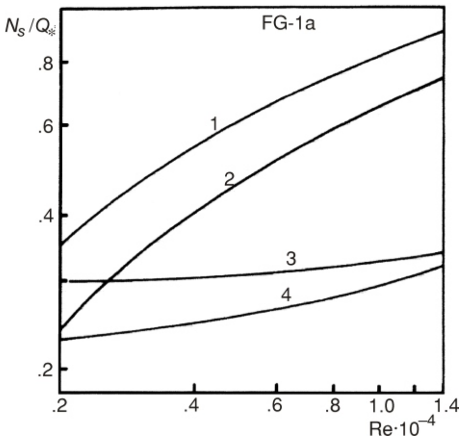

Figure 3. The performance ratio N_s/Q_* *vs.* Reynolds number, [4]
1 – 45^o; 2 – 60^o; 3 – 90^o; 4 – 105^o twist

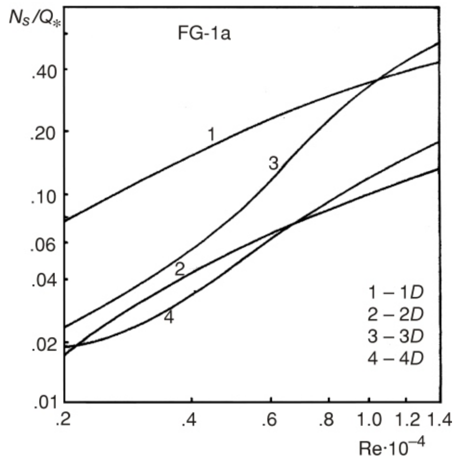

Figure 4. The performance ratio N_s/Q_* *vs.* Reynolds number, [5]

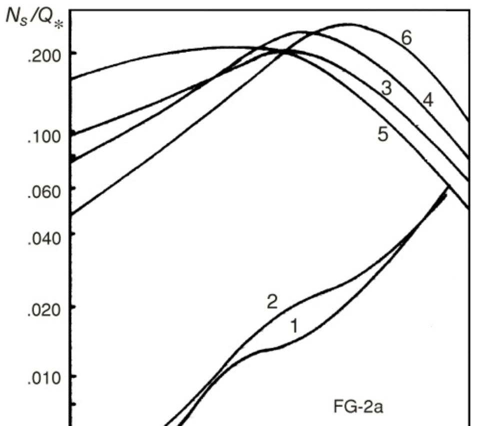

Figure 5. The performance ratio N_s/Q_* *vs.* Reynolds number, [6]
1 – y = 0.731 – against, 2 – y = 0.731 – along;
3 – y = 1.799 – against; 4 – y = 1.799 – along;
5 – y = 2.878 – against; 6 – y = 2.878 – along

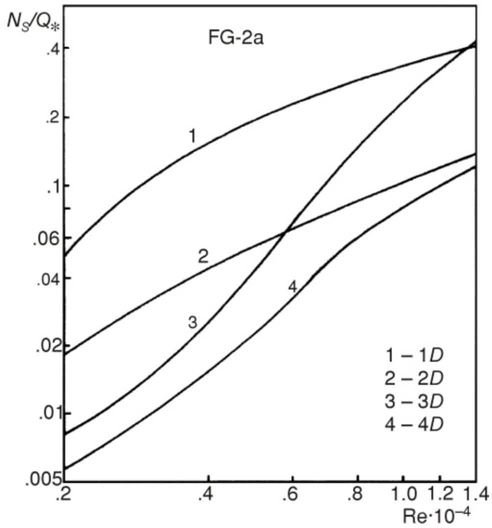

Figure 6. The performance ratio N_s/Q_* *vs.* Reynolds number, [5]

The results for the case FG-2a are presented in figs. 5-7 and they are nearly the same as those for the case FG-1a.

Variable geometry criteria (VG)

In most cases a heat exchanger is designed for a required thermal duty with a specified flow rate. Since the shell-side velocity must be reduced to accommodate the higher friction characteristics of the augmented surface, it is necessary to increase the flow area to maintain $W_* = 1$ and to permit the exchanger flow frontal area to vary in order to meet the pumping power constraint: $N_* > 1$, $L_* < 1$, $\mathrm{Re_R} < \mathrm{Re_S}$.

In the case VG-1 [9] the objective is to reduce surface area $A_* < 1$ with $W_* = 1$ for $Q_* = P_* = 1$.

