Second law analysis of viscous flow through rough tubes subjected to constant heat flux

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Abstract: Entropy generation for a viscous, forced convection through enhanced rough tubes subjected to constant heat flux was numerically investigated. The entropy generation and exergy destruction due to the flow friction and heat transfer was proposed to evaluate the benefits of the utilisation of different enhanced tubes. The model was based on either measurements or empirical correlations for both the flow and heat transfer characteristics in plain and enhanced tubes. Flow of different fluids in rough tubes with threedimensional internal extended surfaces (3-DIES) were studied. Enhanced rough tubes, either with continued or regularly spaced tape inserts were investigated. Based on exergy performance, it was found that the use of hollow 3-DIES tubes in water turbulent flow regime is preferable while segmented twistedtape inserts is attractive in laminar flow. Also, inline 3-DIES rough tubes indicate a lower exergy destruction rate when compared with that of staggered alignments. The correlation for both optimum Reynolds number (Re^*) and minimum percentage exergy destruction (Ψ_{min}%) were obtained for oil and Ethylene Glycol flows in inline 3-DIES rough tubes as a function of the exposed heat flux, Prandtl number and the tube ratio (L/D).

Keywords: forced convection; three-dimensional internal extended surfaces; exergy; tape inserts.

Reference to this paper should be made as follows: Abdel-Moneim, S.A. and Ali, R.K. (2006) 'Second law analysis of viscous flow through rough tubes subjected to constant heat flux', *Int. J. Exergy*, Vol. 3, No. 3, pp.323–343.

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1 Introduction

Different methods for enhancing the heat transfer rate in forced convection, to reduce the size of the heat exchanger and save energy, were discussed by Bergles (1998). Enhancement techniques in fully developed turbulent flows in tubes are classified into: i) methods in which the inner surface of the tube is roughened e.g., with repeated or helical ribbing, by sanding, with internal fins or corrugation ii) methods in which a heat transfer promoter e.g., twisted tape, disks or stream lined shapes are inserted into the tubes. Various methods for such devices have been developed and tested by Bergles (1989) and Ravigururajan and Bergles (1985). Durmus (2004) studied the effect of inserted conical turbulators on the heat transfer and fluid flow characteristics for isothermally heated tubes at the outer surface. Marner and Bergles (1989) tested the effect of inserted twisted tape or spirally internal fins on heat transfer and fluid flow characteristics, with the outer surface of the tube subjected to isothermal heating. Saha et al. (1989) presented experimentally, the characteristics of heat transfer and pressure drop of laminar flow in a circular tube fitted with straight and twisted tapes. Liao and Xin (1995) studied experimentally, the effect of the presence of threedimensional internally extended surfaces(3-DIES) on forced convective heat transfer and pressure drop of Ethylene Glycol in tubes.

Compound enhancement can be obtained by utilising two or more of the existing techniques. Usui et al. (1986) investigated grooved tubes with twisted tape inserts while Zimparov (2002) studied the combination of corrugated tubes and twisted tape inserts. Also, Liao and Xin (2000) studied the augmentation of convective heat transfer inside tubes with three-dimensional internal extended surfaces and twisted tape inserts. Several criteria for evaluating the effectiveness of these augmentation techniques have proposed. Webb (1981) and Abdel-Moneim et al. (2000) have used the efficiency index as a performance evaluation criterion (PEC) to evaluate the performance benefits for varieties of augmentation techniques. The method proposed by Bergles et al. (1974) compares the performance of the augmented surface heat exchanger to meet a defined objective such as maximising the heat load or reducing the surface area.

Heat transfer processes are accompanied by exergy destruction due to entropy generation. This entropy generation is mainly due to the irreversible nature of heat transfer across a finite difference in temperatures and the fluid flow friction accompanied with the augmentation techniques. Therefore, in optimised energy conversion systems, it is essential to understand how entropy is being generated in the convective heat transfer processes to avoid exergy destruction. Bejan (1979, 1980) and San et al. (1987) analysed the entropy generation based on the aspects of the 2nd law of thermodynamics for a convective heat transfer process in different fundamental flow configurations. Sahin (1998) developed a theoretical model based on the exergy concept to predict the entropy generation for a fully developed laminar flow in a duct subjected to a constant wall temperature. Against the results of Bejan (1979), Sahin (1998) found that the entropy generation increases along the duct length. In addition, the entropy generation number following Sahin (1998) is an important parameter in deciding the true merit of a heat transfer augmentation technique. Bejan and Pfistr (1980) and Abdel-Moneim (2002) proposed an evaluation of heat transfer augmentation techniques based on entropy generation. Prasad and Shen (1994) applied a performance evaluation criterion, based on the exergy analysis, to a tubular heat exchanger with wire-coil inserts. In this analysis, the net exergy destruction resulting from the effect of heat transfer across a finite temperature difference and from the flow friction was used as an evaluation criterion. Also, a thermodynamic optimum was obtained by minimising the net exergy destruction rate. Abdel-Moneim and Ali (2005) analysed the entropy generation, based on the 2nd law for a convective heat transfer process in different fundamental flow configurations.

Nag and Mukherjee (1987) modified Bejan's entropy generation method by including the effect of the variation of fluid temperature along a heat transfer duct with constant wall temperature. Also, Zimparov (2000) applied this modified performance evaluation criterion to ten spirally corrugated tubes to assess the benefits of the use of these tubes for heat transfer augmentation. It was found that a 'rib height to diameter' ratio of about 0.04 is an optimum for this type of studied tubes.

In the present work, an evaluation method based on exergy analysis is proposed using the principles of the first and second laws of thermodynamics. The exergy destruction is estimated in nondimensional form. The proposed evaluation technique has been applied on rough tubes containing three dimensional internal extended surfaces (3-DIES) for flows with different fluids. The numerical study is extended to investigate the effect of straight and twisted tapes inserted in both rough and smooth tubes. Also, the effect of segmentation of the inserted tape was studied. The present study aims to provide heat transfer equipment designers with knowledge of the thermodynamic impact due to the entropy generation and exergy destruction associated with the use of different enhanced techniques.

2 Theoretical modelling

The energy balance for a control volume of a length dx for flow with heat transfer through a duct with arbitrary cross section and subjected to a uniform heat flux as shown in Figure 1(a) is,

$$\delta Q = m\, c_p\, \mathrm{d}T. \tag{1}$$

and the heat transfer equation is,

$$\delta Q = q(w\, \mathrm{d}x) = h\, \Delta T\; w\, \mathrm{d}x. \tag{2}$$

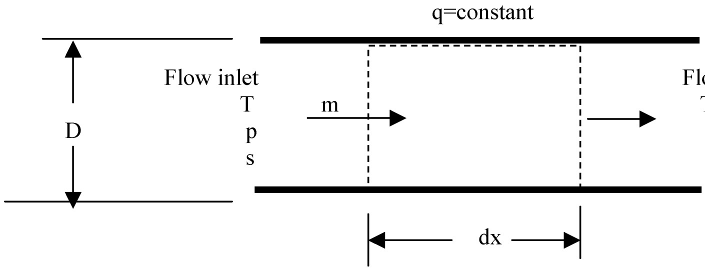
Equations (1) and (2) can be combined to give the flow bulk temperature distribution in the differential form as:

$$\mathrm{d}T = \frac{hw\, \Delta T}{mc_p} \mathrm{d}x. \tag{3}$$

Assuming a steady incompressible flow and applying the entropy production theory (based on the 2nd law of thermodynamics, van Wylen et al. (1994)), the net entropy generation rate for internal flow equals the entropy generation of the flow (mds) plus that of the surroundings ($-\delta Q/T_w$). The minus sign shows that heat is lost from the surroundings. So, the net entropy generation rate for internal flow can be calculated as:

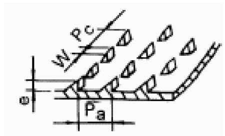
$$\mathrm{d}\dot{S}_{\text{gen}} = m\mathrm{d}s - \frac{\delta Q}{T_w}. \tag{4}$$

Figure 1 The control volume and nomenclatures of the studied enhancement methods: a) control volume in a tube subjected to a uniform heat flux; b) structural geometries of the 3-DIES tube#1 in Liao and Xin (1995); c) Structural geometries of the 3-DIES tube#1 Liao and Xin (2000); d) 3-DIEST tube#1 with continuous and segmented tapes Liao and Xin (2000) and e) layout of a fulllength and segmented twistedtape inserts inside a circular tube#2 presented by Saha et al. (1989)



(a)

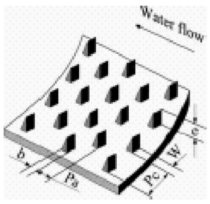
Configuration	e	Pa	W	Pc	Fin aligment
A	0.54	3.78	0.53	2.12	Staggered
B	0.54	4.32	0.61	2.42	Staggered
C	0.95	6.62	0.93	3.7	Staggered
D	0.54	3.78	0.53	2.12	In-Line
E	0.34	2.36	0.33	1.31	Staggered
F	0.74	5.2	0.73	2.91	Staggered
G	0.74	5.2	0.73	4	Staggered



All dimensions in mm

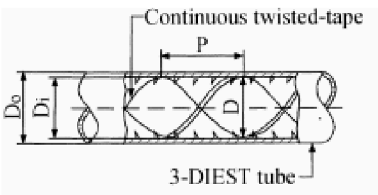
(b)

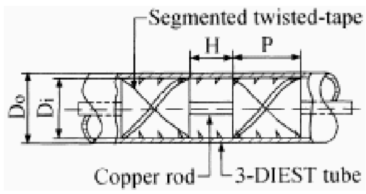
Configuration	e	Pa	W	Pc	Fin aligment
1	1.041	4.247	0.501	2.717	Staggered
2	1.323	5.239	0.597	3.286	Staggered
3	0.878	3.554	0.394	2.118	Staggered
4	1.03	4.234	0.474	2.569	In-Line



All dimensions in mm

(c)





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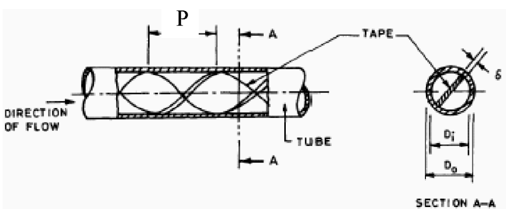
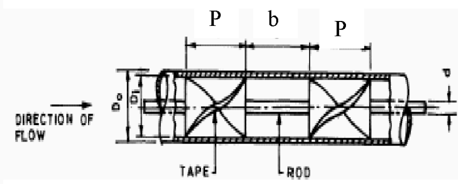




(e)

The entropy geneeration for the flow can be calculated from,

$$T\mathrm{d}s = c_p \mathrm{d}T - \frac{\mathrm{d}P}{\rho}. \tag{5}$$

Substituting the values of ds from equation (5) and δQ from equation (1) into equation (4), yields,

$$\mathrm{d}\dot{S}_{\text{gen}} = mc_p \left(\frac{1}{T} - \frac{1}{T_w} \right) \mathrm{d}T - m \frac{\mathrm{d}P}{\rho T}. \tag{6}$$

Rearranging the righthand side of equation (6) such that the term $((1/T) - (1/T_w))$ becomes $(T_w - T)/(\mathrm{T}\{(T_w - T) + T\})$, this leads to $(\Delta T/(T\{\Delta T + T\}))$ resulting in $(\tau/(T(\tau + 1))$, where $\tau = \Delta T/T$ is a dimensionless temperature difference (temperature difference number). Substituting the value of $(m\ c_p\ \mathrm{d}T)$ in equation (6) by $(h\ \Delta T_w\ \mathrm{d}x)$ as in equation (3), equation (6) becomes,

$$\mathrm{d}\dot{S}_{\text{gen}} = \left(\frac{\tau^2 hw}{\tau + 1} \right) \mathrm{d}x - \frac{m}{\rho T} \left(\frac{\mathrm{d}P}{\mathrm{d}x} \right) \mathrm{d}x, \tag{7}$$

where, the 1st term of the righthand side of equation (7) represents the entropy generation rate due to heat transfer across a finite temperature differenence, while the 2nd term represents the contribution of the flow friction in entropy production. Extensive details of this model were presented in Abdel-Moneim and Ali (2005). On substituting the following dimensionless parameters, equation (7) can be transformed into dimensionless form as:

$$\mathrm{d}\sigma = \left(\frac{\tau^2}{\tau + 1} \right) Nu\ \xi\ \mathrm{d}\chi + 2 \frac{SBr}{\mathrm{Pr}} F\ \mathrm{d}\chi \tag{8}$$

where,

$\sigma = \dfrac{\dot{S}_{\text{gen}}}{mc_p}$: is the entropy generation to thermal capacity ratio

$\tau = \dfrac{\Delta T}{T} = \dfrac{T_w - T}{T}$: is a dimensionless temperature difference

$\xi = \dfrac{w\ k}{mc_p}$: is a dimensionless flow parameter, $\xi = \dfrac{4}{\mathrm{Re_D\ Pr}}$ for pipe flow

$\chi = \dfrac{x}{D}$: is a dimensionless length

$SBr = \dfrac{\mu U_m^2}{kT}$: is pseudo Brinkman number in Prasad and Shen (1994)

$SBr = \dfrac{\mathrm{Pr}.U_m^2}{c_P T}.$

$F = \frac{(-\mathrm{d}P/\mathrm{d}x)D}{2\rho U_m^2}$: is the Fanning friction factor.

The net entropy generation rate $\dot{S}_{\text{gen}}$ can be obtained by integrating equation (8) along the entire length of the tube and by the definition of σ:

$$\dot{S}_{\text{gen}} = \sigma\ m\ c_p. \tag{9}$$

Also, the exergy destruction rate associated with the heat transfer process along the whole duct can be calculated by,

$$\Delta\Psi = T_0 \dot{S}_{\text{gen}} \tag{10}$$

where T_0 is the reference temperature in the thermodynamic scale, K.

3 Method of calculation

The integration of equation (8) requires either measurements or theoretical knowledge about the characteristics of both the heat transfer in terms of Nu and the flow friction in terms of F. In the case when the heat transfer and flow measurements are available, the integration of equation (8) can be accomplished according to the following stepwise procedure:

- with the knowledge of the heat flux (q), flow rate (m) and inlet flow temperature (T_i), equation (1) can be simply applied resulting in a flow bulk temperature distribution, $T(x)$
- the wall temperature along the tube, $T_w(x)$, can be calculated from equation (2) with the aid of the heat transfer characteristics in terms of Nu presented by Saha et al. (1989) and Liao and Xin (1995, 2000)
- the pressure distribution along the duct can be obtained from the fluid flow characteristics presented by Saha et al. (1989) and Liao and Xin (1995, 2000)
- the results of steps (1,2, 3), the local values of the dimensionless terms τ, χ, ξ, SBr, F and Nu can be calculated while Pr can be found at the flow bulk temperature
- by the integration of equation (8) along the duct length, the dimensionless term (σ) can be calculated and the net entropy generation rate $\dot{S}_{\text{gen}}$ can be found from equation (9)
- finally, the exergy destruction rate $\Delta\psi$ can be found by substituting into equation (10).

