

## **INVESTIGATION OF HEAT TRANSFER AUGMENTATION IN A TUBE WITH DIFFERENT MODIFIED TWISTED TAPE INSERTS UNDER THE SAME CONDITIONS**

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### **Abstract**

*The swirl flow devices like twisted tape, winglet, groove, conical ring, wire coil etc. have been used as passive heat transfer augmentation technique. Especially, twisted tape devices have been applied in thermal engineering systems such as boilers, heat exchangers, water heater. Many researches using swirl flow devices particularly twisted tape with different geometries for heat transfer enhancement have been published. In this paper, different modified twisted tapes which had examined in previous studies were compared with each other with regard to heat transfer augmentation. Heat transfer, friction factor and thermal-hydraulic performance factor characteristics were investigated separately under the same conditions such as same twisted ratios and same Reynolds number values. Twisted tape which gives best heat transfer and friction factor results was determined. Derived empirical correlations which had been found by researchers were used for comparison between twisted tapes.*

## **AYNI ŞARTLAR ALTINDA BORU İÇİNDE FARKLI TİP BURGU ELEMENLARIN ISI TRANSFER İYİLEŞTİRMESİNİN İNCELENMESİ**

### **Özetçe**

*Burgu eleman, kanatçık, oluk, konik halka, tel sargı vb. gibi girdap akış cihazları pasif ısı transferi iyileştirme teknikleri olarak kullanılmaktadır. Özellikle, burgu elemanlar, kazan, ısı değiştiricisi, su ısıtıcı gibi ısı mühendislik sistemlerinde uygulanmaktadır. Isı transferini iyileştirmek için farklı geometrilerdeki burgu elemanlar kullanılan birçok araştırma yayınlanmıştır. Bu çalışma kapsamında daha önce yapılan çalışmalarda kullanılan farklı tasarlanan burgu elemanlar, ısı transfer iyileştirmesi bakımından birbirleri ile karşılaştırılmıştır. Benzer burulma oranları ve benzer Reynolds sayıları gibi aynı koşullar altında ısı transferi, sürtünme faktörü ve ısıl hidrolik performans değerleri*

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*ayrı ayrı incelenmiştir. En iyi ısı transfer ve sürtünme faktörü değerlerini veren burgu eleman belirlenmiştir. Araştırmacılar tarafından bulunmuş türetilen denklemler burgu elemanların kıyaslanması için kullanılmıştır.*

**Keywords:** *Heat transfer augmentation, twisted tape*

**Anahtar Kelimeler:** *Isı transfer iyileştirilmesi, burgu eleman*

### **1. INTRODUCTION**

Heat exchangers emerge as the most important elements in energy-intensive systems. The main objective of the energy sector is to make more work with less energy consumption. For this purpose, heat transfer enhancement techniques are being developed in order to increase the efficiency of heat transfer and use of energy. Heat transfer enhancement techniques can be classified into two groups; active techniques which need external power source and passive techniques that do not use external power supply. Both active and passive techniques have been exercised to increase the heat transfer in several ways. Heat transfer enhancement applications can be done by methods such as rotating the heat transfer surface, placing turbulators into the pipe to create a turbulent flow, applying vibration to the heat transfer fluids, creating electrostatic fields and rough surfaces, using fins and elements in different geometries placed into the pipe, applying low frequency vibration to the heat transfer surface [1,2]. Twisted tape swirl generators are most favorable passive heat transfer enhancement devices.

The twisted tape used for heat transfer enhancement generates turbulent flow and increases surface area. The presence of turbulence reduces thermal boundary layer thickness between the surface of the pipe and fluid. Because of that the convective heat transfer increases. The twisted tape performance depending on various parameters such as turbulator length, hydraulic diameter of pipe, the mass flow rate, physical properties of the fluid varies according to the different geometries of turbulators.

