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Performance boost of a helical heat absorber by utilization of twisted tape turbulator, an experimental investigation

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ABSTRACT

Present investigation, experimentally probes the influence of utilization of twisted tape turbulator inside a helical tube. Twisted tapes mix the flow and enhance the heat transfer rate. The pitch of twisted tape was considered as variant parameter. Tube was considered to be under constant heat flux. Also, Re number was within the range of 6000–32000 which denotes that flow was the turbulent flow regime. Various parameters including Nu number, friction factor and entropy generation were evaluated. Furthermore, performance evaluation criteria based on first law and second law of thermodynamics together with economic performance parameters were evaluated. The results revealed that by the increment of water flow rate and the decrement of twist pitch the Nu number would face a rise of 66% in comparison with the smooth helical tube. Also, it was found that the application of twisted tape has more effect on frictional entropy generation rather than thermal entropy generation. Finally, the results indicated that the maximum value of η_c was about 1.59 $\times 10^{-7}$ (\$/J) which presented up to 2.7 times enhancing in the financial beneficial of the helical tube.

1. Introduction

Through the recent decades, the helical tubes were found to be of very interesting kinds of heat transfer equipment. The helical tubes have better thermal performance than straight tubes due to the curvature nature of the following path of the working fluid inside them [1,2]. The curved path causes the secondary flows too be created which leads in heat transfer augmentation [3]. Further to the mentioned points the compactness of the helical tubes is another points that makes them very attractive for the thermal engineers [4–6]. Despite the well thermal performance of helical tubes, many investigations were conducted by researcher to boost the

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performance of helical tubes as much as possible. In the following the authors have tried to provide a brief summary of the most recently conducted investigations in this regard.

The efforts to enhance the heat transfer enhancement in helical tubes are mainly divided in two main groups. The first one is use passive methods and the second group includes the active methods [7,8]. The majority of passive methods are assigned to the application of variation in the geometry and the application of inserts [9]. In this regard, Kumar et al. [10] experimentally investigated the entropic behavior of the circular tube heat exchanger with twisted tape inserts that included V cuts. Their results presented that the minimum entropy generation was found to be for the twisted tape with twist ratio of 3. Through another investigation, the Rueng-payungsak et al. [11] probed the effect of rectangular cut twisted tapes on the performance enhancement of a heat exchanger. Bahuguna et al. [12] reported the effect of triple blade vortex generator on the second law performance of a tube heat exchanger equipped with the mentioned inserts. Their findings represented that the entropy generation due to heat transfer was significantly more than the entropy generation due to friction factor. Promvonge et al. [13] reported that the application of louvered V-winglet vortex generators will lead in 2.48 performance index for a tubular heat exchanger. Also, they proposed empirical correlations for the prediction of Nu number and friction factor. Zhao et al. [14] probed the effect of using vortex generators at the performance of heat exchangers with 180 ° turning point. their reports presented that the application of vortex generators could lead in up to 35% heat transfer enhancement. Chen et al. [15] numerically assessed the effect of helical corrugation on thermal-exergy and financial advantageous of the helical tube. Though their study, the helical tube was supposed to have constant wall temperature. The pitch and the depth of the corrugation was considered to be the varying parameters. Also, in their investigation the flow regime was considered to be in turbulence regime of the working fluid. They reported that the application of circular corrugation would lead in up to 1.54 times enhancement in the Nu number. Furthermore, their results presented up to 6.67 times improvement in the economic advantageous of the helical tube. Wei et al. [16] reported that application of Nano-fluids with circular corrugation leads in significant thermal enhancement of helical tube in comparison with smooth helical tube. Khorasani et al. [17] experimentally investigated the effect of different coil shapes on the frictional performance of the micro coiled tubes. They proposed empirical correlations to predict the friction factor of each considered shape. In another investigation, Khorasani et al. [18] experimentally evaluated the effect of spring wire turbulators which were inserted inside the helical tube on thermal performance of coiled tubes. They reported that the application of spring wire turbulator would lead in 73% performance enhancement of helical tube. Through their investigations the Dean number was within the range of turbulent regime. The dimeter of the wire and the pitch of coiled wire were supposed to be as varying parameters. Pourhedayat et al. [19] proposed an empirical correlation for the prediction of friction factor, Nu number and exergy destruction of helical tube with spring wire inserts. Zhang et al. [20] reported that the application of spherical corrugations leads in augmentation of thermal performance. They reported an increment of up to 66% and 28% for the Nu number and friction factor. Khairul et al. [21] investigated the influence of various Nano-fluids on thermal and frictional parameters of a helically coiled tube. Through their investigation, different nano-fluids having volume fractions within the range of 1–4%. They reported that the application of nano-fluids could lead in 7.14% reduction in the entropy generation. Xu et al. [22] numerically examined the effect of spiral corrugation on the performance of a tube-in-tube helical corrugation. They reported that the application of one-start spiral corrugation could lead in 25% performance enhancement. In a numerical investigation, Kurnia et al. [23] examined the effect of twisted tape inserts on the thermal performance of helical tube. The Twist ratio and the type of working fluid were considered as the varying parameter. The study was conducted for the laminar flow regime. They found that for low Prantdl number, the heat transfer enhancement ratio was more in lower values of Re number. However, for the higher values of Prantdl number, the best heat transfer enhancement ratio was found to be in the Re number within the range of 500–1000. In another numerical investigation, Liaw et al. [24] investigated the effect of Twisted tape inserts on frictional and thermal performance of a helically coiled tube. Through their investigation the tube with twisted tape -having the Twist ratio of 7.86- presented the best thermal performance. Furthermore, they proposed some new correlations for the prediction of Nu number and friction factor for a helical tube equipped with twisted tape.

