Second Law Analysis of Viscous Flow Through Rough Tubes Subjected To Constant Heat Flux

S. A. ABDEL-MONEIM and **R. K. Ali**

Mech. Eng. Dept., Faculty of Eng. (Shoubra), Zagazig University,
108 Shoubra St. Cairo, Egypt, E-mail: Ragab_khalil2004@ yhoo.com

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 utilization 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 water, Ethylene Glycol and ISO VG46 turbine oil in rough tubes with three-dimensional internal extended surfaces (3-DIES) were studied. Also, enhanced tubes either with continues 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 twisted-tape inserts, is attractive in laminar flow. Also, in-line 3-DIES rough tubes indicate lower exergy destruction rate when compared with that of staggered alignments. The correlation for both Re^* and $\Psi_{min}\%$ were obtained for oil and Ethylene Glycol in in-line 3-DIES rough tubes as a function of the different investigated parameters.

KEYWORDS: Forced convection, three-dimensional internal extended surfaces, Exergy, tape inserts

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	twisted-tape 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

Superscripts

*	critical value, optimum

Subscripts:

0	reference value
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

S	entropy, J/K	δ	twisted tape thickness, m
$\dot{S}$	entropy rate, W/K	$\Delta\Psi$	exergy destruction rate, W
s	specific entropy, J/kg.K	ρ	density, kg/m^3
T	temperature, K		
U	flow velocity, m/s		
w	witted perimeter, m		
W	3-DIES width, m		
x	axial distance, m		

Dimensionless terms:

F Fanning friction factor, $F = (-dP/dx)D \big/ 2\rho U_m^2$

Re Reynolds number based on the plain tube diameter, Re =ρUD/μ

SBr pseudo Brinkman number, SBr= μU_m^2 / kT

χ length scale, χ =x/D Pr : Prandtl number , Pr =ν/α

σ entropy generation (entropy generation rate to thermal capacity ratio), $\sigma = \dot{S}_{gen} / mc_p$

τ temperature difference number, $\tau = \Delta T/T = (T_w - T)/T$

ξ flow parameter, $\xi = wk/mc_p$, and for pipe flow ξ=4/ Re.Pr

1. INTRODUCTION

Different methods have been carried out for enhancing the rate of heat transfer in forced convection to reduce the size of the heat exchanger and save energy [1]. Surfaces methods are used on the side of the surface that come into contact with the fluid of low heat transfer coefficient to reduce the thickness of the boundary layer and to introduce better fluid mixing. The primary mechanism for thinning the boundary layer increases he stream velocity and turbulent mixing. Secondary recirculation flows can further enhance advection flows from the wall to the core promote mixing. Flow separation and reattachment within the flow channel also contribute to the heat transfer enhancement. Some of the existing methods for enhancing heat transfer in single phase fully developed turbulent flows in round 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 of such devices have been developed and tested in the past [2,3]. Durmus [4] studied the effect of inserted turbulators on the flow and heat transfer characteristics for isothermally heated tube from the outer surface. Marner and Bergles [5] tested the effect of inserted twisted tape and internal fins on flow and heat transfer characteristics with the outer surface of the tube is subjected to isothermal heating. Saha et al.[6] 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 [7] studied experimentally the effect of presence of three-dimensional internally extended surface (3-DIES) on forced convective heat transfer and pressure drop of Ethylene Glycol in tubes.

Two or more of the existing techniques can be utilized simultaneously to produce and enhancement large then that produced an enhancement produced by one technique which is known as compound enhancement. This is an emerging are of interest and holds promise for practical application. Interactions between different enhancement methods contribute to greater values of the heat transfer coefficients compared to the sum of the corresponding values for the individual techniques used alone. Preliminary studies in compound passive enhancement technique are encouraging. Some examples are rough tube with twisted tape inserts and grooved rough tube with twisted tape inserts [8] or combination of a corrugates tubes and twisted tape inserts are studied in [9]. Also, Liao and Xin [10] studied the augmentation of convective heat transfer inside tubes with three-dimensional internal extended surfaces and twisted-tape inserts. Several criteria for evaluating the effectivness of these augmentation techniques have proposed. Web [11,12] and Abdel-Moneim et al. [13], have used the efficiency index (Stanton number ratio to Fanning friction factor ratio) as a performance evaluation criterion (PEC) to evaluate the performance benefits for varieties of augmentation techniques. The method proposed by Bergles et al.[14] compares the performance of augmented surface heat exchanger to meet a defined objectives, such as maximizing heat load or reducing surface area. These criteria do not take into considerations the thermodynamic impact of heat transfer augmentation techniques. So, the conservation of exergy (the part can b transformed into work) during the augmentation heat transfer process is not studied.

Heat transfer processes are accompanied by thermodynamic impact, irreversibility or 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 optimize energy conversion systems it is essential to understand how entropy is being generated in the convective heat transfer processes to avoid exergy destruction. Bejan [15-18] and San et al. [19] analyzed the entropy generation based on the aspects of 2^{nd} law of thermodynamics for a convective heat transfer process in different fundamental flow configurations. Sahin [20] 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 [16], Sahin [20] found that the entropy generation increases along the duct length. In addition, the entropy generation number following Sahin [20] is an important parameter in deciding the true merit of a heat transfer augmentation technique. Bejan and Pfistr[21] and Abdel-Moneim [22] proposed an evaluation of heat transfer augmentation techniques based on their impact on entropy generation. In this sense, the effective technique is that one leads to a reduction in the entropy generation rate and in accordance reduces the exergy destruction rate. Prasad and Shen [23,24] 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 minimizing the net exergy destruction rate. Abdel-Moneim and Ali [25] analyzed the entropy generation based on the 2^{nd} law aspects for a convective heat transfer process in different fundamental flow configurations.

Nag and Mukherjee [26] modified Bejan's entropy generation method by including the effect of the variation in the fluid temperature along a heat transfer duct with constant wall temperature. Also, Zimparov [27] applied this modified performance evaluation criterion to 10 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 tubes.