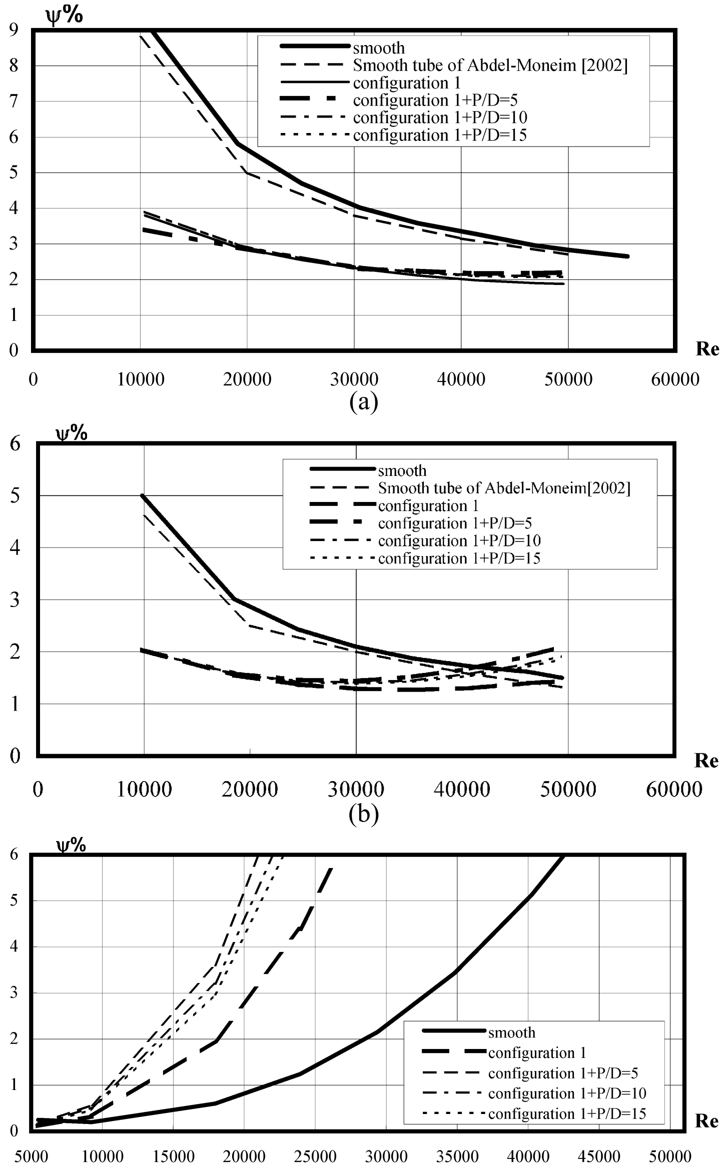
4 Results and discussions

Rough tubes with three-dimensional internal extended surfaces (3-DIES) and tape inserts are widely applied and a lot of studies for heat transfer and flow friction are available in laminar and turbulent flows. The present study was performed to evaluate the enhancement of heat transfer in rough tubes for water, Ethylene Glycol and engine oil flows. Therefore, in addition to the case of smooth tube, rough tubes with tape inserts are evaluated based on exergy analysis using the present performance evaluation criterion. In fact, for turbulent flow, the dominant thermal resistance is limited to a thin boundary layer. Thus a hollow rough tube with 3-DIES is probably more effective because it mixes the flow in the viscous sublayer near the wall and enhances the heat transfer. Rough tube named tube#1 (inner diameter = 13.5 mm and 1.0 m long) presented by Liao and Xin (1995, 2000) with 3-DIES was evaluated using the present performance evaluation criterion for ranges of laminar and turbulent flows ($100 < Re < 50000$). The dimensions of in-line and staggered alignments of 3-DIES are shown in Figures 1(b) and 1(c). The effect of continuous and segmented twisted tape ($5 \leq P/D \leq 15$) inside the rough tube#1 with 3-DIES shown in Figure 1(d) and presented by Liao and Xin (2000) was studied. For laminar flow, the dominant thermal resistance is not limited to the boundary layer. Thus a tube with twisted or straight tape inserts are probably more effective because twisted tapes can mix the bulk flow in the tube core. Smooth tube presented by Saha et al. (1989) and named tube#2 (inner diameter = 12 mm and 1.84 m long) with twisted ($3.4 \leq P/D \leq 10$) and straight tape inserts shown in Figure 1(e) is evaluated using the present PEC in the range of $500 < Re < 2500$. Also, the effect of using segmented tape in a range of $5 \leq b/D \leq 10$ on the exergy destruction rate was presented.

4.1 Exergy destruction rate for water flow

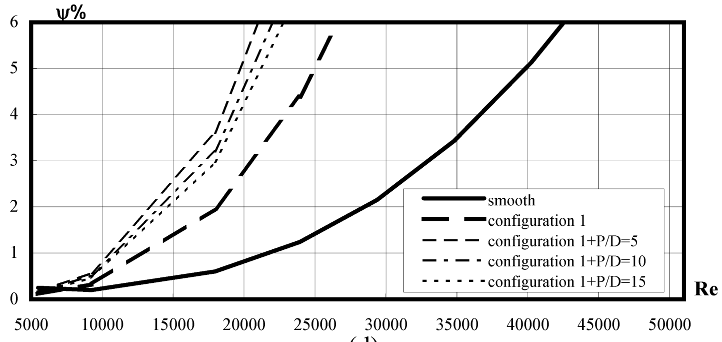
Before performing numerical runs, a validation of the exergy destruction rate of the present analysis was carried out by comparing the present predicated data with that of Abdel-Moneim (2002) for the case of water flow in a smooth tube#1 subjected to uniform heat fluxes as shown in Figure 2. It was found that discrepancies between the present data and that of Abdel-Moneim (2002) are less than 7% over a range of Reynolds numbers from 5000 to 50000. The exergy destruction rate is predicted utilising empirical correlations for both the heat transfer and the flow friction from Liao and Xin (2000). The percent exergy destruction rate, $\Psi\%$, for enhanced tubes initially decreases with the increase in Reynolds number as shown in Figure 2. In this range, the reduction in exergy destruction due to enhanced heat transfer is larger than the increase in exergy destruction due the increased flow friction. As Reynolds number increases, the flow friction becomes dominant and the exergy destruction increases. The combined effect results in a thermodynamic optimum Reynolds number (Re^*) with a minimum exergy destruction rate, $\Psi_{min}\%$. The value of Re^* depends on the heat flux, and its value increases with increasing heat flux. Flow in 3-DIES rough tubes almost sustains exergy destruction rate lower than that for rough tubes with twisted tape inserts as shown in Figure 2. This is due to the small enhancement effect of twisted tape inserts on the heat transfer accompanied by extremely high friction as discussed in Liao and Xin (2000). Also, the behaviour beyond Re^* is reversed due to the extreme increase in the flow friction compared with the enhancement in the heat transfer rate. Smooth tubes with straight and twisted tape inserts were investigated for water flow in the laminar regime and the results are shown in Figures 3 and 4. The exergy destruction rate is predicted utilising empirical correlations for both the heat transfer and the flow friction from Saha et al. (1989). It is observed that the rate of exergy destruction rate is affected by both flow velocity and the tape pitch as shown in Figure 3. It is clear that twisted tape with small P/D has a lower rate of exergy destruction. This is due to the effective enhancement of heat transfer which is dominant, especially in the laminar flow regime. Further more, segmentation of the twisted tape decreases the exergy destruction rate and this effect becomes significant with decreasing space between segments, as shown in Figure 4.

