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HEAT TRANSFER ENHANCEMENT BY A COMBINATION OF SPIRALLY CORRUGATED TUBES WITH A TWISTED TAPES PART 2 – THERMODYNAMIC EFFICIENCY

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ABSTRACT

Extended performance evaluation criteria (PEC) equations for enhanced heat transfer surfaces have been used to assess the multiplicative effect of combination of spirally corrugated tube with twisted tape inserts. Thermodynamic optimum can be defined by minimizing the entropy generation augmentation number compared with the relative increase of heat transfer rate or relative reduction of heat transfer area.

NOMENCLATURE

D	tube diameter, m
A	heat transfer surface area, m^2
h	heat transfer coefficient, W/(m^{2}K)
U	overall heat transfer coefficient, W/(m^{2}K)
W	mass flow rate in heat exchanger, kg/s
P	pumping power, W
$\dot{Q}$	heat transfer rate, W
$\dot{S}_g$	entropy generation rate, W/K
N	number of tubes
L	tube length, m
T	temperature, K

Dimensionless groups

f	Fanning friction factor
Nu	Nusselt number
Re	Reynolds number
St	Stanton number

$A_* = A_R / A_S$ - dimensionless heat transfer surface

$Q_* = \dot{Q}_R / \dot{Q}_S$ - dimensionless heat transfer rate

$P_* = P_R / P_S$ - dimensionless pumping power

$f_* = f_R / f_S$ - ratio of Fanning friction factor

$w_* = w_R / w_S$ – mass flow rate per unit area

$(D_i)_* = D_{i,R} / D_{i,S}$ - dimensionless inside tube diameter

$W_* = W_R / W_S$ - dimensionless mass flow rate

$N_* = N_{t,R} / N_{t,S}$ - ratio of number of tubes

$\Delta T_i^* = \Delta T_{i,R} / \Delta T_{i,S}$ - dimensionless inlet temperature difference between hot and cold streams

$\varepsilon_* = \varepsilon_R / \varepsilon_S$ - ratio of heat exchanger effectiveness

$N_S = \dot{S}_{g,R} / \dot{S}_{g,S}$ - augmentation entropy generation number

Subscripts

i	inside
o	outside
w	wall
R	rough tube
S	smooth tube

INTRODUCTION

The results of the experimental researches for determination of the coefficients of hydraulic resistance and heat transfer at turbulent flow in two spirally corrugated tubes, combined with four twisted tapes with different pitches, are presented in [1]. As can be seen, the increase in the heat transfer coefficient h_i, when using surface turbulizers is always accompanied by an increase in the coefficient of hydraulic resistance f. Then the question is: "Which surface is the best?" It is obvious that the thermo-hydrodynamic characteristics of the tube are the fundamental factor for making the choice, based on systematized evaluation criteria. These criteria are derived on the basis of the first and second laws of thermo-dynamics, pursuing respective goals.

PERFORMANCE EVALUATION

Based on an analysis of the first law of thermo-dynamics, several authors Webb [2] and Bergles et al. [3] have proposed performance evaluation criteria (PEC) which define the performance benefits of an heat exchanger

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having enhanced surfaces, relative to a standard heat exchanger with smooth surfaces subject to various design constraints.

The evaluation criteria based on the first law of thermodynamics pursue one of the following objectives:

1. To reduce the heat transfer area of the heat exchanger *A*, ie. $A_* < 1$, while maintaining the heat output $\dot{Q}$, $Q_* = 1$ and the pumping power to overcome the hydraulic resistances *P*, $P_* = 1$.

2. To increase the thermal conductivity *UA*, keeping the pump power *P* and the total length of the tubes *NL*. The higher value of *UA* can be used in two ways: a) to increase the heat output $\dot{Q}$, $Q_* > 1$, at a constant inlet temperature difference between two fluids, $\Delta T_i^* = 1$; b) a given heat output to be transferred at a lower driving temperature difference, $Q_* = 1$, $\Delta T_i^* < 1$.