In the present work an evaluation method based on exergy analysis is proposed using the principles of the first and second lows of thermodynamics. The exergy destruction is estimated in non-dimensional form. The proposed evaluation technique has applied on rough tubes containing three dimensional internal extended surfaces (3-DIES) for flows with different fluids. Continuous and segmented tape inserts inside rough tubs were also evaluated as augmentation techniques.

2. THEORETICAL MODELING

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 Fig.1a is,

$$\delta Q = m\, c_p\, dT \tag{1}$$

And the heat transfer equation is,

$$\delta Q = q''\,(w\, dx) = h\, \Delta T w\, dx \tag{2}$$

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

$$dT = \frac{hw\,\Delta T}{mc_p} dx \tag{3}$$

Assuming a steady incompressible flow and applying the entropy production theory (based on the 2nd law of thermodynamics, Van Wylen et al. [28]), the net entropy generation rate can be calculated as:

$$d\dot{S}_{gen} = mds - \frac{\delta Q}{T_w} \tag{4}$$

$$Tds = c_p dT - \frac{dP}{\rho} \tag{5}$$

Substituting the values of ds from Eq.5 and δQ from Eq.1 into Eq.4, yields,

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

Rearranging the right-hand side Eq.6 becomes,

$$d\dot{S}_{gen} = \left(\frac{\tau^2 hw}{\tau + 1}\right) dx - \frac{m}{\rho T}\left(\frac{dP}{dx}\right) dx \tag{7}$$

Where, the 1st term of the right-hand side of Eq.7 represents the entropy generation rate due to heat transfer across a finite temperature difference while the 2nd term represents the contribution of the flow friction in the entropy production. Extensive details of this model were presented in Abdel-Moneim and Ali [25]. On substituting the following dimensionless parameters, Eq.7 can be transformed into dimensionless form as:

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

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

$$\dot{S}_{gen} = \sigma\, m\, c_p \qquad \text{Eq.9}$$

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

$$\Delta\Psi = T_0 \dot{S}_{gen} \qquad \text{Eq.10}$$

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

3. RESULTS AND DISCUSSIONS

Rough tubes with 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 flow. 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 in [7 &10] with different configurations shown in Figs. 1b, 1c and 1d was evaluated using the present performance evaluation criterion for ranges of laminar and turbulent flows. 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 discussed in [6] and named tube#2 (inner diameter=12 mm and 1.84 m long) with twisted and straight tape inserts shown in Fig.1e is evaluated using the present PEC. Also, the effect of using segmented tape on the exregy destruction rate was presented.

3.1 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 with that of Abdel-Moneim [22] for the case of water flow in a smooth tube#1 subjected to uniform heat fluxes as shown in Fig. 2. It was found that, discrepancies between the present data and that of Abdel-Moneim [22] are less than 7 % over a range of Reynolds number from 5000 to 50000. The percent exergy destruction rate , Ψ % , for enhanced tubes initially decreases with the increase in Reynolds number as shown in Fig. 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, the flow friction becomes dominant and the exergy destruction increases. The combined effect results in a thermodynamic optimum, Re^*, with a minimum exergy destruction rate, Ψ_{min} %. This value depends on the heat flux and increases with increasing heating load. Flow in 3-DIES rough tubes almost sustain exergy destruction rate lower than that for rough tubes with twisted tape inserts as shown in Fig.2. This is due to the small enhancement effect of twisted tape inserts on the heat transfer accompanied with extremely high friction as discussed in Ventsislav Zimparov [10]. Smooth tubes with straight and twisted tape inserts were investigated for water flow in laminar regime and the results are shown in Figs.2& 3. The exergy destruction rate is predicted utilizing empirical correlations for both the heat transfer and the flow friction from Saha et al. [6]. It is observed that the rate of exergy destruction rate is affected by both flow velocity and the tape pitch as shown in Fig. 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 the space between segments as shown in Fig. 4.

3.2 Exergy Destruction Rate for Oil Flow

The exergy destruction rate is predicted using the experimental data from Ventsislav Zimparov [10] for heat transfer and flow friction for flow in 3-DIES rough tubes with different configurations. As can be seen from Fig. 5 twisted-tape with small P/D has the lowest rate of exergy destruction. This is due to the augmentation of heat transfer which is dominant at low Reynolds number. However, at high Reynolds number hollow 3-DIES rough tubes indicates the lowest rate of exergy destruction and higher critical Reynolds number was obtained. From Fig. 5, it can be concluded that hollow 3-DIES rough tubes are attractive especially for laminar flow at relatively high Reynolds number. This is due to its lower rate of exergy destruction with higher critical Reynolds number that results in a higher Nusselt and consequently lower surface area. The dimensionless entropy generation and exergy destruction rates for oil laminar flow in different configurations of hollow 3-DIES rough tubes are illustrated in Fig. 6. It is clear that, in-line 3DIES rough tubes (Configuration 4), indicates lower entropy generation and exergy destruction rate. This may be attributed to the relative enhancement in heat transfer and increase in the flow friction. 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 number while its effect exceeds that of hollow 3DIES rough tubes with increasing Reynolds number. This may be attributed to the extremely increase in the flow friction with increasing Reynolds number.

3.3 Exergy Destruction Rate for Ethylene Glycol Flow

Figures 8 to 11 show the exergy analysis for Ethylene Glycol flow in augmented rough tubes over $100 < Re < 5000$ utilizing the experimental data for heat transfer and flow friction from Liao and Xin [7] and Ventsislav Zimparov [10]. It is found that in-line 3-DIES rough tubes (configuration D), has lower exergy destruction compared with all staggered configurations. This is according to the relative thermal performance for in-line and staggered alignments. The effect of heat flux and inlet fluid temperature is shown in Figs. 9 &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. 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 decrease the rate of the heat transfer and consequently increases temperature difference between the fluid and the tube surface. Figure 11 shows Ψ versus Re for Ethylene Glycol flow in 3-DIES rough tubes with twisted tape inserts at different values of P/D. A thermodynamic optimum was found at earlier Reynolds number as a result of high viscosity of Ethylene Glycol. The value of critical Reynolds number depends on the twisted-tape pitch ratio, P/D, in addition to the heat flux as shown in Fig. 11.