Many researches using swirl flow devices particularly twisted tape with different geometries for heat transfer enhancement have been published. In this paper , different modified twisted tapes which had

examined in previous studies were compared with each other with regard to heat transfer augmentation. Heat transfer, friction factor and thermal-hydraulic performance factor characteristics were investigated separately under the same conditions such as same twisted ratios and same Reynolds number values.

## 2. THEORETICAL ANALYSIS

The experimental data were reduced by using the following summarized procedure. The energy obtained from hot exhaust gases passing through the inner pipe is equal to the amount of heat transferred to water flowing between two pipes.

$$\dot{m}_{exhaust} C_{p_{exhaust}} (T_{exhaust_o} - T_{exhaust_i}) = hA(T_{wall_{ave}} - T_{exhaust_{ave}}) \quad (1)$$

where  $h$  is the average heat transfer coefficient,  $A$  is the total surface area of the tube wall,  $T_{wall_{ave}}$  is the average temperature of the pipe wall and  $T_{exhaust_{ave}}$  is the average temperature of exhaust gas and they can be expressed as, respectively,

$$T_{wall_{ave}} = \frac{\sum T_{wall}}{N} \quad (2)$$

$$T_{exhaust_{ave}} = \frac{T_{exhaust_{out}} + T_{exhaust_{in}}}{2} \quad (3)$$

Average temperature of the pipe wall is an arithmetic means of the temperature values of the five measuring points at the outer wall surface of the inner pipe. The mean Nusselt number is defined by equation (4).

$$Nu = \frac{hD_h}{k} \quad (4)$$

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where  $D_h$  is the hydraulic diameter for annulus,  $k$  is the thermal conductivity of the fluid. Thermal conductivity is calculated from the fluid properties at the average fluid temperature. In addition, the Reynolds number and friction factor can be found. Friction factor ( $f$ ) can be calculated using different parameters like average velocity,  $u$ , pressure loss,  $\Delta P$ , pipe length,  $L$ , and density at the mean bulk temperature,  $\rho$ .

$$f = \frac{\Delta P}{\rho \frac{u^2 L}{2 D}} \quad (5)$$

The type of the flow can be defined from the Reynolds number which is expressed as:

$$Re = \frac{\rho u_m D_h}{\mu} \quad (6)$$

where  $\mu$  is the dynamic viscosity and  $u_m$  is the average fluid velocity along the pipe cross-section.

Thermal hydraulic performance defined as the ratio of the enhancement in heat transfer to the changes of pressure differences is written as,

$$\eta = \frac{Nu_t}{\left( \frac{f_t}{f_e} \right)^{1/3}} \quad (7)$$

where  $Nu_t$  and  $f_t$  are Nusselt number and friction factor in the tube with twisted tape and  $Nu_e$  and  $f_e$  are Nusselt number and friction factor in the empty tube, respectively.

### 3. PREVIOUS STUDIES

Many researches using swirl flow devices particularly twisted tape with different geometries for heat transfer enhancement have been published. Helically twisted tapes (HTTs) were examined in an attempt to heat transfer enhancement by Eiamsa-ard et al [3]. The experiments were performed using HTTs with three twist ratios ( $y/W$ ) of 2, 2.5 and 3, three helical pitch ratios ( $p/D$ ) of 1, 1.5 and 2 for Reynolds number between 6000 and 20000.

The conventional helical tape (CHT) was also tested for comparison. The obtained results reveal that at similar condition, HTTs give lower Nusselt number and friction but higher thermal performance factor than CHTs. Heat transfer rate and friction factor increase as the tape twist ratio and helical pitch ratio decrease, while the thermal performance shows opposite trend.

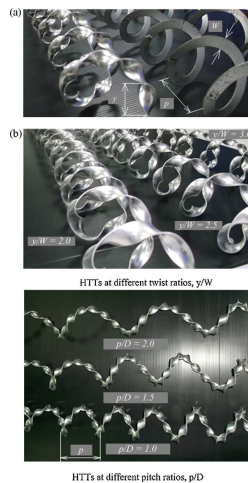


Figure 1. (a) geometries of HTT and CHT and (b) geometric details of helically twisted tapes.