3. To reduce the pumping power *P*, $P_* < 1$, keeping the heat output $Q_* = 1$ and the heat transfer area, $A_* = 1$.

The evaluation criteria that formalize the described objectives and the defining equations are described in detail in [2, 4, 6] and successfully applied to assess the efficiency of spirally corrugated tubes for condensers [6, 7]. The basic equations that follow from the first law of thermodynamics are the following:

a) power ratio for overcoming the hydraulic resistances when the cooling fluid passes through the heat exchanger

$$P_* = f_* A_* w_*^3 \tag{1}$$

b) ratio of the overall conductance coefficients of the two heat exchangers

$$\frac{(UA_i)_R}{(UA_i)_S} = \frac{1+\beta_S}{\dfrac{St_S}{St_R}\left(f_* P_*^{-1} A_*^{-2}\right)^{1/3} + \beta_R A_*^{-1}}, \tag{2}$$

where β reflects chain of thermal resistances along the heat flow path, including those from pollution on the inner and outer side of the tube. The equations of heat transfer areas and mass flow rate are:

$$A_* = N_* L_* \left(D_i\right)_*, \tag{3}$$

$$W_* = N_* w_* \left(D_i\right)_*^2, \tag{4}$$

c) heat power rate

$$Q_* = W_* \varepsilon_* \Delta T_i^*, \tag{5}$$

where: $W_* = W_R / W_S$ - the mass flow rate of the cooling fluids through the heat exchangers, $\varepsilon_* = \varepsilon_R / \varepsilon_S$ - ratio of heat exchanger effectiveness, $\Delta T_i^* = \Delta T_{i,R} / \Delta T_{i,S}$ - ratio of inlet temperature difference between hot and cold streams for both apparatuses.

On the other hand it is well established that the minimization of the entropy generation in any process leads to conservation of useful energy. A solid thermodynamic basis to evaluate the merit of augmentation techniques by the second law analysis has been proposed by Bejan [4,5] developing the entropy generation minimization (EGM) method also known as "thermodynamic optimization".

The two heat exchangers are evaluated depending on the flow of generated entropy during their operation and the evaluation of the influence of the turbulizers is given by the ratio of the flows of generated entropy.

$$N_S = \dot{S}_{g,R} / \dot{S}_{g,S}. \tag{6}$$

Turbolizers in which $N_S < 1$, are thermodynamically more advantageous, since in addition to the increased heat transfer is added and a smaller consumption of exergy in the operation of the apparatus.

An extended version of the criteria under the second law of thermodynamics, under boundary condition $T_w = const$ and taking into account the change in fluid temperature along the length of the tube, is presented in [8].

The evaluation criteria are divided into three main groups depending on the geometric constraints: FG (fixed geometry); FN (fixed number of pipes) and VG (variable geometry). In this study, two spirally corrugated tubes combined with four twisted tapes [1] were evaluated only on the criteria FG-1a, VG-1 and VG-2a [2] in the conditions of a real operating condenser (water heater).

Fixed geometry criteria (FG)

The fixed geometry criteria involve a one-for-one replacement of smooth tubes by enhanced ones of the same basic geometry, e.g., tube envelope diameter, tube length and number of tubes in the tube bundle. The FG-1 cases [2] seek increased heat duty or overall conductance *UA* for constant exchanger flow rate. The pumping power of the enhanced tube exchanger will increase due to the increased fluid friction characteristics of the enhanced surface. For these cases the constraints, inlet temperature difference between two fluids, $\Delta T_i^* = 1$, $W_* = 1$, $N_* = 1$ and $L_* = 1$ require $P_* > 1$.

When the objective is increased heat duty $Q_* > 1$, this corresponds to the case FG-1a, [2,3]. Figure 1 shows the change of Q_* depending on *Re* for all combinations 340 - 344 and 360 – 364. In this case, the heat duty of the unit with spirally corrugated tubes combined with a twisted tape increases significantly.

For tube 340 the increase is in the range 64 - 50% ($Q_* = 1.64 - 1.50$), while for 360 the values are lower 47 - 37% ($Q_* = 1.47 - 1.37$). When the tubes are combined with a twisted tapes, the values of Q_* increase further with decreasing tape pitch and reach maximum values at the smallest step. For tube 344 the increase is $Q_* = 1.98 - 2.00 - 1.95$, while for 364 the values are significantly lower - $Q_* = 1.71 - 1.75 - 1.66$.