Figure 2 Percent exergy destruction rate for water flow in tube#1 presented by Liao and Xin (2000) with twisted tape inserts with different pitches at different heat fluxes (a) q = 93 kW/m^2, (b) q = 46 kW/m^2, (c) q = 4 kW/m^2, and (d) q = 1 kW/m^2



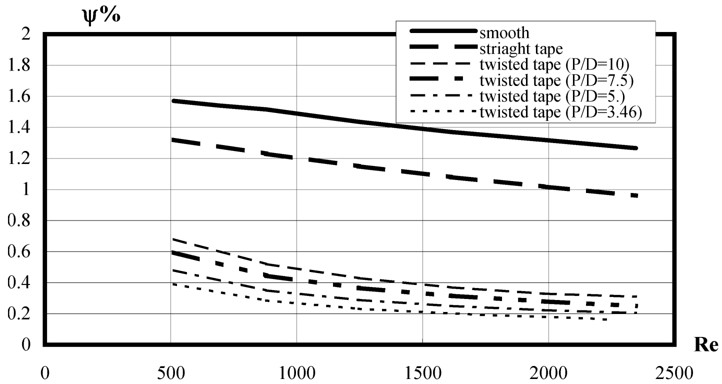
(c)

Figure 2 Percent exergy destruction rate for water flow in tube#1 presented by Liao and Xin (2000) with twisted tape inserts with different pitches at different heat fluxes (a) q = 93 kW/m^2, (b) q = 46 kW/m^2, (c) q = 4 kW/m^2, and (d) q = 1 kW/m^2 (continued)

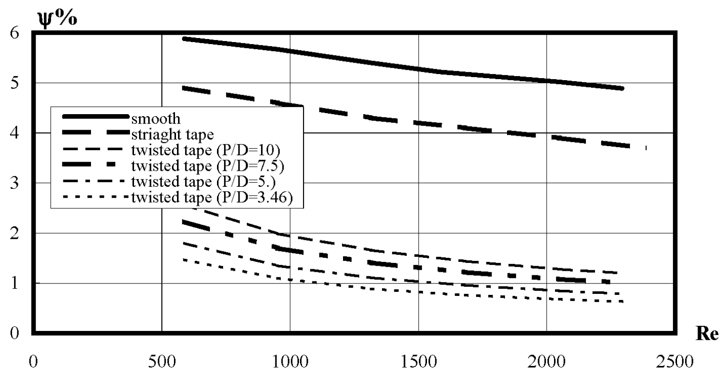


(d)

Figure 3 Exergy destruction rate for laminar water flow in tube#2 presented by Saha et al. (1989) with twisted tape inserts at different pitches and at different heat fluxes (a) q = 1 kW/m^2 and (b) q = 4 kW/m^2

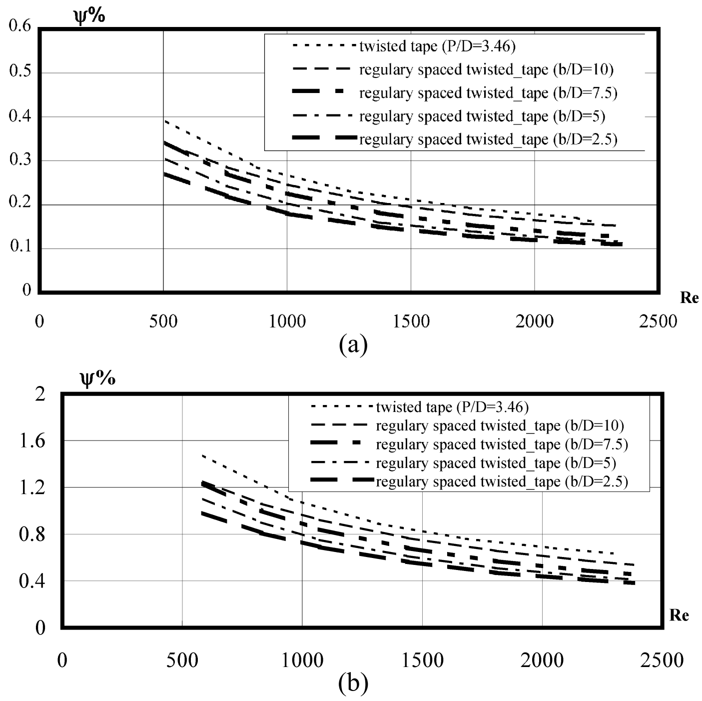


(a)



(b)

Figure 4 Exergy destruction rate for laminar water flow in tube#2 presented by Saha et al. (1989) with regularly spaced twisted tape inserts at P/D=3.46 and at different heat fluxes (a) q = 1 kW/m^2 and (b) q = 4 kW/m^2



4.2 *Exergy destruction rate for oil flow*

The exergy destruction rate is predicted using the experimental data from Liao and Xin (2000) for heat transfer and flow friction for oil flow in 3-DIES rough tubes with different configurations. As can be seen from Figure 5, twisted tape with small P/D has the lowest rate of exergy destruction at low Reynolds numbers. This is due to the augmentation of heat transfer which is dominant at low Reynolds numbers. However, at high Reynolds numbers, hollow 3-DIES rough tubes indicates the lowest rate of exergy destruction and a higher critical Reynolds number was obtained. From Figure 5, it can be concluded that hollow 3-DIES rough tubes are attractive, especially for laminar flow at a relatively high Reynolds number. This is due to its lower rate of exergy destruction with higher critical Reynolds number resulting in a higher Nusselt number and consequently lower surface area. The dimensionless entropy generation and exergy destruction rates for oil laminar flow in different configurations (in Figure 1(c)) of hollow 3-DIES rough tubes are illustrated in Figure 6. It is clear that inline 3DIES rough tubes (Configuration 4) indicate a lower entropy generation and exergy destruction rate. This may be attributed to the relative enhancement in heat transfer and the increase in the flow friction of inline configuration when compared with that of the staggered arrangement. Figure 7 illustrates the effect of the presence of segmented and continuous twisted tape in tube#1 on the exergy destruction rate. It is noticed that continuous and segmented twisted tape have lower rate of exergy destruction at low Reynolds numbers while its effect exceeds that of hollow 3DIES rough tubes with increasing Reynolds numbers. This may be attributed to the extremely increase in the high viscous flow friction with increasing Reynolds number.

Figure 5 Percent exergy destruction rate for laminar oil flow in smooth and rough tubes (with different configurations of three-dimensional internal extended surfaces) presented by Liao and Xin (2000) with twisted tape inserts (a) Configuration 1, (b) Configuration 2; (c) Configuration 3; and (d) Configuration 4