The highest thermal performance factor of 1.29 was achieved by utilizing the tape with the largest twist ratio ( $y/W=3$ ) and helical pitch ratio ( $p/D=2$ ) at Reynolds number of 6000. The geometries of helically twisted

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tapes (HTTs) together with conventional helical screw tapes (CHTs) are shown in Fig. 1a and b. The experimental results of Nusselt number (Nu), friction factor (f) and thermal performance factor ( $\eta$ ) were fitted using least square regression analysis and empirical correlations were derived.

$$Nu = 0.053 Re^{0.796} Pr^{0.4} \left(\frac{y}{W}\right)^{-0.127} \left(\frac{P}{D}\right)^{-0.188} \quad (8)$$

$$f = 12.653 Re^{-0.295} \left(\frac{y}{W}\right)^{-0.652} \left(\frac{P}{D}\right)^{-1.513} \quad (9)$$

$$\eta = 3.377 Re^{-0.148} \left(\frac{y}{W}\right)^{-0.091} \left(\frac{P}{D}\right)^{0.317} \quad (10)$$

Eiamsa-ard et al [4] also reported the influences of twin-counter / co-twisted tapes on heat transfer rate, friction factor and thermal enhancement index. The tests were conducted using the twin counter twisted tape (CTs) and the twin co-twisted tapes (CoTs) with four different twist ratios ( $y/w = 2.5, 3.0, 3.5, 4.0$ ) for Reynolds numbers range between 3700 and 21000 under uniform heat flux conditions. The results showed that the CTs are more efficient than the CoTs for heat transfer enhancement. Heat transfer rates in the tube fitted with the CTs are around 12.5–44.5% and 17.8–50% higher than those with the CoTs and ST, respectively. The maximum thermal enhancement ( $\eta$ ) obtained at the constant pumping power by the CTs with  $y/w = 2.5, 3.0, 3.5$  and  $4.0$ , are 1.39, 1.24, 1.12 and 1.03, respectively. The twin counter twisted tape (CTs) and the twin co-twisted tapes (CoTs) were shown in Fig. 2.

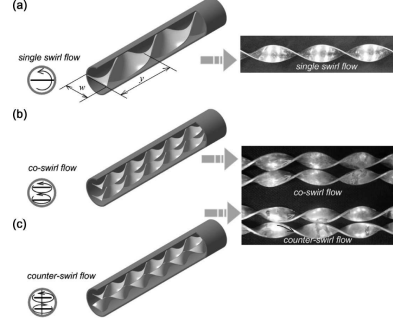


Figure 2. Test tube with twisted tape inserts: (a) single twisted tape (ST), (b) twin co-twisted tapes (CoTs) and (c) twin counter twisted tapes (CTs).

The derived empirical correlations from the experimental results of the empty tube fitted with the CoTs and CTs can be writing in term of twist ratio ( $y/w$ ), Reynolds number ( $Re$ ) and Prandtl number ( $Pr$ ) as follows:

The tube fitted with the CTs (counter-swirl flow generators):

$$Nu = 0.473Re^{0.66} Pr^{0.4} \left( \frac{y}{W} \right)^{-0.9} \quad (11)$$

$$f = 72.29Re^{-0.53} \left( \frac{y}{W} \right)^{-1.01} \quad (12)$$

$$\eta = 2.8Re^{-0.016} \left( \frac{y}{W} \right)^{-0.624} \quad (13)$$

The tube fitted with the CoTs (co-swirl flow generators):