In this case, it should be noted that the increase in Q_* is accompanied by an increase in pumping power $P_* > 1$ and it may be necessary to replace the pump with a more powerful.

The augmentation entropy generation number N_S is calculated from Zimparov [8]. For this case it is

$$N_S = \frac{1}{1+\phi_o}\left\{Q_* \exp\left\{B\left[1-\frac{St_R}{St_S} D_*^{-0.8}\right]\right\}\times\right.$$

$$\left.\left\{\frac{T_{i,S}}{T_{o,S}} + Q_*\left[1-\frac{T_{i,S}}{T_{o,S}}\right]\right\}^{-1} + \phi_o \frac{f_R / f_S}{D_*^{4.75}}\right\} = f\left(Re_R\right). \tag{7}$$

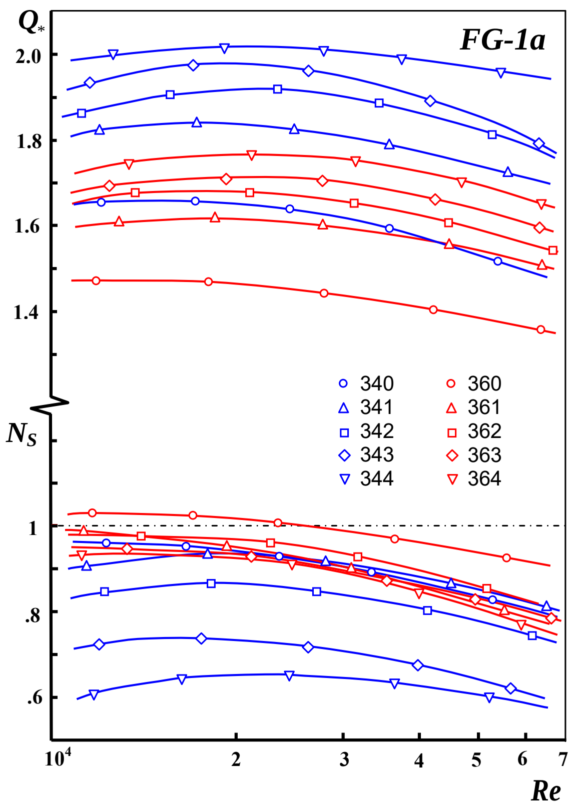

Fig. 1. Increased heat transfer rate and augmentation entropy generation number vs. Reynolds number

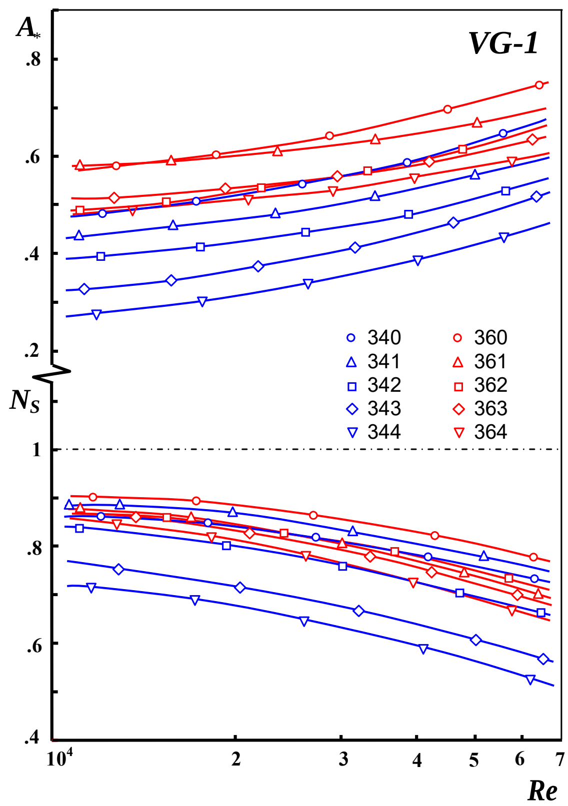

Fig. 2. Reduced heat transfer surface area and augmentation entropy generation number vs. Reynolds number

As can be seen from Fig. 1, all the tubes reduce the entropy generation evaluated through the number N_S. The minimum values of N_S for the corrugated tubes applied alone are $N_S = 0.95 - 0.80$ (tube 340) and $N_S = 1.03 - 0.91$ (tube 360). When the compound enhancement technique of this kind is applied this minimum values of N_S approach $N_S = 0.58 - 0.65$ (tube 344) and $N_S = 0.93 - 0.76$ (tube 364).