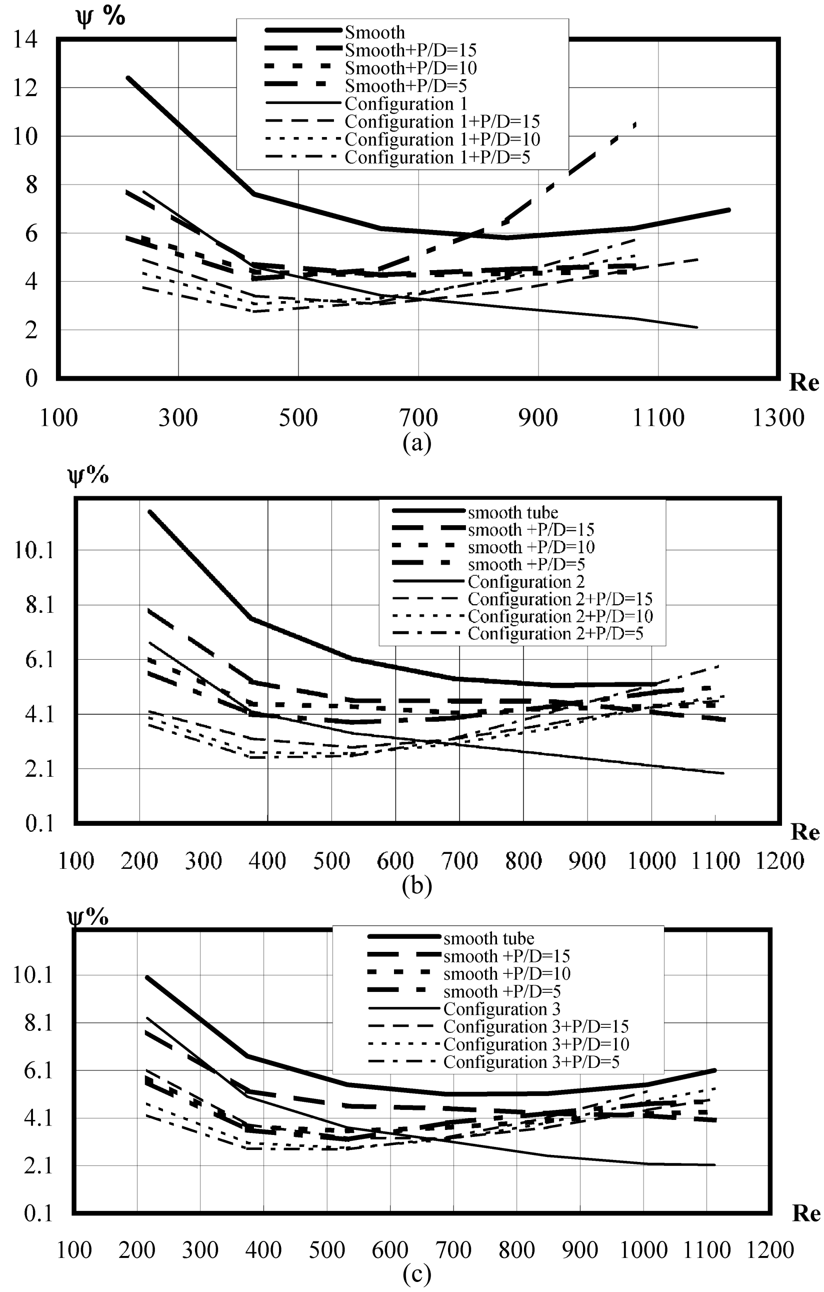
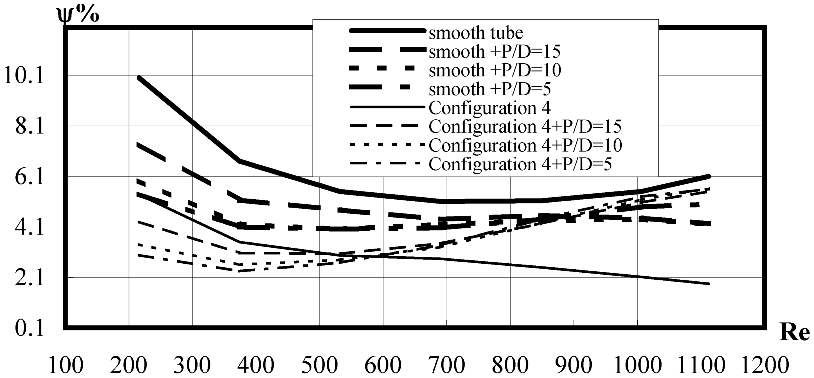
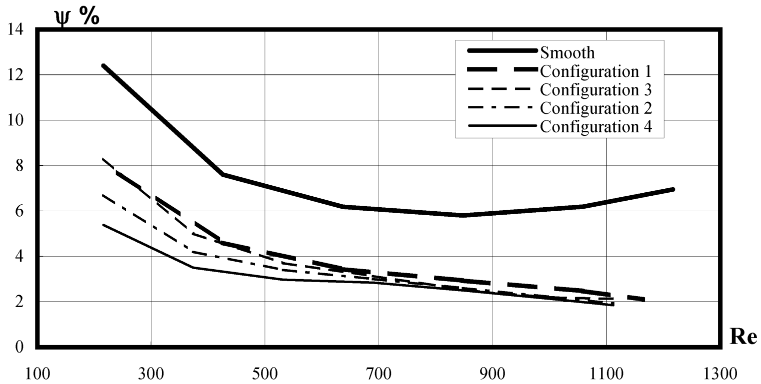


Figure 5 Percent exergy destruction rate for laminar oil flow in smooth and rough tubes (with different configurations of three-dimensional internal extended surfaces) presented by Liao and Xin (2000) with twisted tape inserts (a) Configuration 1, (b) Configuration 2; (c) Configuration 3; and (d) Configuration 4 (continued)

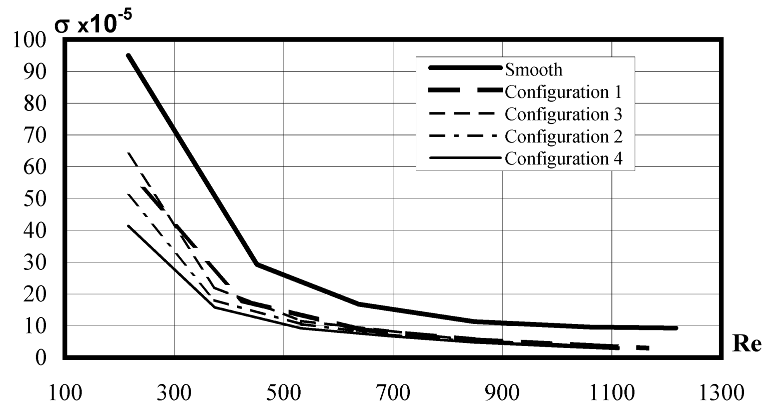


(d)

Figure 6 Dimensionless entropy generation and percent exergy destruction rate for laminar oil flow in tube#1 presented by Liao and Xin (2000) with different configurations of three-dimensional internal extended surfaces. (a) percent exergy destruction rate and (b) dimensionless entropy generation

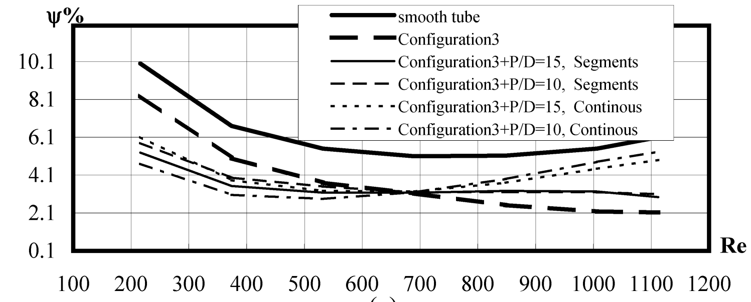


(a)

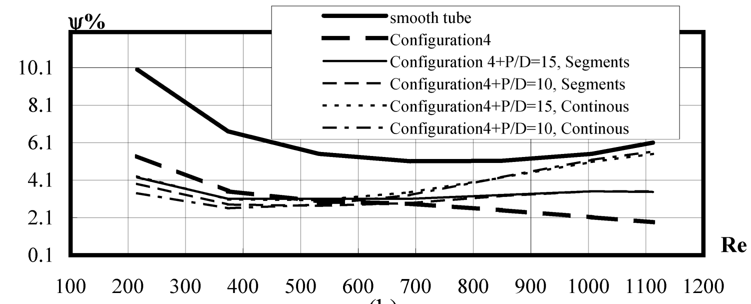


(b)

Figure 7 Percent exergy destruction rate for laminar oil flow in tube#1 (with different configurations of three-dimensional internal extended surfaces) presented by Liao and Xin (2000) with continuous and segment twisted tape (a) configuration 3 and (b) configuration 4



(a)