The entropy generation number is calculated from [10]:

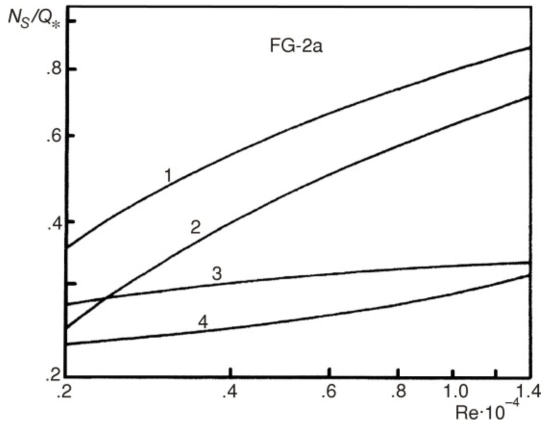

Figure 7. The performance ratio N_S/Q_* *vs.* Reynolds number, [4]
1 – 45°; 2 – 60°; 3 – 90°; 4 – 105° twist

$$N_S \quad \frac{1}{1 \quad \phi_\mathrm{o}} \quad \frac{f_\mathrm{R}}{f_\mathrm{S}} A_* \; {}^{0.364}$$

$$D_*^{2.091} \exp B \;\; 1 \quad \frac{\mathrm{St_R}}{\mathrm{St_S}} \; \frac{f_\mathrm{R}}{f_\mathrm{S}} \; {}^{0.291} \quad A_*^{0.709} D_* \, {}^{0.127} \quad \frac{\phi_\mathrm{o}}{N_*} \quad \mathrm{f(Re_R)} \tag{9}$$

The group $A_* N_S$ *vs.* Reynolds number, for the channels investigated in [4-6] is shown in figs. 8-10. For this case the best performance has a tube 4 [5], whereas the tubes 1, 2 [6] have worse performances even than tube 2 [5].

The cases VG-2 [9] aim at increased thermal performance ($U_\mathrm{R}A_\mathrm{R}/U_\mathrm{S}A_\mathrm{S}$ or $Q_* > 1$) for $A_* = 1$ and $P_* = 1$. They are similar to the cases FG-2. When the objective is $Q_* > 1$, case VG-2a [9], an additional constraint is ΔT_i^*. The last case considered is VG-2a where the objective is increased heat rate $Q_* > 1$ for $W_* = 1$ and $A_* = P_* = 1$. The values of N_S are calculated following [10]:

$$N_S \quad \frac{1}{1 \quad \phi_\mathrm{o}} \quad Q_* \quad \frac{f_\mathrm{R}}{f_\mathrm{S}} \; {}^{0.364}$$

$$D_*^{2.091} \exp B \;\; 1 \quad \frac{\mathrm{St_R}}{\mathrm{St_S}} \; \frac{f_\mathrm{R}}{f_\mathrm{S}} \; {}^{0.291} \quad D_* \, {}^{0.127} \quad \frac{T_\mathrm{iS}}{T_\mathrm{oS}} \quad Q_* \quad 1 \quad \frac{T_\mathrm{iS}}{T_\mathrm{oS}} \; {}^{1} \quad \frac{\phi_\mathrm{o}}{N_*} \quad \mathrm{f(Re_R)} \tag{10}$$

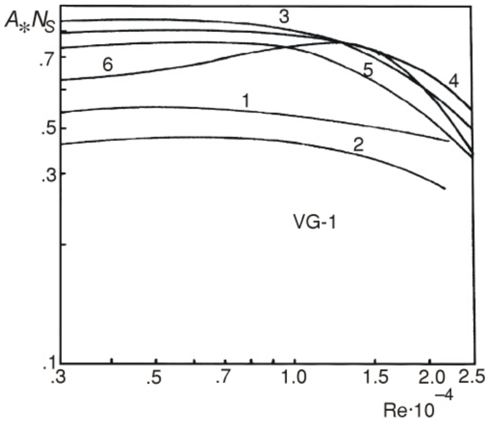

Figure 8. The performance group $A_* N_S$ *vs.* Reynolds number, [6]
1 – y = 0.731 – against; 2 – y = 0.731 – along; 3 – y = 1.799 – against; 4 – y = 1.799 – along; 5 – y = 2.878 – against; 6 – y = 2.878 – along

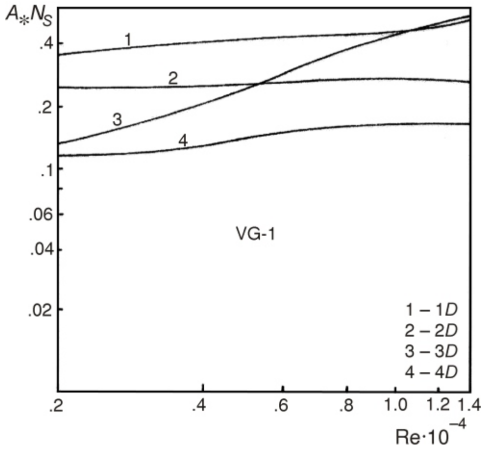

Figure 9. The performance group $A_* N_S$ *vs.* Reynolds number, [5]