This leads to an additional significant gain from the reduced flow of generated entropy in the heat exchanger.

Variable geometry criteria (VG)

In most cases a heat exchanger is designed for a required thermal duty with a specified flow rate. Since the tube-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 permit the exchanger flow frontal area of bundle to vary, $N_* > 1$, in order to meet the pumping power constraint $P_* = 1$. In the case VG-1 [2, 3] the objective is to reduce heat transfer surface area $A_* < 1$ with constraints $Q_* = 1$, $P_* = 1$, $W_* = 1$ and $\Delta T_i^* = 1$.

The entropy generation number is calculated from [8],

$$N_S = \frac{1}{1+\phi_o}\left\{\left(\frac{f_R}{f_S}A_*\right)^{-0.364} D_*^{2.091}\, exp\left\{B\left[1-\frac{St_R}{St_S}\times \left(\frac{f_R}{f_S}\right)^{-0.291} A_*^{0.709} D_*^{-0.127}\right]\right\}+\phi_o\right\} = f\left(Re_R\right). \quad (8)$$

In Fig. 2 shows the estimates for the two groups of tubes. The best indicators are again for combination 344, which can achieve the greatest reduction of heat transfer area, which is 73 - 55% ($A_* = 0.27 - 0.45$). For the other tube, the best performance is the combination 364, in which the reduction of heat transfer area is 51 - 39% ($A_* = 0.49 - 0.61$). The reduction of the entropy generation is also significant, $N_S = 0.72 - 0.52$ for tube 344 and $N_S = 0.85 - 0.65$ for tube 364. It is seen that with reducing step of the twisted tapes the efficiency under both laws of thermodynamics increases.

The cases VG-2 [2, 3] aim at increased thermal performance for $A_* = 1$ and $P_* = 1$. When the objective is $Q_* > 1$, for case VG-2a, the additional constraint is $\Delta T_i^* = 1$.

The last case considered is VG-2a where the objective is increased heat rate $Q_* > 1$ for $W_* = 1$ $A_* = P_* = 1$ and $\Delta T_i^* = 1$. The values of N_S are calculated following [8],

$$N_S = \frac{1}{1+\phi_o}\left\{\left(\frac{f_R}{f_S}A_*\right)^{-0.364} D_*^{2.091}\, exp\left\{B\left[1-\frac{St_R}{St_S}\times \left(\frac{f_R}{f_S}\right)^{-0.291} D_*^{-0.127}\right]\right\}\left\{\frac{T_{i,S}}{T_{o,S}}+Q_*\times \left(1-\frac{T_{i,S}}{T_{o,S}}\right)\right\}^{-1}+\phi_o\right\} = f\left(Re_R\right) \quad (9)$$