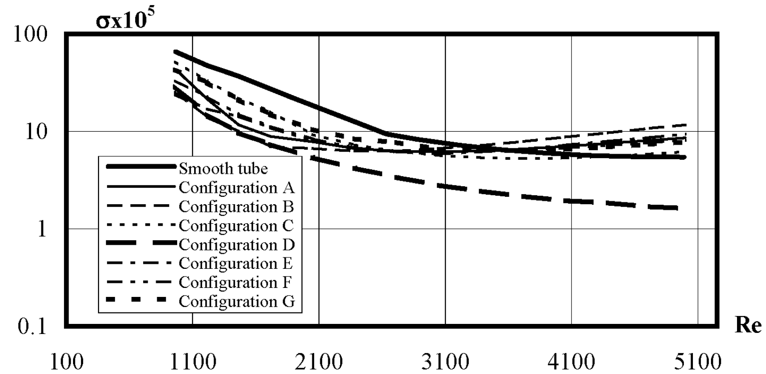


(b)

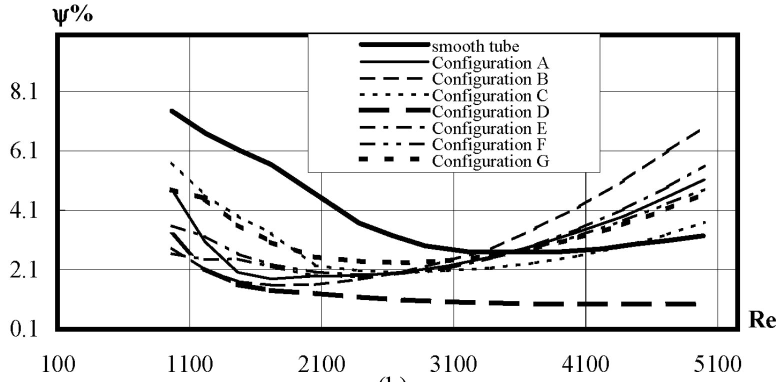
4.3 Exergy destruction rate for ethylene glycol flow

Figures 8–11 show the exergy analysis for Ethylene Glycol flow in augmented rough tubes over $100 < Re < 5000$ utilising the experimental data for heat transfer and flow friction from Liao and Xin (1995, 2000). It is found that inline 3-DIES rough tubes (configuration D in Figure 1(b)) have lower exergy destruction compared with all staggered arrangements. This is according to the relative thermal performance for inline and staggered alignments. The effect of heat flux and inlet fluid temperature is shown in Figures 9 and 10. In fact, increasing the heat flux increases the exergy destruction rate as a result of the increase in the temperature difference between the fluid and the tube surface, which increases the heat transfer irreversibility. Also, the rate of exergy destruction increases with the increase in inlet temperatures. This may be attributed to the decrease in Prandtl number of Ethylene Glycol which decreases the heat transfer rate and consequently increases temperature difference between the fluid and the tube surface. Figure 11 shows Ψ vs. Re for Ethylene Glycol flow in 3-DIES rough tubes shown in Figure 1(d) with twisted tape inserts at different values of *P*/*D*. A thermodynamic optimum was found at earlier Reynolds numbers as a result of high viscosity of Ethylene Glycol and the high pressure drop due to the presence of the twisted tape. The value of the critical Reynolds number depends on the twisted-tape pitch ratio, *P*/*D*, in addition to the heat flux as shown in Figure 11.

Figure 8 Dimensionless entropy generation and percent exergy destruction rate for Ethylene Glycol flow in tube#1 presented by Liao and Xin (1995) with with different configurations of three-dimensional internal extended surfaces (q = 18 kW/m^2, Ti = 310 K) (a) dimensionless entropy generation and (b) percent exergy destruction rate

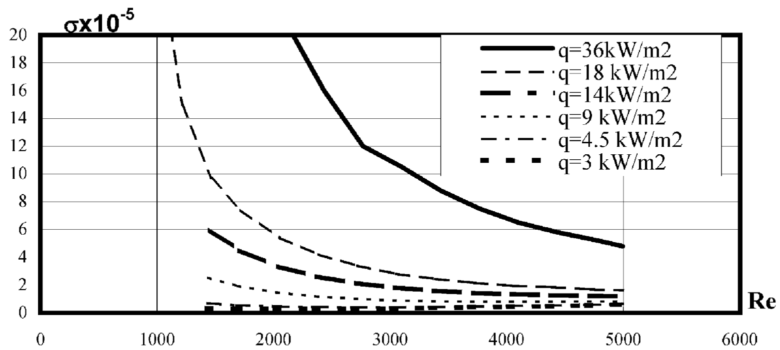


(a)



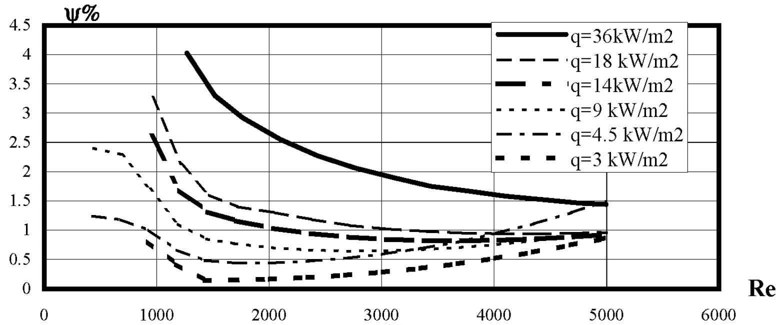
(b)

Figure 9 Dimensionless entropy generation and percent exergy destruction rate for Ethylene Glycol flow in tube#1 presented by Liao and Xin (1995) for different heat fluxes (Ti = 310 K) (a) dimensionless entropy generation and (b) percent exergy destruction rate



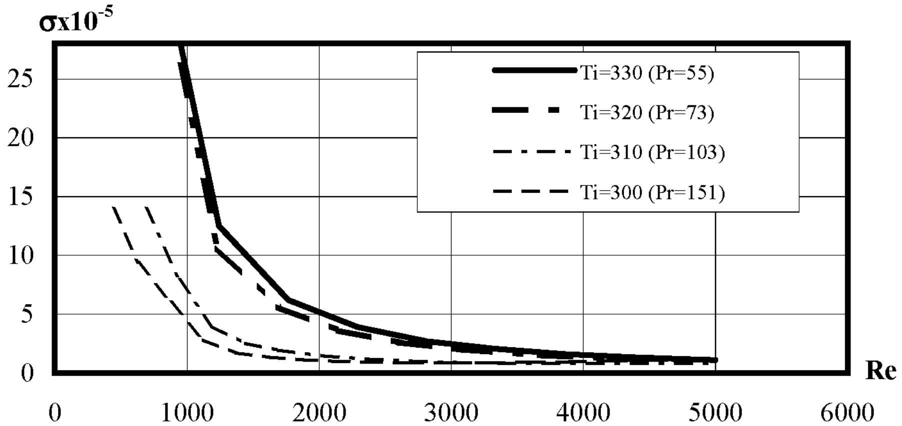
(a)

Figure 9 Dimensionless entropy generation and percent exergy destruction rate for Ethylene Glycol flow in tube#1 presented by Liao and Xin (1995) for different heat fluxes (Ti = 310 K) (a) dimensionless entropy generation and (b) percent exergy destruction rate (continued)