On view of the present study, exergy optimization becomes more significant in the new designs and even in retrofitting of the existed heat transfer equipment. Also, empirical correlations that can determine the 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 in-line 3-DIES rough tubes as follows:

i) For Ethylene Glycol Flow:

$$Re^* = 451.41q^{05185} (L/D)^{0.03729} Pr^{-0.6363} \quad (11)$$

$$\psi_{min}\% = 3.3971 Re^{*-0.3781} \exp(-0.000387 Re^*) \quad (12)$$

Figure12-a shows that the correlations, Eqs.(11, 12), are valid with maximum deviations of $\pm$20% in estimating Re* and Ψ_{min}% within the ranges of the different parameters as :
$1.0 \le q^{"} \le 14$ kW/m^2, $37 \le L/D \le 222$ and $58 \le Pr \le 108$.

ii) For oil flow :

$$Re^* = \exp(0.000141q + 0.0001166(L/D) - 0.003757 Pr + 6.74) \quad (13)$$

$$\psi_{min}\% = \exp((0.0001246q - 0.0005584(L/D) - 0.00011977 Pr + 0.389) \quad (14)$$

These correlations, Eqs. (13, 14), are valid within $\pm$22% maximum deviations in both Re^* and Ψ_{min}% for $1.0 \le q \le 8$ kW/m^2, $74 \le L/D \le 222$ and $293 \le Pr \le 535$.

4. CONCLUSIONS

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

1) It is preferred to use hollow 3-DIES tubes in water turbulent flow regime while segmented twisted-tape inserts are attractive in laminar flow.
2) In high viscous laminar flow, hollow 3-DIES rough tubes are preferable at relatively high Reynolds number while it is preferred to use low P/D twisted-tape inserts at low range of Reynolds number.
3) When comparing the exergy performance of 3-DIES rough tube, in-line configuration is preferred over the staggered alignments.
4) Empirical correlations for both Re_{cr} and Ψ_{min}% were obtained for oil and Ethylene Glycol in in-line 3-DIES rough tubes as a function of the different investigated parameters.

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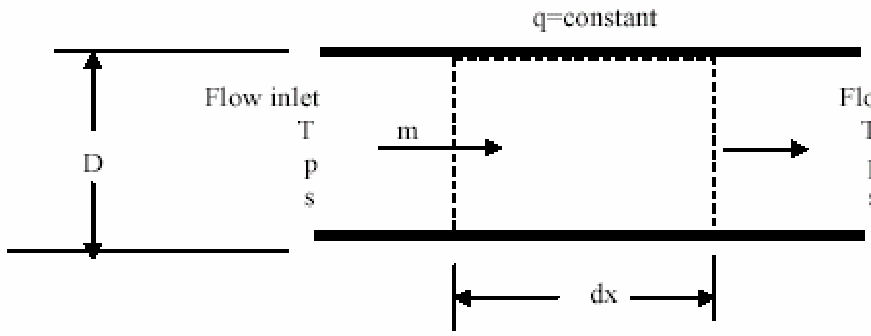
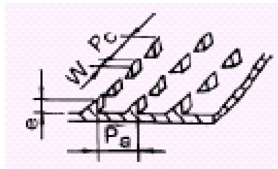


a) Control volume in a tube subjected to a uniform heat flux

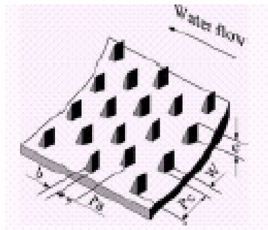
Configuration	e	Pa	W	Pc	Fin aligment
A	0.54	3.78	0.53	2.12	Staggered
B	0.54	4.32	0.61	2.42	Staggered
C	0.95	6.62	0.93	3.7	Staggered
D	0.54	3.78	0.53	2.12	In-Line
E	0.34	2.36	0.33	1.31	Staggered
F	0.74	5.2	0.73	2.91	Staggered
G	0.74	5.2	0.73	4	Staggered



All dimensions in mm

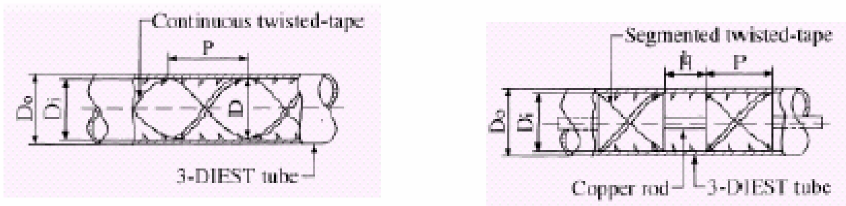
b) Structural geometries of the 3-DIES tube#1 in [7]

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

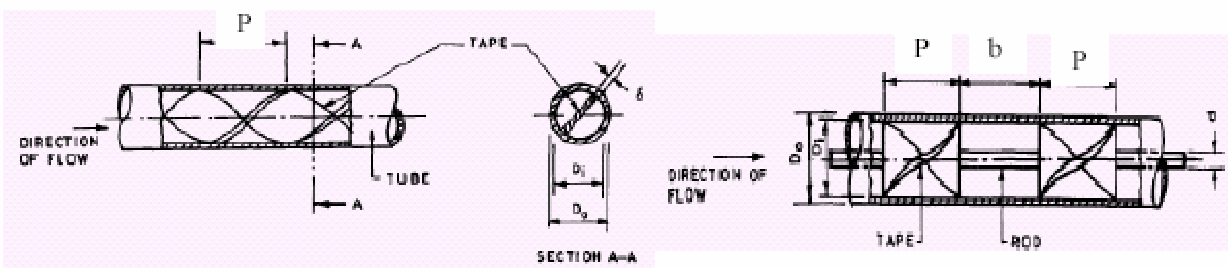


All dimensions in mm

c) Structural geometries of the 3-DIES tube#1 [10]



d) 3-DIEST tube#1 with and continuous segmented tape [10]



e) Layout of a full-length and segmented twisted-tape inserts inside a circular tube#2 [6]