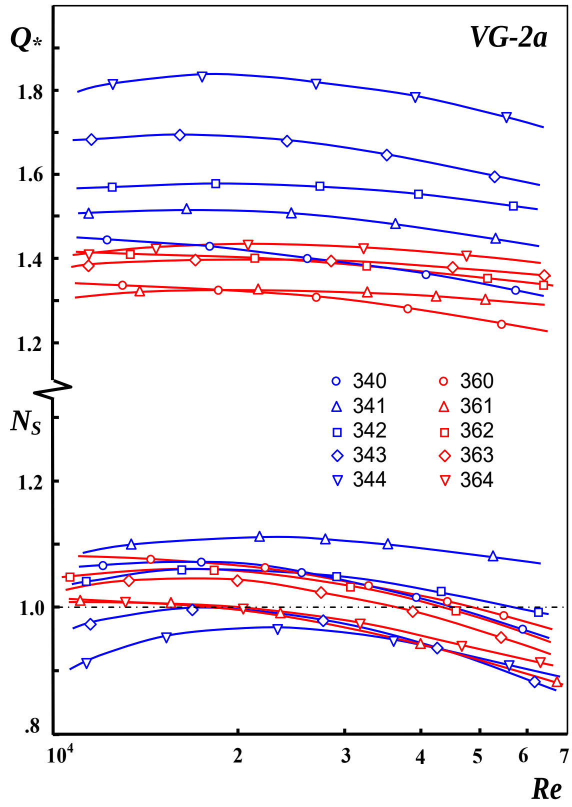

Fig. 3. Increased heat transfer rate and augmentation entropy generation number vs. Reynolds number

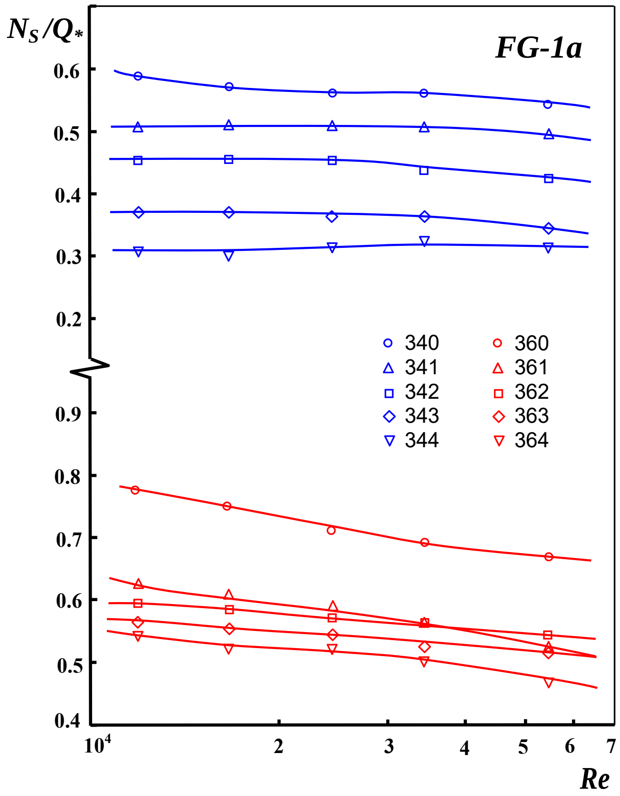

Fig. 4. The ratio N_S/Q_* vs. Reynolds number

The results are presented in Fig. 3, as changes of Q_* and N_S for the studied range of *Re*. When the heat exchanger is furnished with corrugated tubes alone $Q_* = 1.45-1.32$ (tube 344) and $Q_* = 1.33-1.23$ (tube 360). When is used a combination with a twisted tape, the values of Q_* reach $1.80-1.83-1.73$ (tube 344) and $Q_* = 1.41-1.43-1.40$ (tube 364). The values of N_S are as follows: $N_S = 1.08-0.96$ for the corrugated tube alone (tube 340) and $N_S = 1.09-0.97$ (tube 360), $N_S = 0.89-0.97$ for a compound enhancement technique (tube 344) and $N_S = 1.01-0.91$ (tube 364).

Highs there again the combination 344. It can achieve the highest increase of the heat rate of the unit at the lowest reduction of the entropy generation.

When inspecting some cases, it is possible for a tube to have best performance assessed by first-law analysis whereas another one has the best performance using the second-law analysis. Then, the question is: On what basis shall we select the tube – first law or second law?

Discussion of this issue suggests that the evaluation and comparison of the heat transfer enhancement techniques should be 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 $N_S < 1$, 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 $A_* N_S = f(Re_R)$ might be defined to connect the two objectives pursued by the first and second law analysis and as a basis for thermodynamic optimization.

Figures 4 and 6 shows $N_S / Q_* = f(Re)$ for the cases FG-1a and VG-2a and Fig. 5 show $A_* N_S = f(Re_R)$ for the case VG-1. For all the cases considered, on all three criteria, the best performance have one and the same tube – 344 being far superior to others.