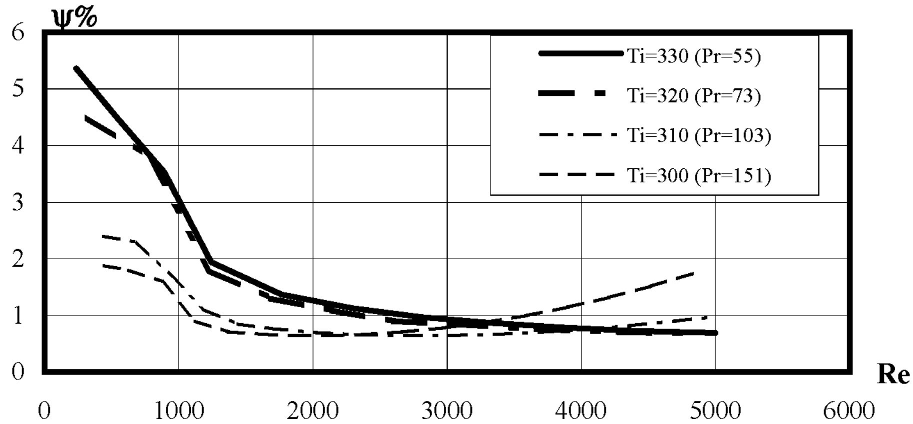


(b)

Figure 10 Dimensionless entropy generation and percent exergy destruction rate for Ethylene Glycol flow in tube#1 (Configuration D) presented by Liao and Xin (1995) for different inlet temperatures at q = 9 kW/m^2

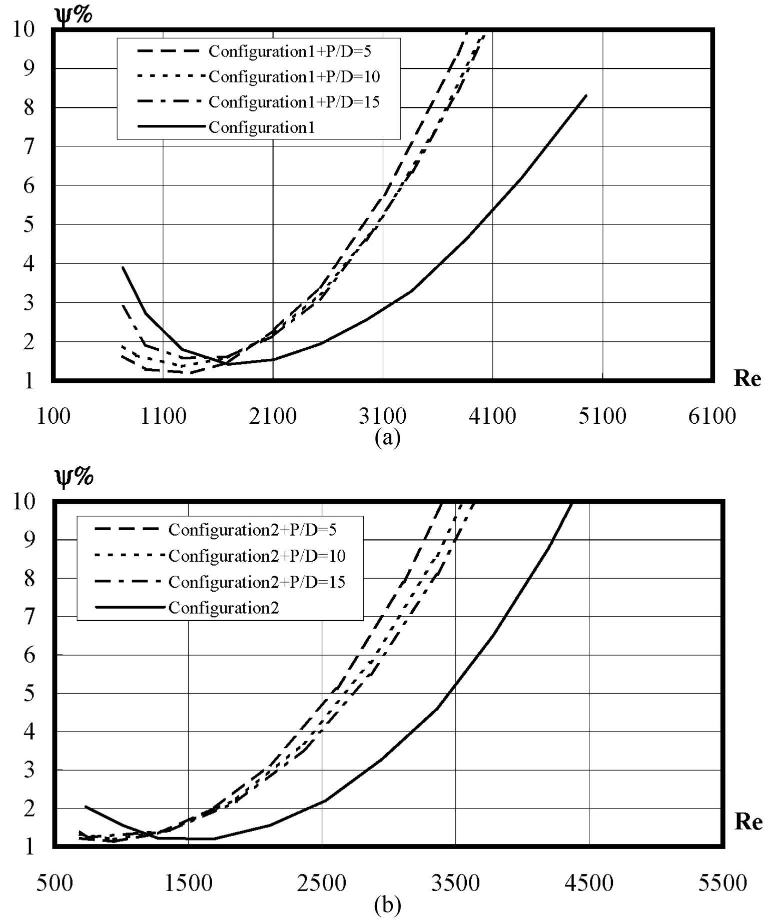


(a)



(b)

Figure 11 Percent exergy destruction rate for Ethylene Glycol flow in tube#1 presented by Liao and Xin (2000) with twisted tape inserts (q = 9 kW/m2 and Ti = 310 K)



On view of the present study, exergy optimisation becomes more significant in new designs and even in retrofitting of the existed heat transfer equipment. Also, empirical correlations that can determine an optimum Reynolds number that has minimum rate of exergy destruction are more essential for the designers. Therefore, the present model was used to predict the minimum rate of exergy destruction and to specify the corresponding optimum Reynolds number for oil and Ethylene Glycol in inline 3-DIES rough tubes having lower rate of exergy destruction as follows:

For ethylene glycol flow

$$\mathrm{Re}^* = \exp(0.7964q'' + 0.000562(L/D) - 0.00781\,\mathrm{Pr} + 8.11) \qquad (11)$$

$$\psi_{\min}\% = 3.3971\,\mathrm{Re}^{*-0.3781}\exp(-0.000387\,\mathrm{Re}^*). \qquad (12)$$

Figure 12(a) shows that the correlations, equations (11) and (12), are valid, with maximum deviations of $\pm$11% in estimating Re* and $\Psi_{\min}$% within the ranges of the different parameters as:

$0.2343 \leq q'' \leq 1.2497, 37 \leq L/D \leq 222$ and $58 \leq \Pr \leq 108$.

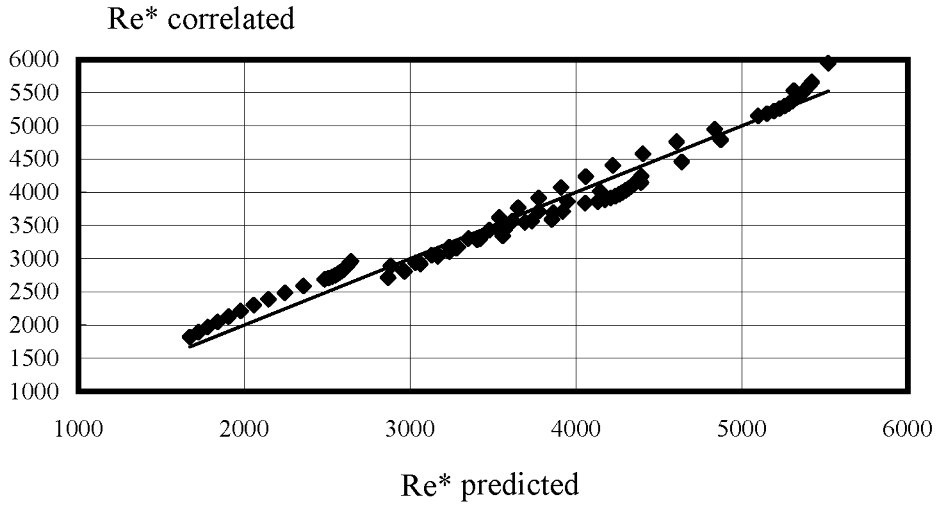
For oil flow

$$\mathrm{Re}^* = \exp(0.9027q'' + 0.0001166(L/D) - 0.003757\,\Pr + 6.79) \tag{13}$$

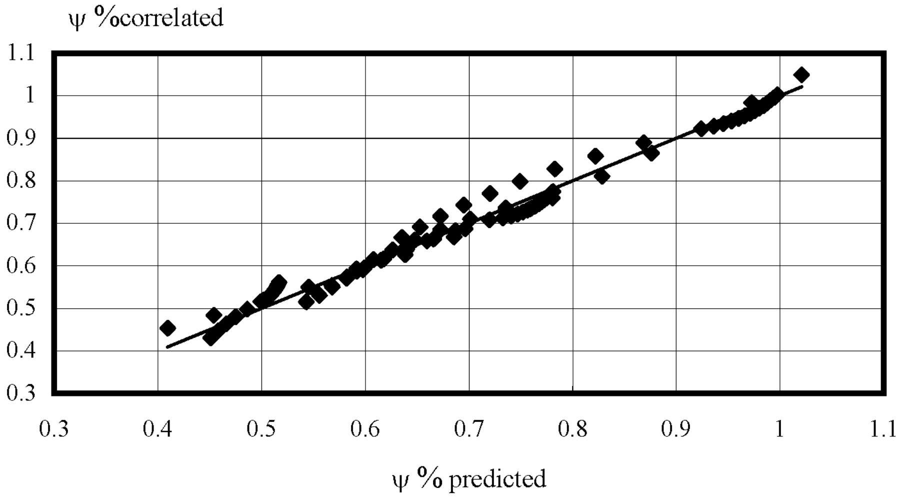
$$\psi_{\min}\,\% = \exp(0.798q'' - 0.000558(L/D) - 0.00012\,\Pr + 0.389). \tag{14}$$

These correlations, equations (13) and (14), are valid within $\pm$9.8% maximum deviations in both Re^* and $\Psi_{\min}$% for $0.0905 \leq q'' \leq 1.267$, $74 \leq L/D \leq 222$ and $293 \leq \Pr \leq 535$.