$$Nu = 0.264Re^{0.66} Pr^{0.4} \left( \frac{y}{W} \right)^{-0.61} \quad (14)$$

$$f = 41.7Re^{-0.52} \left( \frac{y}{W} \right)^{-0.84} \quad (15)$$

$$\eta = 1.82Re^{-0.0186} \left( \frac{y}{W} \right)^{-0.38} \quad (16)$$

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Gunes et al. [5] experimentally studied the heat transfer and pressure drop in a tube with coiled wire inserts (Fig. 3) placed separately from the tube wall in turbulent regime. The experiments were performed three different pitch ratios ( $P/D = 1, 2, 3$ ) and two different distances ( $s = 1\text{mm}, 2\text{mm}$ ) for Reynolds numbers range between 4105 and 26400 by authors. The Nusselt number and friction factor increase with decreasing pitch ratio ( $P/D$ ) and distance( $s$ ) for coiled wire inserts. The highest overall enhancement efficiency of 50% was achieved for the coiled wire with  $P/D = 1$  and  $s = 1\text{ mm}$  at Reynolds number of 4220.

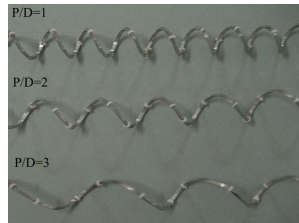


Figure 3. The coiled wire inserts with Teflon rings( $s = 2\text{ mm}$ ).

The experimental results of Nusselt number and friction factor for coiled wire inserts were correlated as given in below:

$$Nu = 0.077156Re^{0.716692} Pr^{0.4} \left(\frac{P}{D}\right)^{-0.253417} \left(\frac{s}{D}\right)^{-0.124382} \quad (17)$$

$$f = 3.970492Re^{-0.367485} \left(\frac{P}{D}\right)^{-0.31182} \left(\frac{s}{D}\right)^{-0.157719} \quad (18)$$

Flow friction and heat transfer behaviour in a twisted tape swirl generator (Fig.4) inserted tube were investigated experimentally by Bas and Ozceyhan [6]. The effects of twist ratios ( $y/D=2, 2.5, 3, 3.5, 4$ ) and clearance ratios ( $c/D=0.0178$  and  $0.0357$ ) were discussed in the range of Reynolds number from 5132 to 24989, and the typical one ( $c/D=0$ ) was also tested for comparison. The Nusselt number increase with the decrease of clearance ratio and twsit ratio, also increase of Reynolds number. Heat transfer enhancement tends to decrease with the increase of Reynolds



number. The highest heat transfer enhancement was achieved as 1.756 for  $c/D = 0.0178$  and  $y/D=2$  at Reynolds number of 5183.



Figure 4. The twisted tapes inserts with tefflon rings

The experimental results of Nusselt number, friction factor and heat transfer enhancement for the tube with twisted tape inserted were correlated depending on twist ratio ( $y/D$ ) and clearance ratio ( $c/D$ ) as follows:

$$Nu = 0.406903Re^{0.586556} Pr^{0.38} \left(\frac{y}{D}\right)^{-0.443989} \left(\frac{c}{D}\right)^{-0.055072} \quad (19)$$

$$f = 6.544291Re^{-0.452085} \left(\frac{y}{D}\right)^{-0.730772} \left(\frac{c}{D}\right)^{-0.1579} \quad (20)$$

$$\eta = 9.750184Re^{-0.177983} Pr^{0.38} \left(\frac{y}{D}\right)^{-0.183513} \left(\frac{c}{D}\right)^{0.009558} \quad (21)$$

Promvonge et al. [7] investigated turbulent convective heat transfer characteristics in a helical-ribbed tube fitted with twin twisted tapes experimentally (Fig. 5). The experiment was carried out in a double tube heat exchanger using the helical-ribbed tube having a single rib-height to tube-diameter ratio,  $e/D_H = 0.06$  and rib-pitch to diameter ratio,  $P/D_H = 0.27$  as the tested section. The insertion of the double twisted tapes with twist

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ratio,  $Y$ , in the range of 2.17 to 9.39 was to create vortex flows inside the tube. Experiments were carried out for the Reynolds number of about 6000 to 60000. The experimental results revealed that the co-swirling inserted tube performs much better than the ribbed/smooth tube alone at a similar operating condition. The co-swirl tube at  $Y \approx 8$  yields the highest thermal performance at lower Reynolds number ( $Re$ ).