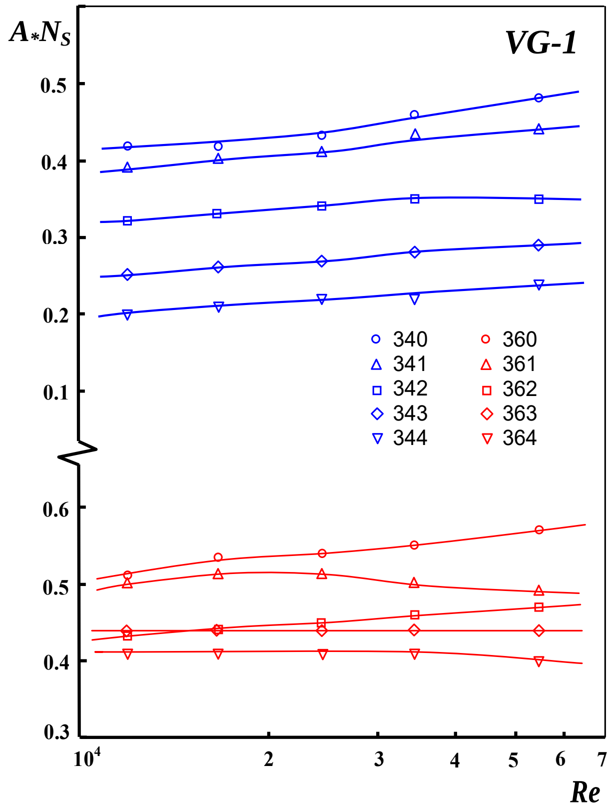

Fig. 5. Group $A_* N_S$ vs. Reynolds number

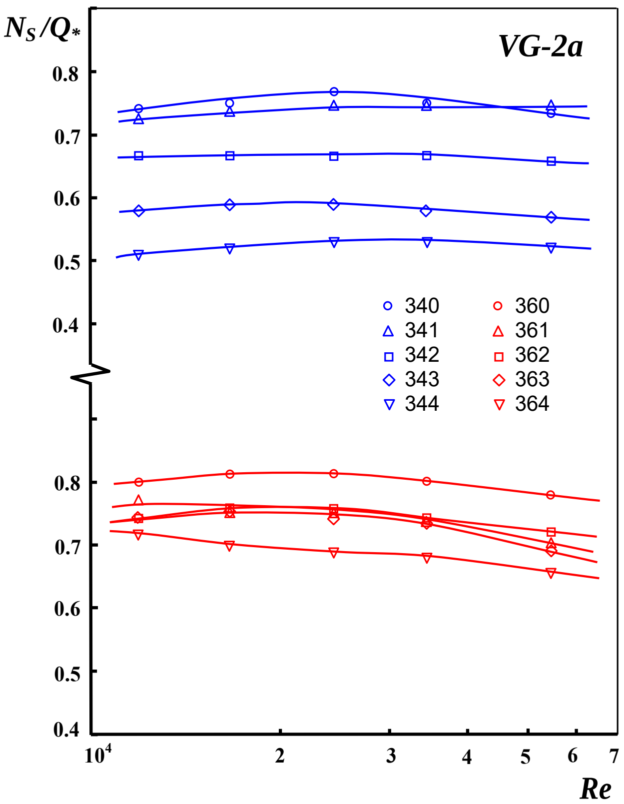

Fig. 6. The ratio N_S/Q_* vs. Reynolds number

CONCLUSIONS

The use of corrugated tube in conjunction with twisted tape is proposed as method of enhancing heat transfer in single-phase turbulent flow. It has been shown experimentally that the heat transfer is enhanced very considerably when the corrugated tube has a certain values of the geometrical parameters e/D_i, p/e, β_* and the relative peach H/D_i decreases.

Extended performance evaluation criteria have been used to assess the benefits of replacing the smooth tubes with spirally corrugated tubes combined with twisted-tape inserts. An additional increase of the heat transfer rate or reduction of heat transfer area can be achieved by an appropriate combination of corrugated tube with twisted tape. The reduction of the entropy generation is also significant. The results discussed imply that the evaluation and comparison of the heat transfer enhancement techniques should be made on the basis of both the first and the second law analysis.

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