As mentioned before, the active methods were another attractive mechanism that were used to enhance the thermal performance of the helical tubes. One of the very interesting active heat transfer enhancement method, was the application of multi-phase flows. In this regards, Moradi et al. [25] reported that the air/water two phase flow inside a vertical helical tube has better thermal performance than helical tube with single phase working fluid. Their results presented 26% enhancement in the heat transfer coefficient. In another investigation, Abdzadeh et al. [26] conducted an experimental investigation and reported that air water two phase flow could provide up to 20% performance enhancement in helical tube.

From the literature review, it could be concluded that the heat transfer enhancement method in helical tubes are very interesting for the researchers and thermal engineers. It was found that the application of twisted tapes on thermal behavior of helical tubes were probed numerically. However, there was no experimental data in this regard. Furthermore, there was no report on the entropic behavior and economic performance of twisted tapes inside the helical tubes. The present investigation, not only provides experimental results for thermal and frictional behavior of helical tube, but also, by presenting economic and entropic performance of the helical tubes provides industrially applicable results for the thermal engineers. Therefore, the main novelty of the present paper is consisted of three major aspects, first, providing experimental results for the application of twisted tape inside the helical tube, second: providing evaluations based on the second law of thermodynamics for the application of twisted tape inside the helical tube and the finally presentation of economic assessment of the application of twisted tape inside the helical tube which was not provide in the previous investigations.

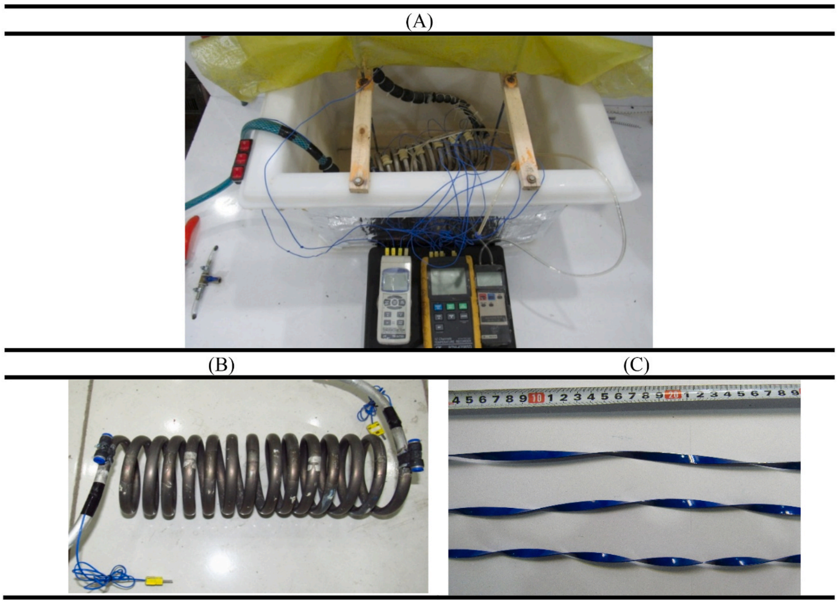

Fig. 1. illustration of setup, B) helical tube, C) twisted tape turbulator.

Table 1
Range of investigated parameters.