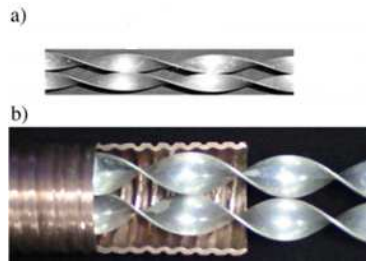


Figure 5. Details of (a) double twisted tapes and (b) helical-ribbed tube with double twisted tape insert.

The empirical correlations developed for the co-swirl ribbed-tube with double twisted tapes are;

$$Nu = 0.238Re^{0.627} Pr^{0.3}(Y)^{-0.346} \quad (22)$$

$$f = 31.675Re^{-0.4}(Y)^{-0.458} \quad (23)$$

Gurlek [8] investigated influences of turbulators as V-winglet twisted tape (Fig. 6) on the heat transfer rate ( $Nu$ ), friction factor ( $f$ ) and the thermal hydraulic performance ( $\eta$ ) in concentric tube heat exchangers experimentally and numerically in the doctoral thesis. The experiments were conducted using twisted tapes with four pitch ratios ( $PR = 2.88, 3.33, 4.44$  and  $5.55$ ) for Reynolds number ranging from 7000 and 15000. The best heat transfer value and the highest friction factor value were obtained by using V-winglet with 130 mm pitch distance. However twisted tape with 250 mm pitch distance provides a higher thermal hydraulic performance value than

twisted tape with 130 mm pitch distance which are exposed to the dominant effect of the friction factors.

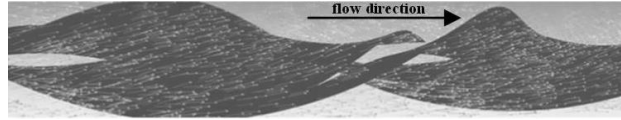


Figure 6. Velocity vectors at V-winglet type twisted tape

Empirical correlations for friction factor ( $f$ ) and Nusselt number ( $Nu$ ) were derived by using the results obtained from experimental studies. Correlations were derived for the tube fitted with V-winglet twisted tape in the range of Reynolds number between 7000 and 15000 for four different pitch distances separately as follows:

$$\begin{aligned} Nu &= 0,6437 Re^{0,4983} Pr^{-0,07477} & PR &= 2.88 \\ f &= 14,02 Re^{-0,3916} \end{aligned} \quad (24)$$

$$\begin{aligned} Nu &= 0,6769 Re^{0,4916} Pr^{-0,01616} & PR &= 3.33 \\ f &= 5,414 Re^{-0,3057} \end{aligned} \quad (25)$$

$$\begin{aligned} Nu &= 0,7351 Re^{0,4842} Pr^{0,05629} & PR &= 4.44 \\ f &= 11,62 Re^{-0,4222} \end{aligned} \quad (26)$$

$$\begin{aligned} Nu &= 0,449 Re^{0,5261} Pr^{-0,04494} & PR &= 5.55 \\ f &= 431,1 Re^{-0,8296} \end{aligned} \quad (27)$$

#### 4. RESULTS

The experimental results of heat transfer, friction factor and thermal hydraulic performance in a tube fitted with various turbulence generators in previous studies were compared in this section. Derived empirical correlations for each turbulator generator were used for comparing.

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Common variables were taken into consideration when making this comparison. The calculations were conducted using derived empirical correlations for ten various turbulence generators with common twist ratio ( $y/D = 2.88$ ). The calculations were performed for different the flow rates in the range of Reynolds number from 7000 to 15000. This Reynolds number range was chosen for involving in all previous studies.

Relationships between Nusselts number and Reynolds number in a tube fitted with ten various turbulence generators are shown in Fig. 7. In this figure, turbulence generators were compared each other. According to the general trend, the Nusselt number increases with increasing Reynolds number. High values of heat transfer rates are observed in high Reynolds numbers. The highest Nusselt number value was achieved with V-winglet twisted tape which is applied by Gurlek [8] at Reynolds number of 7000.