Fig.1 The control volume and the nomenclatures of the studied enhancement methods

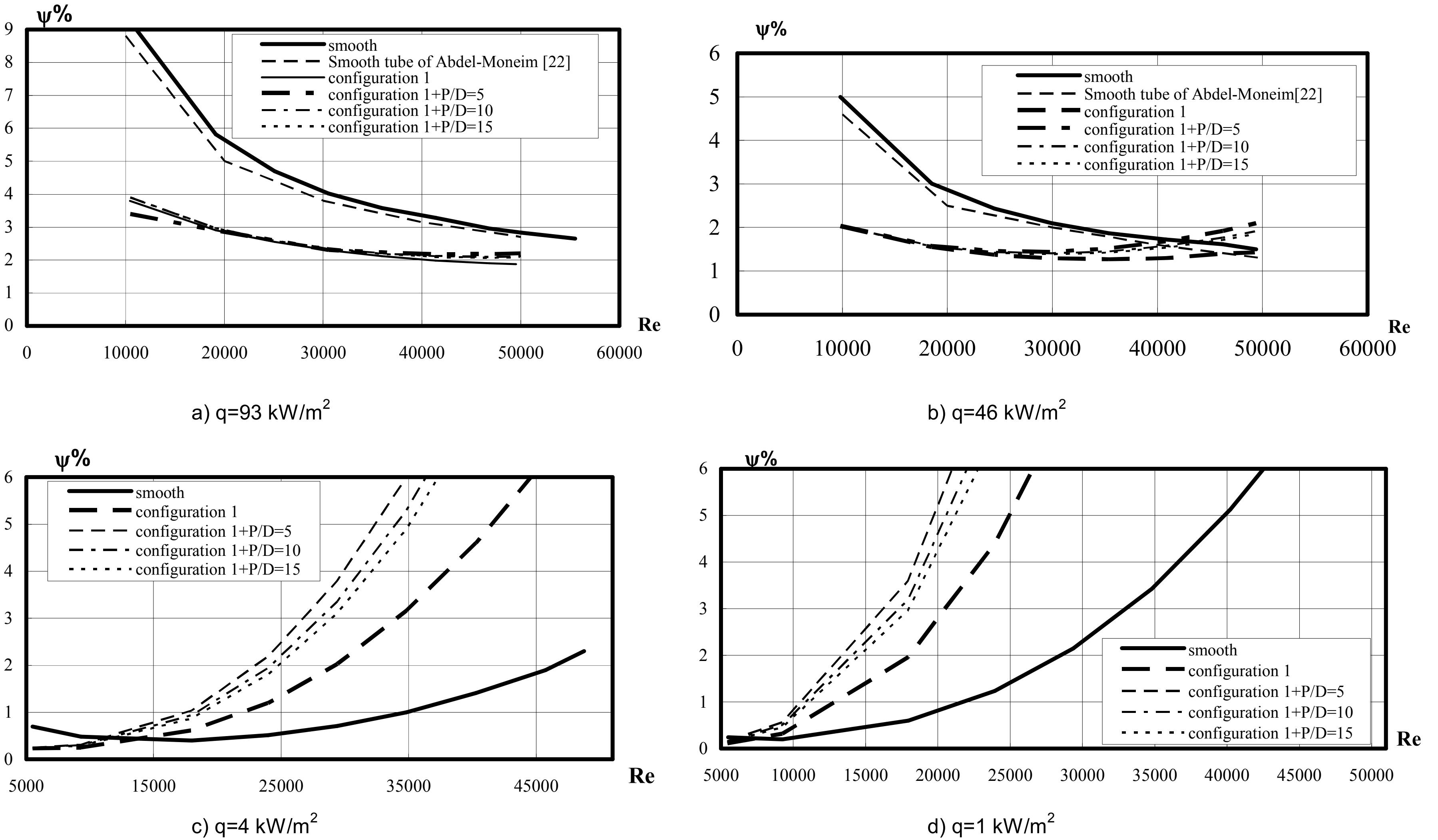


Fig. 2 Percent exergy destruction rate for water flow in tube #1 discussed in [10] with twisted tape inserts with different pitches at different heat fluxes

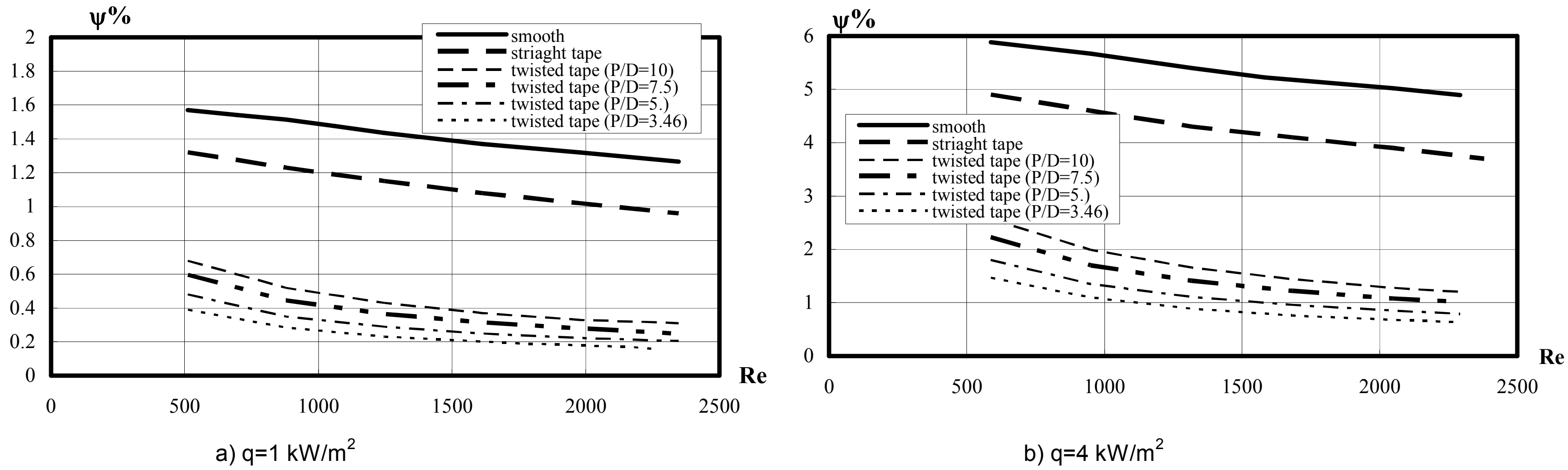


Fig. 3 Exergy destruction rate for laminar water flow in tube#2 discussed in [6] with twisted tape inserts with different pitches at different heat fluxes.