Parameter	Range
$\dot{m}[L/h]$	100,200, 300, 400,500
Re	6000–32000
Heat Flux $[W]$	2800–3000
D_{in} (Inner diameter of tube) $[m]$	0.01
D_H (Helix diameter) $[m]$	0.1
P_{TT} (Tape Pitch) $[m]$	0.05,0.075, 0.1
P_{CT} (Coiled Pipe Pitch) $[m]$	0.025
L(tube and twisted tape length) $[m]$	4.396

2. Experimental details, validation and parameter definition

2.1. Setup definition

The present investigation deals with the experimental evaluation of the effect of twisted tape inserts on the thermal frictional, entropic and economic performance of a helically coiled tube. To this aim an experimental test rig was fabricated and developed. Fig. 1 A, presents the general form of the experimental setup. The setup mainly was consisted of a plastic reservoir that was filled with tape water. The water was heated by three individual resistant heaters that totally produced a namely 3 KW thermal energy. the thermal energy after heating the water was extracted by the cold water passing through the helical tube. it should be noted that this type of heat exchange systems is particularly used as the heating system in domestic applications. Indeed, the main part that should be enhanced to improve the thermal performance is the convection heat transfer inside the helical tube. It should be noted that the buoyant effects in the reservoir was neglected and it was assumed that the outer wall of the helical tube received the constant heat flux via conduction form of heat transfer. As mentioned before, to enhance the thermal performance of the helical tube the twisted tape turbulators was used. The twisted tape turbulator had three different pitches of 5, 7.5 and 10 cm. The twisted tapes were fixed by the friction between the twisted tape and the helical tube. The geometrical properties of the tube and the twisted tape together with other test condition are provided in Table 1. It should be noted that despite the fact that no main criterion has been proposed for choosing the size of the experimental test section, the available data in the literature was used to select the appropriate size of the test section (including the helical tube and the twisted tapes). Indeed, the authors have tried to hydraulically and geometrically make a good similarity between the present investigation and those available in the literature.

As could be seen from Table 1, the flow rate was within 100–500 lit/h which provided a Re number within the range of 6000–32000. This range of investigation was used to ensure the flow was within the range of turbulent flow regime. It is noteworthy that for defining the Re number for each certain case the associated Hydraulic diameters were used in the calculation of Re Number. The heat flux was assumed to be in range of 2800–3000 W. The inner diameter of tube was about 1 cm and the helix diameter of helical tube was about 10 cm. The pitch of coiled tube, the length of helical tube and twisted tapes were constant and had the dimensions of 0.025, 4.396 and 4.396 m, respectively. As it could be seen, five different water flow rates were considered in this investigation to

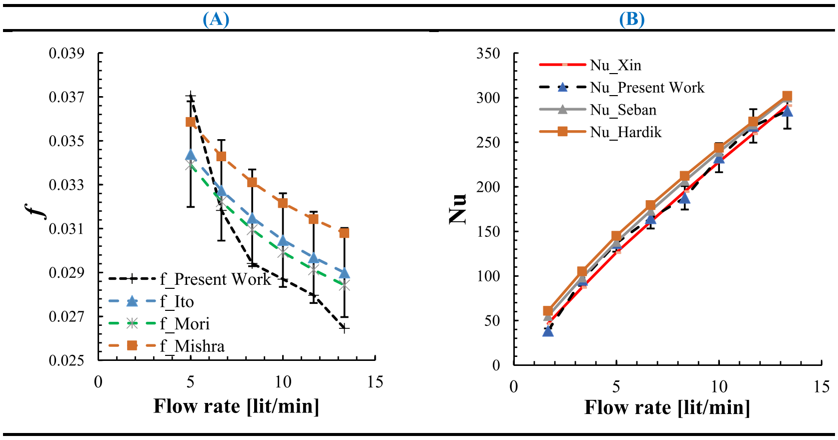

Fig. 2. Comparison of the present results with the available data in the literature.

Table 2
Presentation of uncertainty of results.

Variable	Unit	Value
Water	Liters per minute	±0.25
Outlet temperature water	$^\circ C$	±0.5
Inlet temperature water	°C	±0.5
Nu number	%	±9.65
Friction factor	%	±3
Entropy generation rate	%	±10.85
Uncertainty in the values read from the tables	%	±0.15 – 0.1

check effect of flow rate. It should be noted that for each certain pitch of twisted tape the flow rates were repeated. Another noteworthy point is about the data recording, it should be noted that the data was recorded after the tests reached the thermally stable condition. For recording the thermal data (wall temperature and bulk temperature of water flow in inlet and outlet of tube) K type thermocouples were used. The temperatures that were sensed by K type thermocouples were recorded by a 12 Channel data logger (Lutron BTM-4208 SD). It should be noted that the mentioned set of temperature sensing instruments provided an accuracy of $\pm 0.5\ ^\circ C$. Also, it should be noted that for sensing the pressure drop values, a digital differential pressure drop sensing instrument was used (Lutron PM 9001) which had the accuracy of 2%.