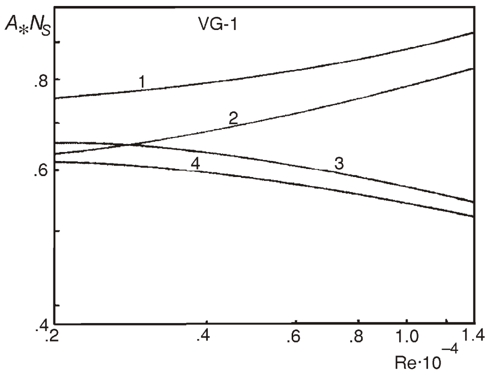

Figure 10. The performance group $A_* N_S$ *vs.* Reynolds number, [4]
1 – 45 °; 2 – 60°; 3 – 90°; 4 – 105° twist

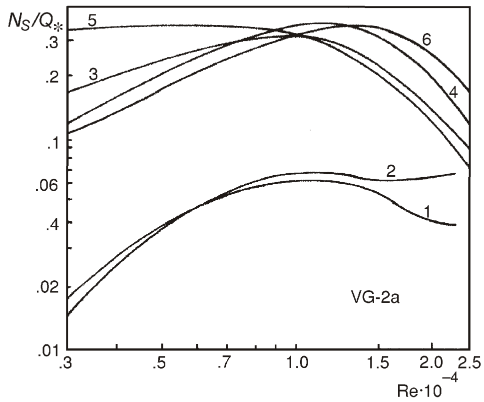

Figure 11. The performance ratio N_S/Q_* *vs.* Reynolds number, [6]
1 – y = 0.731 – against; 2 – y = 0.731 – along; 3 – y = 1.799 – against; 4 – y = 1.799 – along; 5 – y = 2.878 – against; 6 – y = 2.878 – along

For this case, in the range Re < 10^4, the best performance has tube 4 [5], but for Re > 10^4 the best one is tube 1 [6]. The results presented in figs. 2-13 have been obtained assuming that the internal heat transfer coefficient of the inner tube is relatively large in comparison with the heat transfer coefficient in the annulus. Consequently, in this case of water through the annulus heating with refrigerants flowing in the inside of a tube, the

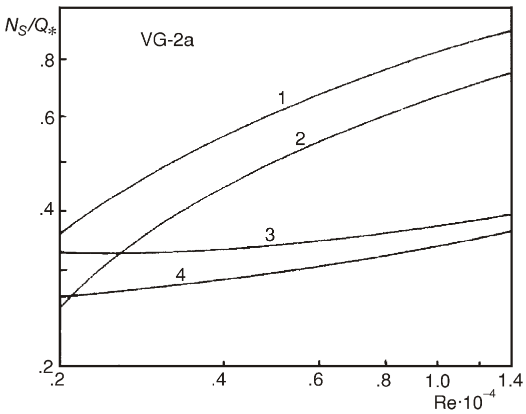

Figure 12. The performance ratio N_S/Q_* vs. Reynolds number, [4]

1 – 45 °; 2 – 60°; 3 – 90°; 4 – 105° twist

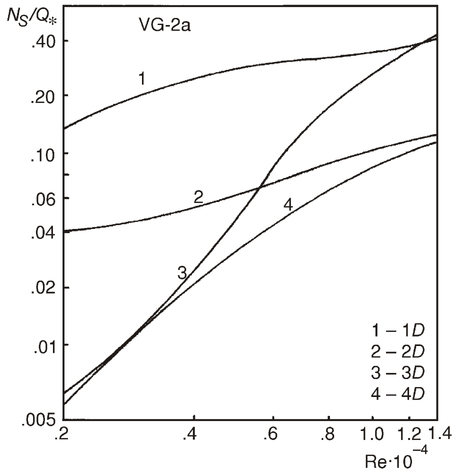

Figure 13. The performance ratio N_S/Q_* vs. Reynolds number, [5]

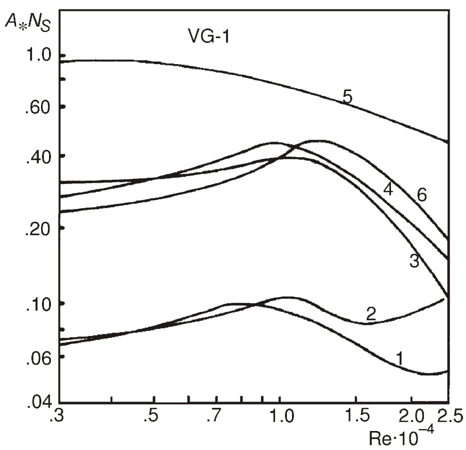

Figure 14. The performance group A_*N_S vs. Reynolds number, [6]