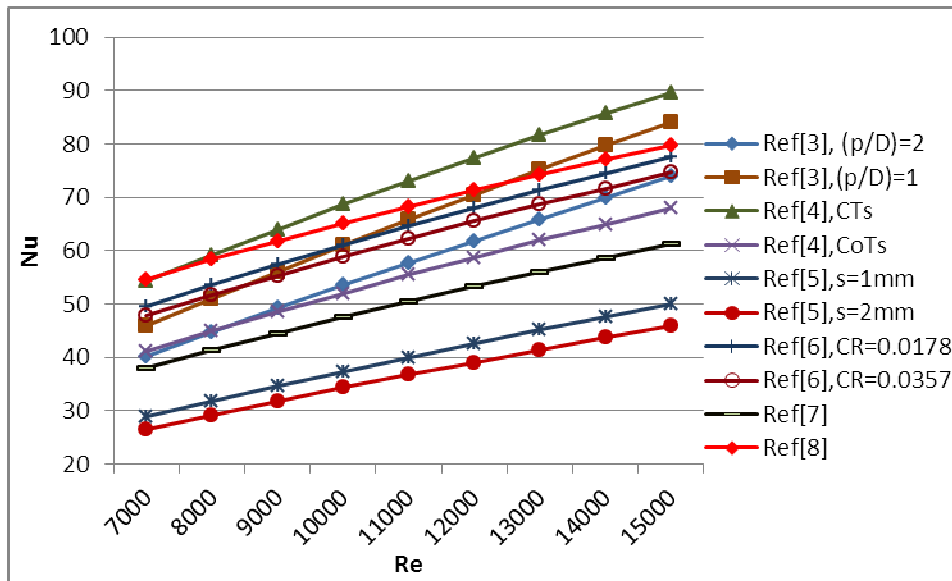


Figure. 7. Effects of the ten various turbulence generators on heat transfer

On the other hand the twin counter twisted tape (CTs) which is applied by Eiamsa-ard et al. [4] provided highest heat transfer rate in all the turbulence generator models at higher Reynolds number values. The lowest heat transfer rate was obtained with the wire inserts which is studied by Gunes et al [5]. The reason of this, special geometric shaped twisted tapes produce rotational motion, stronger swirl intensity, longer fluid travelling distance and a better tangential contact between the fluid and the tube wall.