2.2. Validation of results

For any investigation, the credibility of the results is dependent on the comparability of the new results with those that may be available in the literature. To this aim, the results of the present investigation (the results associated with smooth tube) are compared with those that were available in the literature. For the frictional results the present work was compared with the results of Ito et al. [27], Mori et al. [28] and Mishra et al. [29]. Fig. 2 A present the comparison between frictional results of the present investigation with those mentioned before. It is observed that the experimental results were in good agreement with the predicted results. It is noteworthy that the deviation between he results of present paper and the considered reference papers, were less than the uncertainty amount of the considered papers. Which denotes that the data of the present paper are credible. On the other hand, for the credibility of the thermal results, the results of present investigation were compared with the papers published by Hardik et al. [29], Seban et al. [29] and Xin et al. [30]. Fig. 2 B, depicts the comparison of the *Nu* number results of present investigation and those available in the literature. It is seen that the results of present investigation are in good agreement with the available data in the literature which denotes the credibility of the present results. The maximum deviation for the frictional results was about 10% and the maximum deviation associated with the *Nu* number results was about 3%.

3. Uncertainty analysis

The uncertainty of results is one of the most important part of any experimental investigation that denotes the validity of the results. In the present investigation a very well know method that was proposed by Moffat et al. [31]. It should be noted that the mentioned methodology was based on three repetition of the experimental tests. The mentioned method was widely used by other researchers [18,25,32–37] that denotes the credibility of the considered method. The achieved results for the uncertainty of the results in the present paper are provided in Table 2.

4. Results

In the present study, thermal and economical effects of twisted tape inside a helical tube experimentally have been investigated. In the following section, outcomes of pointed work are analyzed in two separated parts for thermal and economical topics.

4.1. Thermal and frictional analysis

In order to investigate the thermal performance of helical tube, application of twisted tapes with three different pitches of $Pitch = 5.0\ 7.5\ and\ 10\ cm$ in five different flow rates of $\dot{m} = 100,\ 200,\ 300,\ 400\ and\ 500\ L/h$ have been probed. With the aim of examining the heat transfer rate enhancement for different test cases, the Nusselt number alongside with flow rate is depicted in Fig. 1 a. Moreover, pressure drop as an essential parameter in thermal systems performance, is presented in terms of friction factor in Fig. 1 c.

For the calculation of Nu number, the below process was follow. The total thermal energy gained by the working fluid could be calculated as follows.

$$\dot{q} = \dot{m}c_p\left(T_{fo} - T_{fi}\right) \tag{1}$$

It should be noted that based on the experiments done, the maximum heat loss was the 8% of the thermal energy produced by three heaters that were fabricated in the bottom of the plastic reservoir.

Then this thermal energy should be equal to what was gained through the convection terms.

$$\dot{q} = \overline{h}A(\overline{T_w} - T_b) \tag{2}$$

Then the $\overline{h}$ (as the mean convective heat transfer coefficient) could be calculated as follows.

$$\overline{h} = \frac{\dot{q}}{A(\overline{T_w} - T_b)} \tag{3}$$

Having the $\overline{h}$, the Nu number is calculated by the below equation.

$$Nu = \frac{\overline{h}D_h}{K_f} \tag{4}$$

In the above equations, the $\dot{q}$ stands for the total energy that is gained by the working fluid. The terms $\dot{m}$ and c_p denote the mass flow rate and the specific heat of the working fluid. Also, the terms $T_{fo}$and T_{fi} are the outlet temperature of the flow and the inlet temperature of the working fluid respectively. It should be noted that the terms A, $\overline{T_{w,i}}$ and T_b stand for the heat transfer surface, mean inner wall temperature and the bulk temperature of the working fluid. For the calculation of the heat transfer surface, the following explanation is noteworthy. Indeed, it was almost impossible to check if the twisted tapes were in complete connection with the heated wall or not (if a complete connection is available then the twisted tape should be considered as fins; the heat transfer surface is increased). But since the width of twisted tape were less than inner diameter of helical tube (inner diameter of the tube was 10 mm, and the width of the twisted tape was 9 mm), the complete connection was almost impossible. Furthermore, looking to the available literature, it could be seen that the heat transfers between the heated walls and the twisted tape were neglected in almost all previous investigations. Even, through the numerical simulations, a gap was considered to be between the twisted tape and the heated wall [23, 24]. Through the present study -following the available literature-the connection between the heated walls and the twisted tape was neglected and only the inner surface of the helical tube was considered as the heat transfer surface.