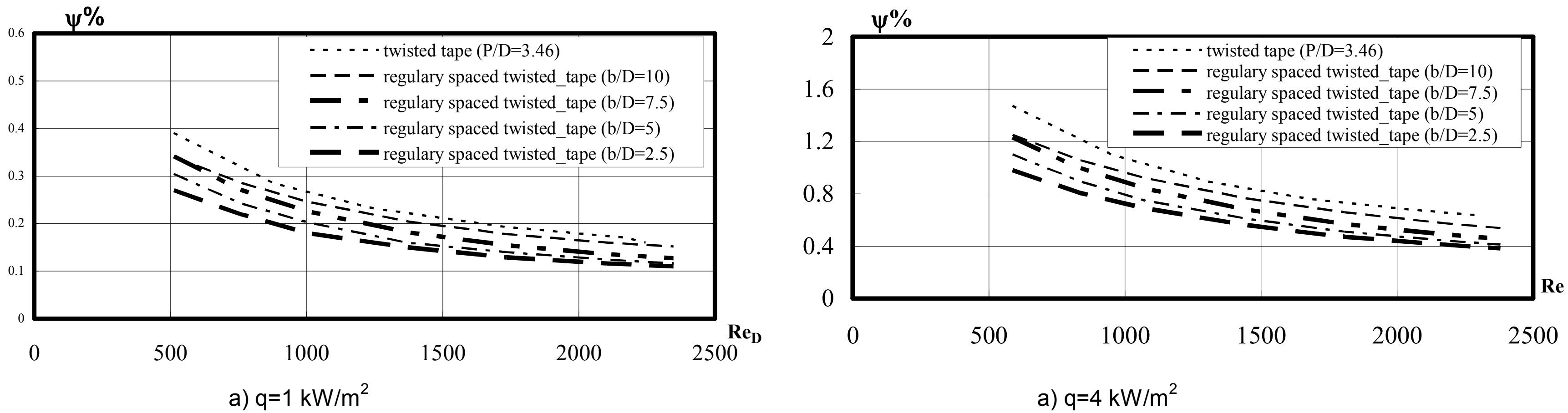
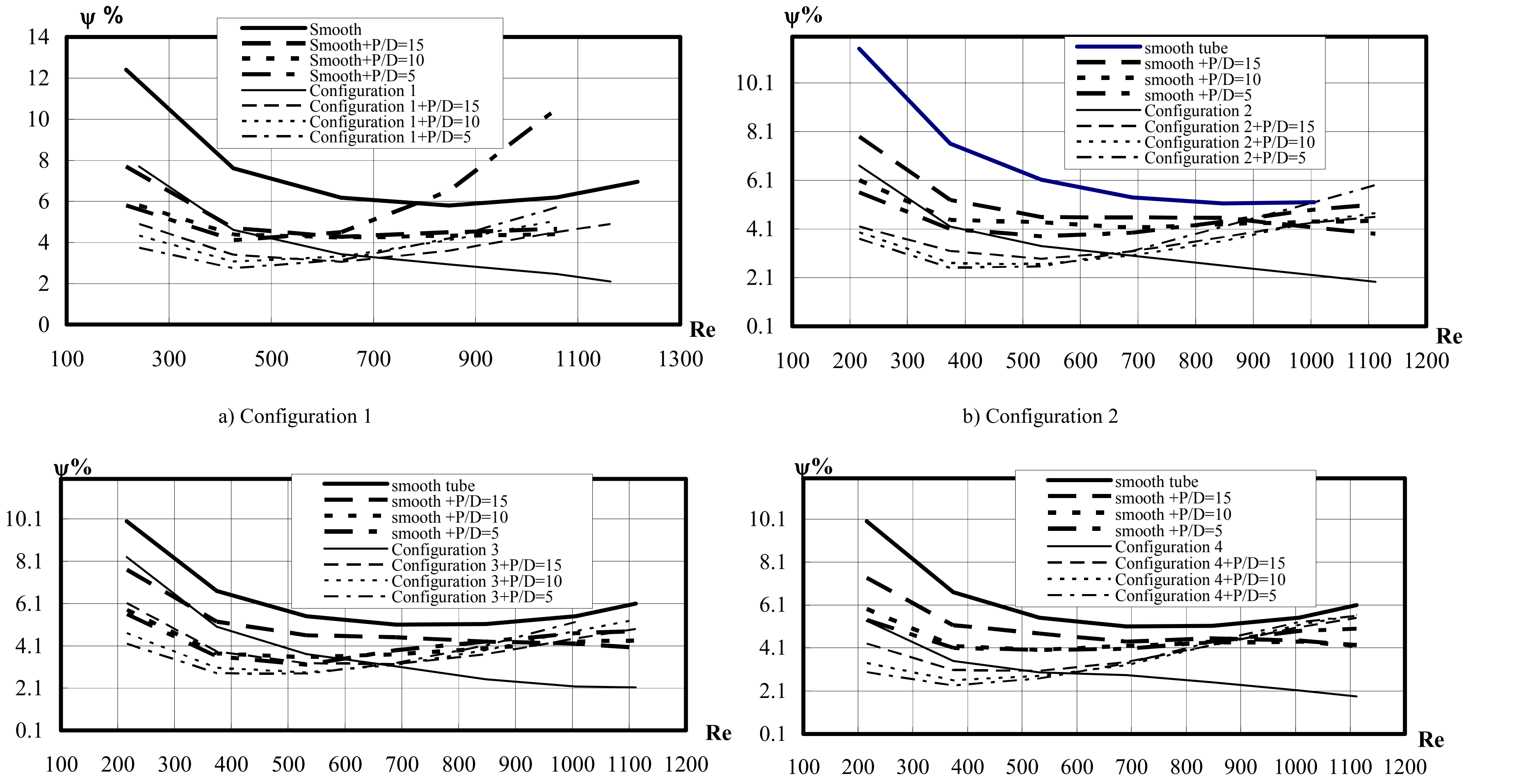


Fig. 4 Exergy destruction rate for laminar water flow in tube #2 discussed in [6] with regulary spaced twisted tape inserts at P/D=3.46 and at different heat fluxes.