1 – y = 0.731 –, 2 – y = 0.731 – along;
3 – y = 1.799 – against; 4 – y = 1.799 – along;
5 – y = 2.878 – against; 6 – y = 2.878 – along

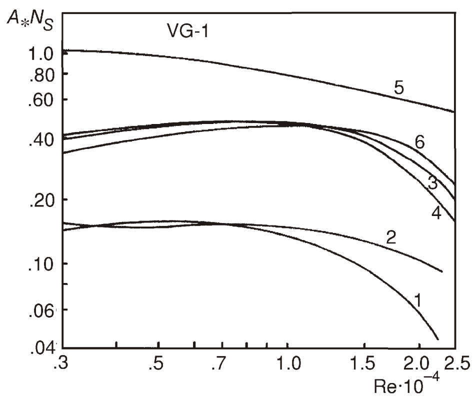

Figure 15. The performance group A_*N_S vs. Reynolds number, [6]

1 – y = 0.731 –, 2 – y = 0.731 – along;
3 – y = 1.799 – against; 4 – y = 1.799 – along;
5 – y = 2.878 – against; 6 – y = 2.878 – along

heat transfer coefficient of condensing refrigerant is assumed relatively large $h_i = \infty$ and the thermal resistance on the annulus side is the highest. But, in many cases, for intube boiling or condensation of refrigerants, the inside heat transfer coefficient is relatively small, and has to be augmented.

Figures 8, 14, and 15 show influence of the inside heat transfer coefficient on the performance of the tubes [6] for the case VG-1. The values of the inside heat transfer coefficient are as follows: $h_i = \propto$ (fig. 8); h_i = 2127 W/m^{2}K (plain tube – fig. 14), and h_i = 5290 W/m^{2}K (Microfin #3 – fig. 15), Thome [11]. When the inside wall of the inner tube is smooth or micro-finned, the best performance has tube 1, but when the inside heat transfer coefficient becomes large and the internal thermal resistance is negligible, the best performance has tube 2.

Conclusions

Extended performance evaluation criteria equations have been used to assess the thermodynamic efficiency of some techniques to enhance heat transfer in the annulus of tube-in-tube heat exchangers, such as: angled spiraling tape inserts, a round tube inside a twisted square tube and spiraled tube inside the annulus. The heat transfer enhancement in the shell can be supplemented by heat transfer augmentation in tubes using twisted tape inserts or micro-finned tubes. The effect of the thermal resistance of the condensing refrigerant could also be taken into consideration. The evaluation of the performance of each technique has been made on the basis of the first and second law analyses, taking into account some design or operational constraints. The results show that in most of the cases considered, the angled spiraling tube insert technique is the most efficient.

Nomenclature

A – heat transfer surface area, [m^2]
A_* – non-dimensional heat transfer surface (=A_R/A_S), [–]
B – constant, [–]
D – tube diameter, [m]
D_* – non-dimensional tube diameter (=D_R/D_S), [–]
f – Fanning friction factor, [–]
h – heat transfer coefficient, [W/m^2 K]
L – tube length, [m]
L_* – non-dimensional tube length, (=L_R/L_S), [–]
N_t – number of tubes, [–]
N_S – augmentation entropy generation number, [–]
N_* – ratio of number of tubes (=N_{tR}/N_{tS})
P – pumping power, [W]
P_* – non-dimensional pumping power (=P_R/P_S)
Δp – pressure drop, [Pa]
$\dot{Q}$ – heat transfer rate, [W]
$\dot{Q}_*$ – non-dimensional heat transfer rate ($\dot{Q}_R$ / $\dot{Q}_S$)
Re – Reynolds number, [–]
St – Stanton number, [–]
T – temperature, [K]
T – temperature difference, [K]

ΔT_i^* – non-dimensional inlet temperature difference between hot and cold streams ($=\Delta T_{iR}/\Delta T_{iS}$)
U – overall heat transfer coefficient, [W/m²K]
u – flow velocity, [m/s]
u_m^* – non-dimensional flow velocity ($=u_{mR}/u_{mS}$)
W – mass flow rate in heat exchanger, [kg/s]
W_* – non-dimensional mass flow rate ($=W_R/W_S$)
y – twist ratio ($=z/2D_i$), [–]
z – pitch of the angled spiraling tape, [m]

Greek Letters

α – helix angle of tape (fig. 1), [°]
ε_* – ratio of heat exchanger effectiveness ($=\varepsilon_R/\varepsilon_S$), [–]
θ – tape angle (fig. 1), [°]
ϕ_o – irreversibility distribution ratio, [–]

Subscripts

i – inside, or value at $x = 0$
m – mean value
R – rough tube
S – smooth tube
o – outside, or value $x = L$

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