Furthermore, the $\overline{T_{w,i}}\ and\ T_b$ were calculated by 5 and 6, respectively.

$$\overline{T}_{w,i} = \frac{\sum_{i=1}^{10} T_{w,i}}{10} \tag{5}$$

$$T_b = \frac{T_{f,o} + T_{f,i}}{2} \tag{6}$$

The term $T_{w,i}$ was the inner wall temperature of the helical tube and was predicted by considering the thermal resistance of the Aluminum made helical tube. The following equation denote that how the inner wall temperatures were calculated.

$$T_{wi} = T_{wo} - \frac{Q \ln\left(\frac{D_o}{D_i}\right)}{2\pi KL} \tag{7}$$

In Eq. (7), the Do and Di denote the outer diameter and inner dimeter of the tube. The K stand for the conductive heat transfer coefficient of the flow. Also, the term L stand for the length of tube in the straight form.

For the frictional assessments, the friction factor was used. To calculate the friction factor, the Darcy equation was utilized. The mentioned equation is defined as below:

$$f = \frac{2D_h \times \Delta P}{\rho L V^2} \tag{8}$$

In Eq. (8), the terms D_h, ΔP, V, ρ and L stand for the definition of hydraulic diameter, pressure drop, velocity of main stream, density of operating fluid and the length of patch, respectively.

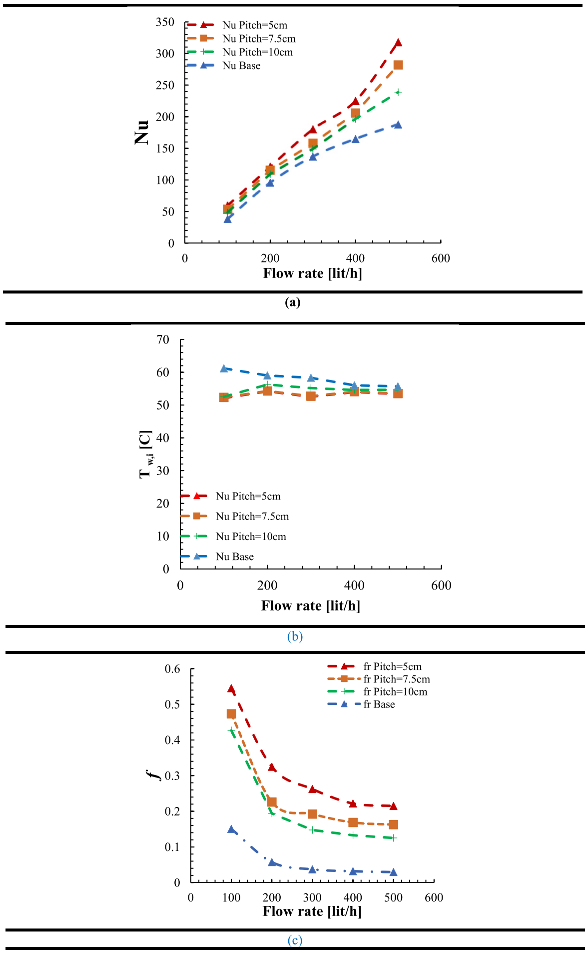

Fig. 3. Comparison of **a)** Nusselt Number and **b)** Inner wall temperatures, c) friction factor for tubes with different pitches of twisted tape.

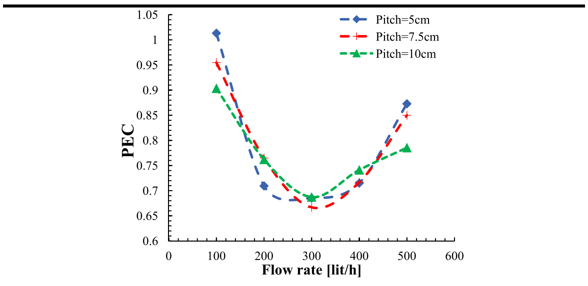

Fig. 4. Thermal efficiency in comparison with flow rate for different twisted tapes.

As could be understood from Fig. 1 a, by decreasing the pitch of twisted tape, dimensionless heat transfer coefficient faces a considerable increment. Moreover, results indicate that *Nu* number's enhancement in higher flow rates is prominent. For instance, for case with Pitch = 5 cm, *Nu* was improved up to 55,62% and 69.37% in flow rates of 100 and 500 L/h, respectively. Pointed finding occurs due to strongly mass bulk rotational movements that results in disruption of thermal boundary layer. Also, the values of mean wall temperature for considered cases are presented in Fig. 3 b. It is seen, that for the base case the wall temperature reduces by the increment of the flow rate. For the cases with twisted tape, firstly the values of mean wall temperature are less than that of base case, Secondly, by the decrement of the twist pitch the values of mean wall temperature reduces. This is due to increment of the heat transfer rate which results in reduction of wall temperature. Indeed, higher heat transfer rate reduces the equilibrium wall temperature.