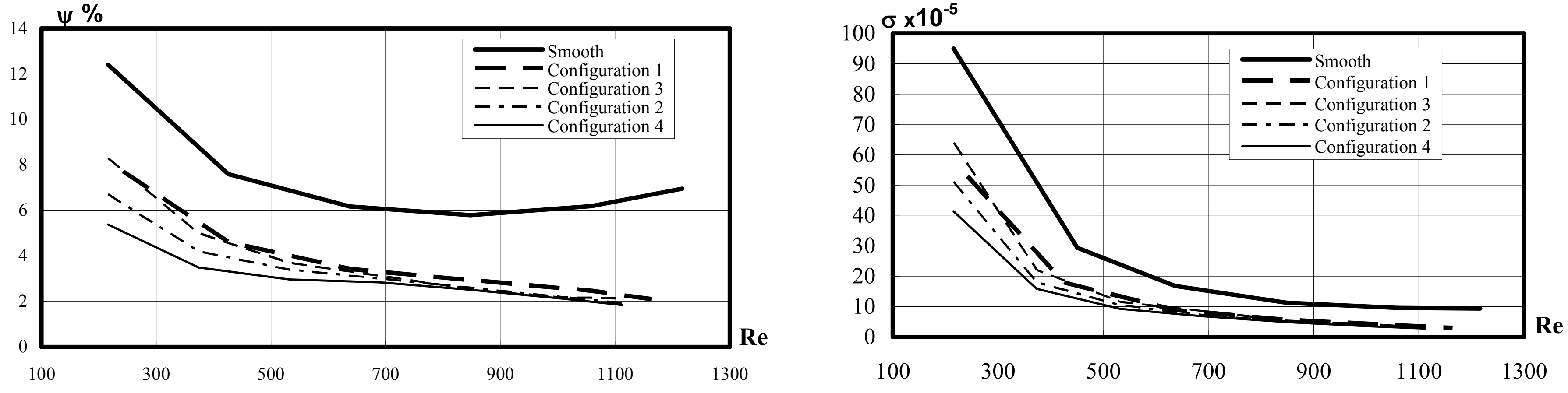
a) Configuration 1

b) Configuration 2

c) Configuration 3

d) Configuration 4

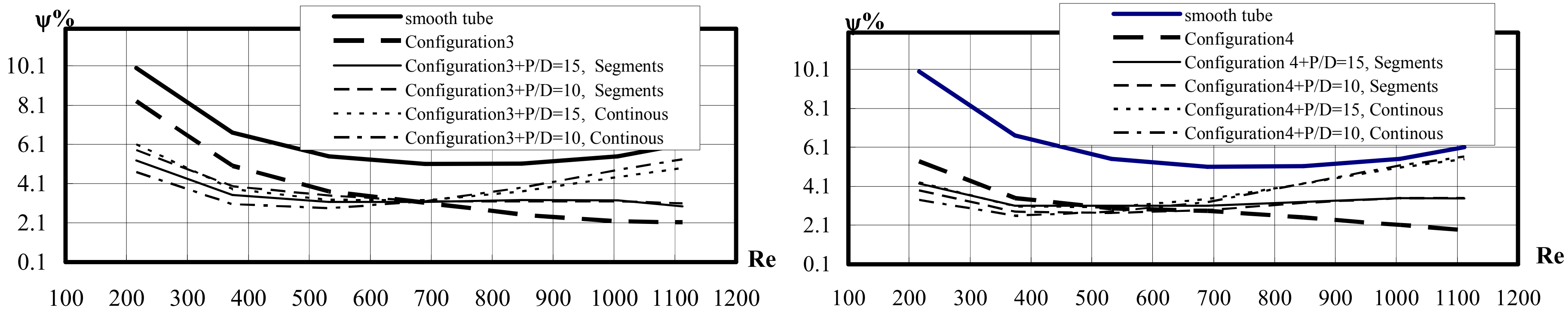
Fig. 5: Percent exergy destruction rate for laminar oil flow in smooth and rough tubes (with different configurations of three-dimensional internal extended surfaces) discussed in [10] with twisted tape inserts.



a) Percent exergy destruction rate

b) Dimensionless entropy generation

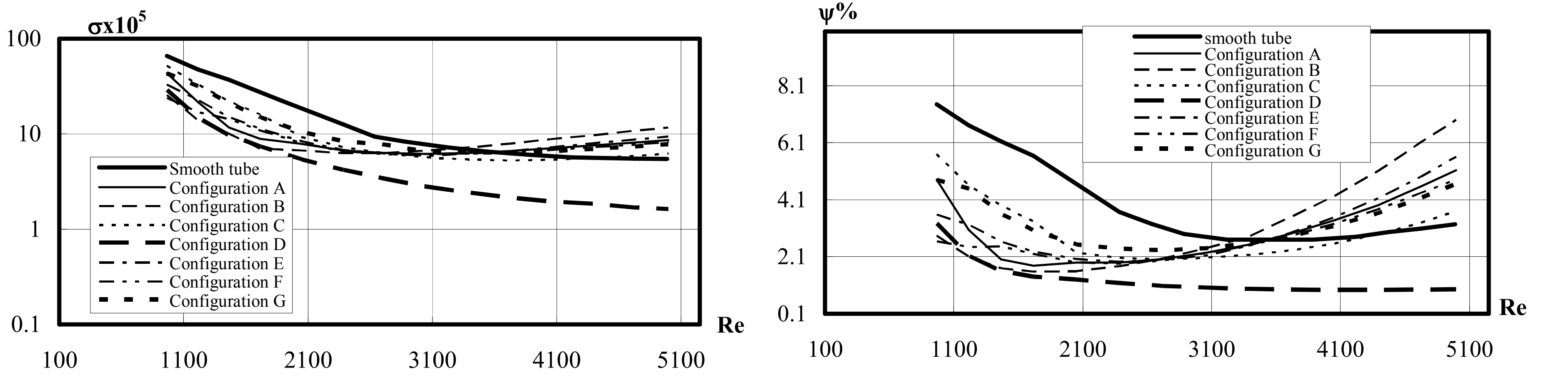
Fig. 6: Dimensionless entropy generation and percent exergy destruction rate for laminar oil flow in tube #1 discussed in [10] with different configurations of three-dimensional internal extended surfaces.