Figure 12 Validity of present correlations for mimimum percent exergy destruction rate and critical Reynolds number (a) validity of the present correlations for Ethylene Glycol flow in in-line 3-DIEST configuration (tube#1 discussed by Liao and Xin (1995)) and (b) validity of present correlations for oil flow in in-line 3-DIEST configuration (tube#1 discussed Liao and Xin (2000))

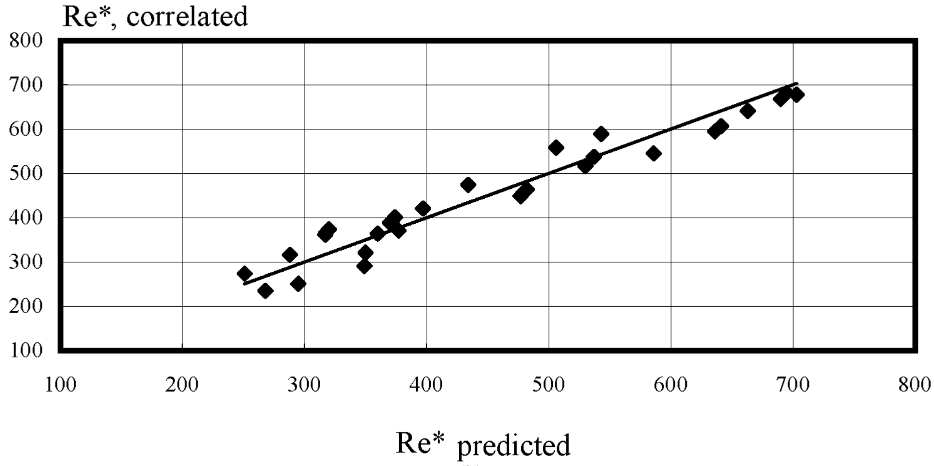


(a)

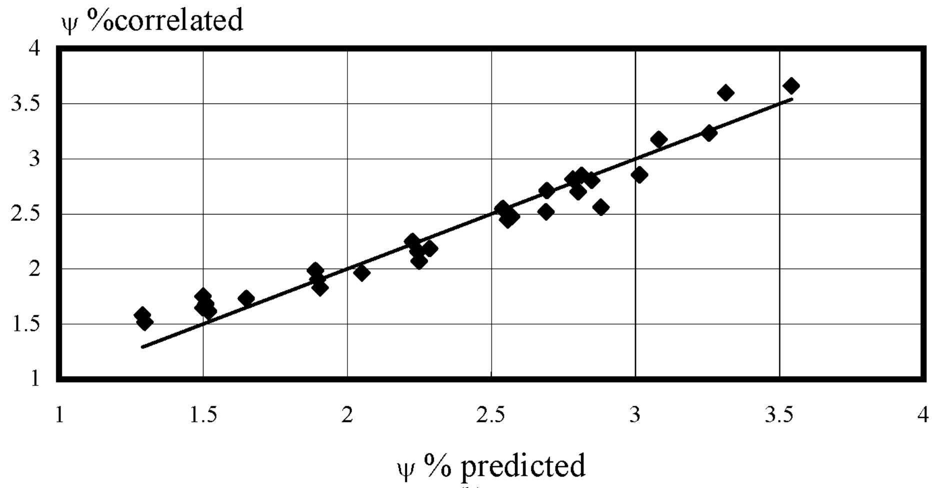


(a)

Figure 12 Validity of present correlations for mimimum percent exergy destruction rate and critical Reynolds number (a) validity of the present correlations for Ethylene Glycol flow in in-line 3-DIEST configuration (tube#1 discussed by Liao and Xin (1995)) and (b) validity of present correlations for oil flow in in-line 3-DIEST configuration (tube#1 discussed Liao and Xin (2000)) (continued)



(b)



(b)

5 Conclusions

The following conclusions can be drawn based on the exergy performance:

- it is preferred to use hollow 3-DIES tubes in water turbulent flow regime while segmented twistedtape inserts are attractive in laminar flow
- in high viscous laminar flow, hollow 3-DIES rough tubes are preferable at relatively high Reynolds numbers while it is preferred to use low P/D twistedtape inserts at a low range of Reynolds numbers

- when comparing the exergy performance of 3-DIES rough tube, inline configuration is preferred over the staggered alignments
- empirical correlations for both Re^* and $\Psi_{min}\%$ were obtained for oil and Ethylene Glycol in inline 3-DIES rough tubes as a function of the different investigated parameters.

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Nomenclature

Symbols	
b	Space length, m
c_p	Specific heat at constant p, J/kg.K
d	Inner rod diameter, m
D	Tube inner diameter, m
e	Fin height, m
h	Heat transfer coefficient, W/m^2.K
k	Thermal conductivity, W/m.K
L	Tube length, m
m	Fluid mass flow rate, kg/s
p	Flow pressure, Pa
P	Twistedtape pitch, m
Pa	Axial pitch of 3-DIES, m
Pc	Circumferential pitch of 3-DIES, m
Q	Heat transfer rate, W
q	Heat flux, W/m^2
R	Tube radius, m
S	Entropy, J/K
$\dot{S}$	Entropy rate, W/K
s	Specific entropy, J/kg.K
T	Temperature, K
U	Flow velocity, m/s

w	Wetted perimeter, m
W	3-DIES width, m
X	Axial distance, m
Superscripts	
*	Critical value, optimum
$''$	Dimensionless value
Subscripts	
0	Reference value (at 298)
e	At exit
gen	Generation
i	At inlet
m	Mean value
min	Minimum
p	Constant pressure
w	At tube wall
D	Inner diameter
Greek letters	
Δ	Difference value
δ	Twisted tape thickness, m
$\Delta\Psi$	Exergy destruction rate, W
ρ	Density, kg/m^3
μ	Dynamic viscosity, kg/m.s
Dimensionless terms	
b/D	The ratio of space length (between the tape segments) to tape diameter
F	Fanning friction factor, $F = (-\mathrm{d}P/\mathrm{d}x)D/2\rho U_m^2$
Re	Reynolds number based on the plain tube diameter, $Re = \rho UD/\mu$
SBr	Pseudo Brinkman number, $SBr = \mu U_m^2/kT$
Pr	Prandtl number, $Pr = \nu/\alpha$
P/D	The ratio of tape pitch to its diameter, L/D Tube ratio
q''	Dimensionless heat flux, $q'' = \dfrac{qR}{K_o T_o}$
σ	Entropy generation (entropy generation rate to thermal capacity ratio), $\sigma = \dot{S}_{\text{gen}}/mc_p$
τ	Temperature difference number, $\tau = \Delta T/T = (T_w - T)/T$
ξ	Low parameter, ξ = wk/mc$_p$, and for pipe flow ξ = 4/Re.Pr
χ	Length scale, $\chi = x/D$