Fig. 2 c illustrates friction factor trends in case of increment of the flow rate for the cases considered in the present investigation. It was seen that for the cases with twisted tape the friction factor has higher values. Indeed, the presence of the twisted tape results in the

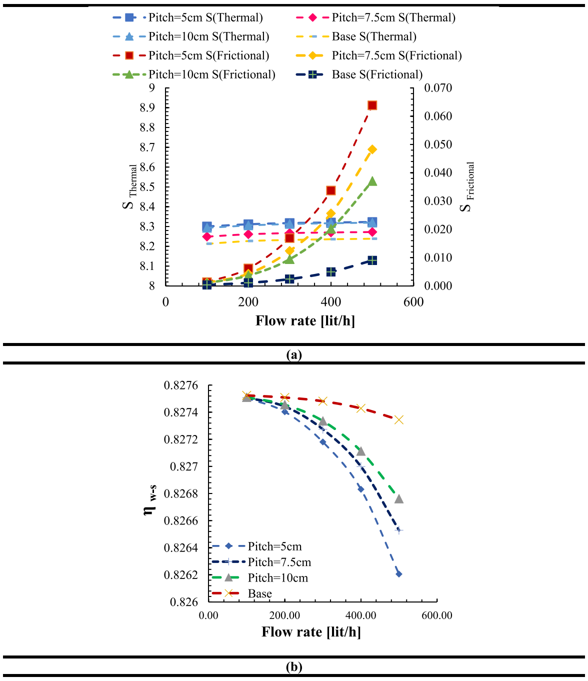

Fig. 5. a) Entropy generation and **b)** η_{w-s} vs flow rate in different pitches of twisted tape.

appearance of swirling flows. The combination of swirling flows with the available secondary flows of the helical path leads in intensification of turbulence viscosity of the flow. This phenomenon results in augmentation of friction factor. Mentioned results show that, friction factor in pipe with Twisted tape having the pitch of 5 cm faces 262%–630% augmentation in friction factor in minimum and maximum flow rates, respectively.

4.2. Performance evaluations based on first law of thermodynamics

Aiming to evaluate the overall efficiency with taking pressure drop into account, Performance Evaluation Criteria (PEC) has been calculated via formula 9. The results of PEC factor are represented in Fig. 4.

$$PEC = \frac{\left(\frac{h_{en}}{h_s}\right)}{\left(\frac{f_{en}}{f_s}\right)^{1/3}} \tag{9}$$

The presented data in Fig. 4 demonstrates that best thermal efficiency takes place in flow rate of 100 and 500 lit/h. In addition, in case with pitch = 5 cm in spite of highest friction factor coefficient, due to the elevated heat transfer rate, the thermal efficiency has better values. From the point of view of PEC which is based on first law of thermodynamics, cases with flow rates of 200 and 300 and 400 Lit/h are less efficient. However, the cases with flow rate of 100 and 500 lit/h are beneficial. The best value of PEC factor was about 1.013 and was seen for the case that had the twist pitch of 5 cm and the flow rate 100 lit/h.

4.3. Performance evaluations based on second law of thermodynamics

In the previous parts, evaluations were based on first law of thermodynamics. In the following section, outcomes of study will be analyzed based on second law of thermodynamics. For this purpose, Thermal and entropy generation separately illustrated in Fig. 5 a. Entropy generation due to thermal and frictional loses could be calculated as follow:

Thermal entropy generation

$$\dot{S}_{gen.\,T} = \dot{m}c_p \ln\left(\frac{T_{fo}}{T_{fi}}\right) \tag{10}$$

Frictional entropy generation:

$$\dot{S}_{gen.\,f} = \frac{\dot{m}}{\rho}\Delta P \frac{ln\left(\frac{T_{f0}}{T_{fi}}\right)}{(T_{f0} - T_{fi})} \tag{11}$$

According to Eq. (10), entropy generation in heat transfer phenomena is related to flow rate ($\dot{m}$), inlet temperature and outlet temperature of flow ($T_{fi}.T_{fo}$). In the present work due to applying constant heat flux in all of testes and having almost fixed inlet

Table 3
A detailed information and values used in calculation of *NPV* .

	Variant	Variant symbol	Value	Unit
Physical variants for water	Thermal conductivity	k_f	0.64	$W/(mK)$
	Specific heat capacity	C_p	4195	$J/(kgK)$
	Dynamic viscosity	μ	6.54×10^{-4}	Pa s
	Prandtl number	Pr	4	
	Density	ρ	997	kg/m^3
Operating variants	Ambient temperature	T_0	298.15	K
	Operating time within a year	τ	2.585×10^7	s
Economic variants	Cost of each meter smooth coil	$\lambda_{l.s}$	9	$\$/m$
	Cost of each meter of tube equipped with twisted tape	$\lambda_{l.cr}$	12.78	$\$/m$
	Interest rate	i	10.5%	
	The number of years the system is considered to be used	n	20	years
	Salvage value	SV	0	$
	Cost of each unit of thermal exergy	ε_q	3.48×10^{-8}	$\$/J$
	Weighting factor	α	4.1	