a) Configuration 3

b) Configuration 4

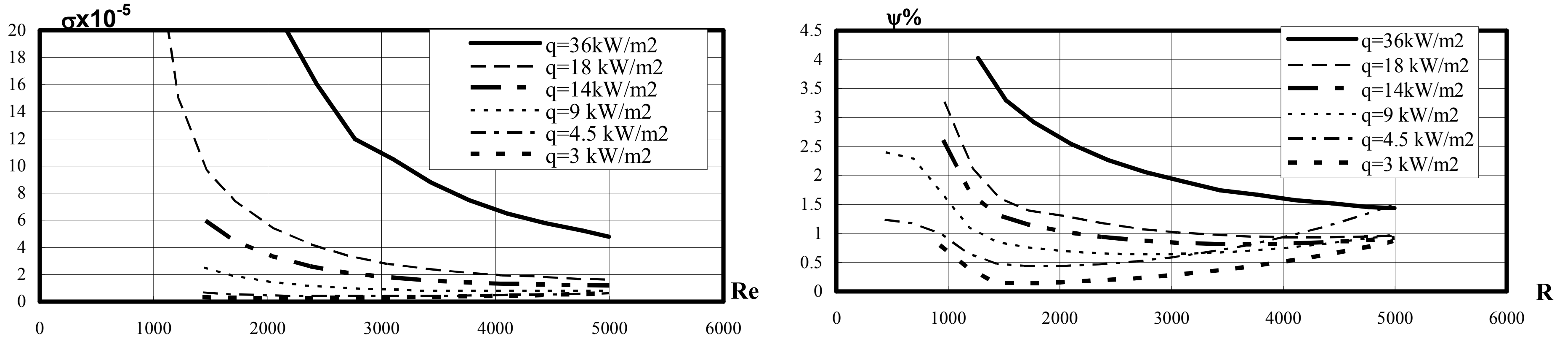
Fig. 7: Percent exergy destruction rate for laminar oil flow in tube#1 (with different configurations of three-dimensional internal extended surfaces) discussed in [10] with continous and segment twisted tape



a) Dimensionless entropy generation

b) percent exergy destruction rate

Fig. 8 : Dimensionless entropy generation and percent exergy destruction rate for Ethylene Glycol flow in tube#1 discussed in [7] with with different configurations of three-dimensional internal extended surfaces ($q=18$ kW/m^2, Ti=310 K)



a) Dimensionless entropy generation

b) percent exergy destruction rate

Fig. 9 :Dimensionless entropy generation and percent exergy destruction rate for Ethylene Glycol flow in tube#1 discussed in [7] for different heat fluxes (Ti=310 K)

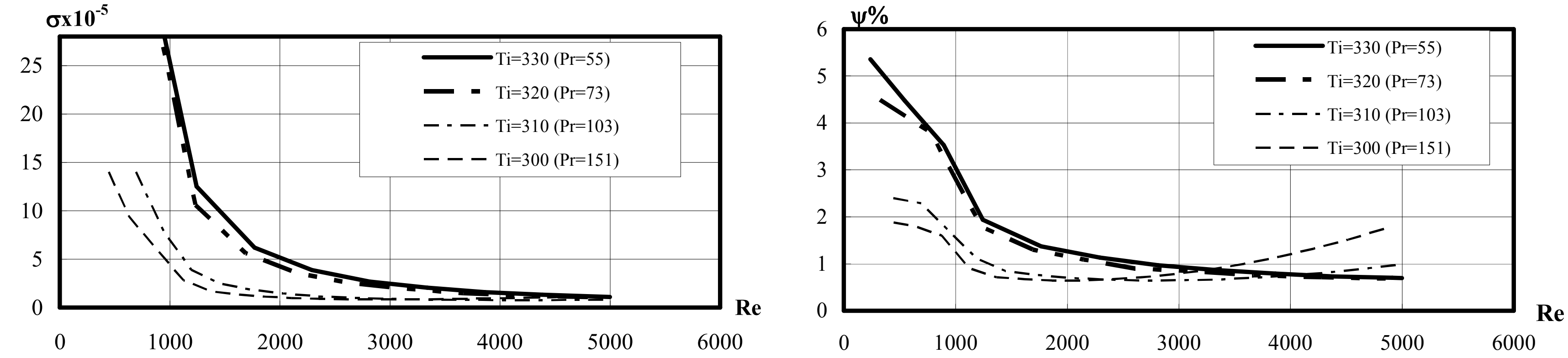


Fig. 10 :Dimensionless entropy generation and percent exergy destruction rate for Ethylene Glycol flow in tube#1 (Configuration D) discussed in [7] for different Inlet temperatures at q=9 kW/m^2