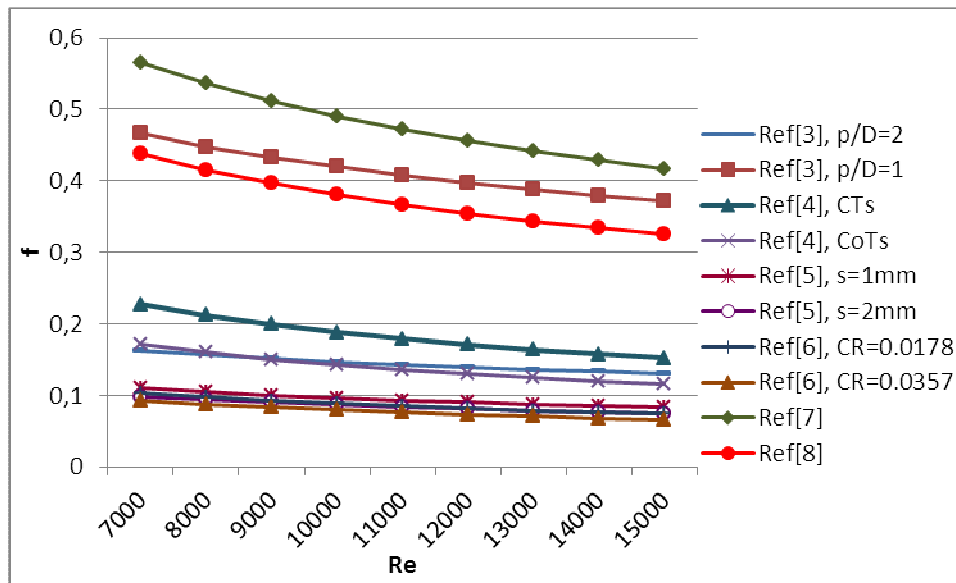


Figure 8. Effects of the ten various turbulence generators on friction factor

Fig. 8 shows the variation of the friction factor (f) with Reynolds number (Re) for ten different turbulence generator models. In the figure, the friction factor value tends to decrease with the increase of Reynolds number. The lowest friction factor is observed with twisted tape swirl generator with clearance ratio of 0.0357 which is applied by Bas and Ozceyhan [6] while the highest friction factor is obtained with helical-

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ribbed tube fitted with twin twisted tape which is applied by Promvonge et al [7].

Fig. 9 presents the variation of the thermal-hydraulic performance ( $\eta$ ) at the same pumping power with Reynolds number for six different turbulence generator models. The effect of Reynolds number indicates the level of efficiency of the twisted tapes. The thermal hydraulic performance decreases with increasing Reynolds number. The performance values under the critical value  $\eta = 1$ , is observed with helically twisted tapes [3] at all Reynolds number. This results explains that helically twisted tapes are not efficient in terms of thermal and hydraulic performance. The values of thermal hydraulic performance obtained from using various turbulence generators are in the range of 0.73-1.39 depending on geometric properties. According to the result twisted tape swirl generator with clearance ratio of 0.0357 which is applied by Bas and Ozceyhan provides the highest thermal hydraulic performance values.

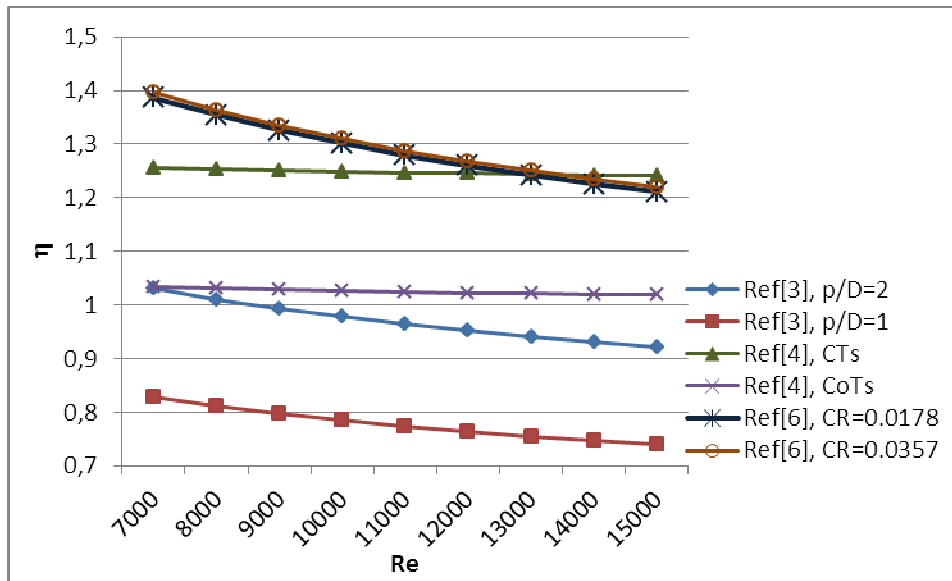


Figure 9. Effects of the ten various turbulence generators on thermal-hydraulic performance

## 5. CONCLUSIONS

Obtained results can be summarized as below:

1. According to the general trend, the Nusselt number increases with increasing Reynolds number.
2. The twin counter twisted tape (CTs) provided highest heat transfer rate in all the turbulence generator models at high Reynolds number values.
3. The friction factor value tends to decrease with the increase of Reynolds number. Helical-ribbed tube fitted with twin twisted tape provides highest friction factor value.
4. The thermal hydraulic performance decreases with increasing of Reynolds number. Twisted tape swirl generator with clearance ratio of 0.0357 provides the highest thermal hydraulic performance values.

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