temperature, outlet temperature was almost same in different cases (in same flow rates). Because of mentioned point, the thermal entropy generation ($\dot{S}_{gen.T}$) in all of cases was almost equal and the minor deviation was occurred due to slight error in fixed heat flux exertion which could be seen from Fig. 5 a. On the other hand, Fig. 5 a shoes that with decreasing the pitch of twisted tape, due to increment of pressure drop, frictional entropy generation increases. It was seen that maximum ratio of frictional entropy generation inside the helical tube (the base mode as the reference mode) was 6.91, whereas the maximum ratio of thermal entropy generation was about 1.01 which denoted that the twisted tape turbulator mostly increases the frictional loses. It was found that by the increment of flow rate, the frictional loses faces a sharp increment, however, for the thermal entropy generation curves this augmentation is very minor. Indeed, the increment of flow rate (increment of turbulence intensity) has more influence on the augmentation of frictional losses of flow rather than increasing the heat transfer rate inside the helical tube.

Fig. 5 b, represents the behavior of Whitte-Shamsunder efficiency factor (η_{w-s}). This parameter was developed as a decision making criteria and considers both the quality and quantity of the heat transfer system. Indeed, previously mentioned parameters (entropy generation and PEC factor) were based only on one of the first law or second law of thermodynamics. Whereas this parameter considers both of the mentioned concepts. The mentioned parameter is formulated as following:

$$\eta_{w-s} = 1 - \frac{T_0 \dot{S}_{gen}}{\dot{q}} \tag{12}$$

According to Eq. (12) and considering the constant heat flux assumption, $\dot{S}_{gen}$ in the only effective parameter and with attention to Fig. 5 b, frictional entropy generation could be the source of η_{w-s} behavior. As could be understood from Fig. 5 b and based on the second law of Thermodynamics, despite of heat transfer enhancement (first law of thermodynamic), system is not efficient and with increasing of the pitch of tape the η_{w-s} has increased. The maximum values of the η_{w-s} was always seen to be in the bas tube. The best value for the η_{w-s} was found to be in low water flow rates which had the amount of 0.828 which presented about 1% better performance when compared to the case with twisted tape with twist pitch of 5 cm. By the increment of water flow rate, the distance between the curves associated with base mode and cases with twisted tape increases. The minimum values of η_{w-s} in comparison with base mode was about 0.826 and was seen in the water flow rate of 500 lit/h and at the case with twist pitch of 5 cm. Indeed, by the increment of water flow rate the increment of frictional losses dominates the thermal energy absorbs.

4.4. Economic analysis

In order to evaluate the results of a study from industrial and practical point of view, economical investigation results are essential. Since the focus of most of researchers was first and second law of thermodynamics, the lack of economic study in pointed topics is completely evident. Intending to publish a complete analysis of results and making decision more reliable and realistic, this section is presented. To this aim the parameter of Net price for each unit of transferred heat load (η_p) [38] is considered to be assessed. Which is considered to be defined as below (Eq. (13)):

$$\eta_p = \frac{NPV}{Q} = \frac{NPV}{GC_p\left(T_{fo} - T_{fi}\right)\tau} \tag{13}$$

Through which the NPV is defined by Eq. (14).

$$NPV = \tau\left(\varepsilon_q \Delta E_q - \varepsilon_p \Delta E_p\right) - [I_0 - SV(P/F.i.n)](A/P.i.n) \tag{14}$$

The NPV is called as the net profit value. Besides, the I_0 was considered as the first cost of heat exchanger and is considered to be calculated based on the geometry of the tube ($I_0 = \lambda_l \times L$). Also, the λ_l denotes the cost each meter of the considered type of tube. Furthermore, “n" stands for the total time that the heat exchanger is assumed to be used. On the other hand, “SV” denotes the salvage values and was expected to be Zero in the present investigation. Also, the $(P/F.i.n)$ and $(A/P.i.n)$ stand for the factor of profit and factor of capital recovery. Besides, the variant of τ is total time that the system is supposed to be used during a year. Through Eq. (14), the

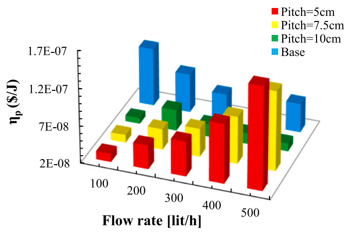

Fig. 6. a) Net profit per unit transferred heat load.

variants of the ΔE_q and ΔE_p denote the thermal exergy that is recovered and losses due to the frictional behavior of the operating fluid, respectively. It is noteworthy that a comprehensive explanation about the economic evolution of the heat exchangers was presented by Wu et al. [38]. Furthermore, the details of considered factors and the values of some fixed parameters are presented in Table 3.