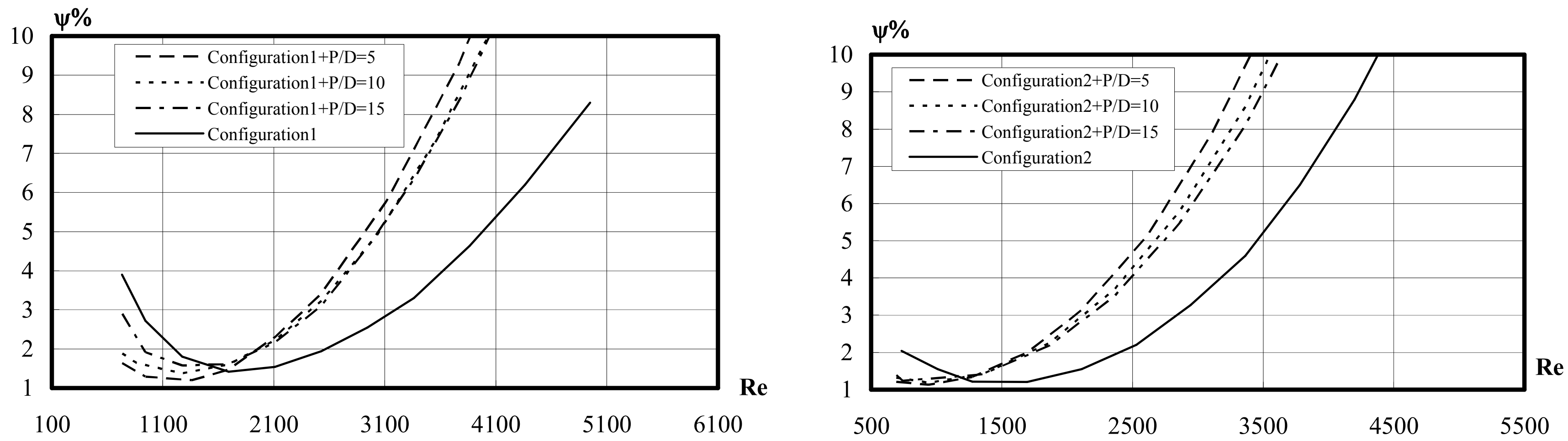
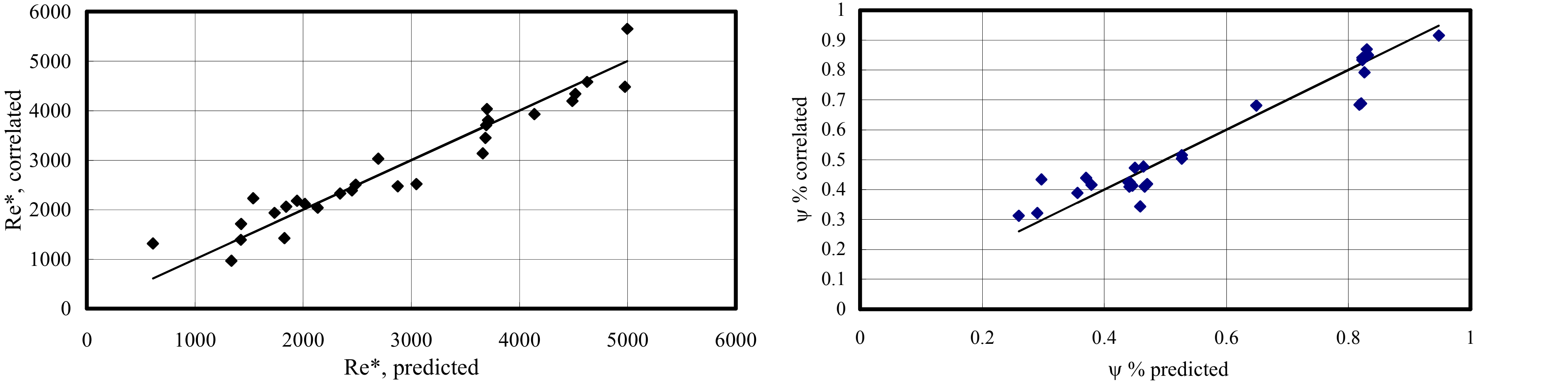
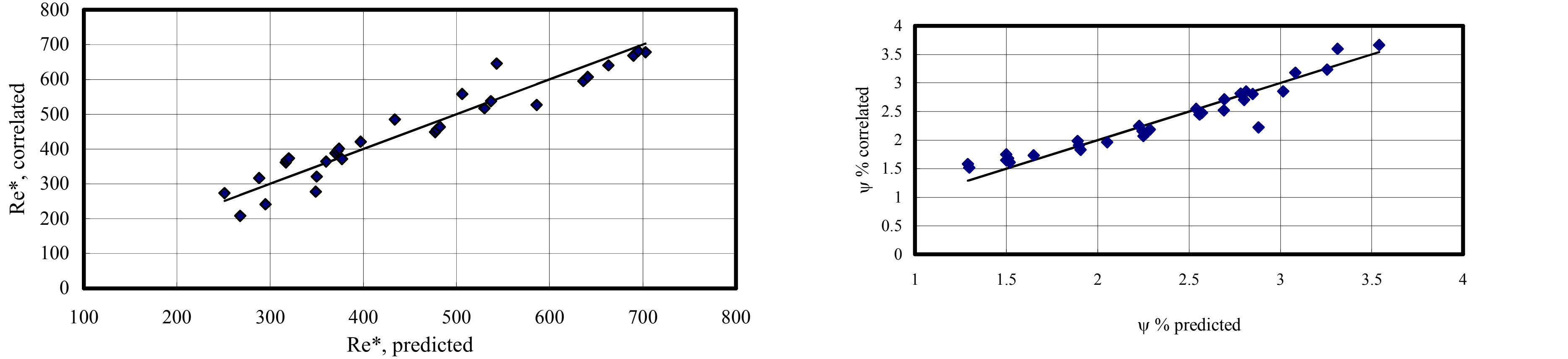


Fig. 11: Percent exergy destruction rate for Ethylene Glycol flow in tube#1 discussed in [10] with twisted tape inserts (q=9 kW/m2 anf Ti=310 K)



a) Validity of the present correlations for Ethylene Glycol flow in in-line 3-DIEST configuration (tube#1 discussed in [7])



b) Validity of present correlations for oil flow in in-line 3-DIEST configuration (tube#1 discussed in [10])

Fig.12 :Validity of present correlations for mimimum percent exergy destruction rate and critical Reynolds number