Fig. 6 depicts the η_c in terms of flow rate for different pitches of twisted tape. It is seen that the cases that included the twisted tape have more values of η_c. This point denotes that the application of twisted tape inside the helical tube economically is beneficial; especially, by the increment of water flow rate, it could be seen that the deviation between the cases with turbulator and the base case fins significant amount. The maximum value of η_c was about 1.59 $\times 10^{-7}$ ($/J) which presented up to 2.7 times enhancing in the financial beneficial of the helical tube.

5. Conclusions

The present paper investigates the effect of twisted tape with different pitches on various parameters of helical tube. The tube was considered to be under constant heat flux that was exerted in a condition that was very similar to domestic usage of the helical tubes. The thermal, frictional and performance parameters together with economic performance parameters were evaluated. The most highlighted out comes of the present paper are provided as below:

- The results indicated that by the application of twisted tape turbulator the Nusselt number would face an augmentation of up to 66%.
- It was found that the maximum augmentation in the Nu number was observed in the smallest pitch and higher water flow rates.
- It was seen that maximum ratio of frictional entropy generation inside the helical tube (the base mode as the reference mode) was 6.91, whereas the maximum ratio of thermal entropy generation was about 1.01 which denoted that the twisted tape turbulator mostly increases the frictional loses.
- The maximum values of the η_{w-s} was always seen to be in the bas tube. The best value for the η_{w-s} was found to be in low water flow rates and had the amount of 0.828
- The best value of PEC factor was about 1.013 and was seen for the case that had the twist pitch of 5 cm and the flow rate 100 lit/h
- The maximum value of η_c was about 1.59 $\times 10^{-7}$ ($/J) which presented up to 2.7 times enhancing in the financial beneficial of the helical tube

Author statement

Changgui Xie: Resources, Formal analysis, Investigation; **Ibrahim B. Mansir**: Writing and editing-revised draft, Investigation; **Ibrahim Mahariq**: Investigation, Methodology, **Pradeep Kumar Singh:** Writing - review & editing, Conceptualism, **Akbar Arsalanloo:** Resources, Formal analysis, setup preparation, Running experiments, **Rehman Muhammad Shahzad:** Writing - review & editing, Methodology, Formal analysis; **Fahd Jarad**: writing original draft.

Declaration of competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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Nomenclature

A	cross section of tube, (m^2)
Cp	Specific heat capacity, $(J/(kgK))$

D_H	Coil diameter (m)
D_{in}	Inner diameter of tube (m)
e	Specific flow exergy (J/kg)
f	Friction factor
$\dot{m}$	Mass flow rate (kg/s)
h	Convective heat transfer coefficient, $W/(m^2K)$
i	Interest rate
I_0	Primary cost of coil $(\$)$
L	coil length (m)
n	Life of coil, year
N_L	Length to inner diameter ratio of, $N_L = L/d$
N_{Tw}	Dimensionless temperature of wall, $N_{Tw} = T_w/[\mu^2/(\rho^2 C_p d^2)]$
N_{Ti}	Dimensionless temperature variance, $N_{Ti} = \Delta T_i/T_0$
Nu	Nusselt number
P_{TT}	Tape Pitch (m)
t	Thickness of twisted tape (m)
P_{CT}	Coil pitch (m)
∇	p Pressure drop (Pa)
Pr	Prandtl number
Q	Heat load (J)
Re	Reynolds number
St	Stanton number
$\dot{S}_{gen,T}$	Thermal entropy generation
$\dot{S}_{gen,f}$	Frictional entropy generation
W	Heat flux (W)
T	Temperature (K)
SV	Maintenance value of coil at the end of year ($)
η_{W-S}	White-Shansunder efficiency

Greek symbols

α	Weighting factor
ρ	Fluid density (kg/m^3)
k_f	Thermal conductivity of working fluid, $(W/(mK))$
θ	Dimensionless inlet temperature difference, $\theta = \Delta T_i/T_w$
λ_l	Cost of each certain length of coil $(\$/m)$
ε_q	Unit price of exergy in accordance with heat transfer, $(\$/J)$
ε_p	Unit price of exergy in accordance with pressure drop $(\$/J)$
η_p	Net profit of unit transferred heat load $(\$/J)$
ΔP	Pressure drop (Pa)
ΔT_i	Temperature variation, $\Delta T_i = T_w - T_{fi}$ (K)
μ	Dynamic viscosity (Pa s)
τ	Annual operating time of coil, (s)

Subscripts

f	Fluid
fi	Inlet fluid
fo	Outlet fluid
w	Wall

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