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THE EFFECTS OF TWISTED STRIPS WITH DIFFERENT LENGTH ON HEAT TRANSFER AND PRESSURE DROP IN CONCENTRIC HEAT EXCHANGER

Zeki ARGUNHAN [1,*], Cengiz YILDIZ [2], Emin EL [3]

[1] *Bitlis Eren University, Department of Mechanical Engineering, Türkiye, zargunhan@beu.edu.tr*
* *Corresponding author*
[2] *Fırat University, Department of Mechanical Engineering, Türkiye, cyildiz@firat.edu.tr*
[3] *Bitlis Eren University, Department of Mechanical Engineering, Türkiye, eel@beu.edu.tr*

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ABSTRACT

This study investigated the use of twisted strips as passive turbulators to improve heat transfer efficiency in concentric heat exchangers. In the experiments, four different lengths of twisted strips were designed (l = 0.25 m, 0.5 m, 0.75 m, and 1 m) and their performance was evaluated in a system with air and water fluids. The effects of twisted strips on heat transfer and pressure drops are investigated in both parallel and countercurrent flow patterns, and the results are analyzed in terms of Nusselt and Reynolds numbers. The results indicated that tubes using twisted strips achieved significant increases in the heat transfer coefficient compared to straight tubes. The highest increase in heat transfer performance reached 78% when the length of the twisted strip was $l/L=1$. The Nusselt number increased by a factor of 1.2 to 1.8, depending on the length of the twisted strips. The shortest strip length ($l/L=0.25$) resulted in the lowest heat transfer performance. However, these improvements were accompanied by significant increases in pressure drop; for full-length strips, the pressure drop increased by nearly 100%. The pressure drop increased slightly as the Reynolds number increased. The swirling flow generated by the twisted ribbons plays an important role in increasing the heat transfer. Despite the increases in pressure drops, the energy loss is negligible compared to the heat transfer gain achieved. In conclusion, the length ratio of the twisted strips significantly affects the thermal performance of the heat exchanger while increasing the pressure losses. These findings demonstrate the effectiveness of twisted strips as passive turbulators.

1 INTRODUCTION

Heat exchangers are extensively utilized in heating and cooling systems. Enhancing heat transfer efficiency can facilitate the design of smaller, more economical, and energy-efficient heat exchangers.

Heat transfer enhancement can be achieved through various augmentation strategies, which are generally classified into three primary categories: (i) Passive Methods, (ii) Active Methods, and (iii) Combined Methods [1]. Passive methods typically involve alterations to the surface or geometry of the flow passage, such as using inserts or additional devices [2]. Conversely, active methods are more intricate, requiring external energy input to achieve the intended flow modifications and improve heat transfer rates. Combined methods integrate multiple approaches to create a synergistic improvement in performance.

Passive techniques, particularly those employing inserts in the flow passage, offer advantages over active techniques due to their straightforward and cost-effective manufacturing processes, and their ease of application in existing heat exchangers. Consequently, extensive research has been conducted on passive methods and their fluid flow dynamics. Swirl flow generators are a commonly employed solution to enhance heat transfer rates in such setups [3].

Swirl flow devices, recognized as passive heat transfer enhancement methods, generate a rotational flow along an axis parallel to the main flow direction, thereby increasing heat transfer performance [4]. Research on swirling flows in pipes is typically divided into two categories: continuous swirl and decaying swirl. Investigations have explored the heat transfer and pressure drop behavior of fluids in tubes with continuous swirl flows [2], [5], while studies on decaying swirl flows have focused on understanding their thermal transfer characteristics [6], [7].

Regarding compound techniques, R. Babu et al. [8] carried out an extensive review of both experimental and numerical studies addressing passive compound heat transfer enhancement methods. Notable examples discussed in the review included setups such as helical ribbed tubes combined with double twisted tape inserts, dimpled tubes integrated with swirl generators in the form of twisted tapes, and irregular wire coils used in conjunction with twisted tapes. These compound approaches were found to deliver significantly higher heat transfer coefficients compared to the use of a single enhancement strategy alone.

R. Mashayekhi et al. [9] investigated the thermo-hydraulic performance of a bent elliptical pipe integrated with bent tape inserts, focusing on factors such as geometric configurations, Reynolds numbers, and varying slopes. The findings demonstrated that the Nusselt numbers and friction factors were higher compared to those observed in smooth elliptical pipes. Similarly, B. Kumar et al. [10] explored the influence of perforated and V-cut twisted tapes, showing that modifications in twist ratios and V-cut dimensions led to increases in both the Nusselt number and the friction factor.

Y. Hong et al. [11] proposed a synergistic design involving a spiral corrugated pipe with multiple twisted bands for liquid-gas heat exchange. This configuration enhanced turbulent heat transfer and flow performance, leading to higher Nusselt numbers. Likewise, M. M. K. Bhuiya et al. [12] explored perforated triple-twisted tape inserts with varying porosities, finding that reduced porosity improved heat transfer and thermal efficiency. B. K. Dandoutiya and A. Kumar [13] studied the use of W-cut twisted belts in a double-pipe heat exchanger, showing that turbulence and flow separation induced by the inserts enhanced heat transfer rates. R. Behcet et al. [14] placed a propeller-type turbulator at the inlet of the inner tube to increase heat transfer in parallel-flow heat exchangers and experimentally investigated the effect of this structure on heat transfer and friction losses. In the experiments, they used air as hot fluid and water as cold fluid and made measurements at Reynolds numbers between 8000 and 24000. The use of a turbulator increased the heat transfer by 25.5%-50.3%, while increasing the friction losses by about five times. Exergy analysis showed that the exergy losses were 15% lower in the turbulator system compared to the empty pipe. As a result, it was determined that the use of turbulators both improves thermal performance and provides thermodynamic advantages. Z. Argunhan and C. Yıldız [15] experimentally investigated the effects of rotary generators with different numbers of holes on heat transfer and pressure drop in a nested tube heat exchanger. In the experiments, rotary generators with a 55° blade angle and one to four circular holes were installed at the start of the inner tube, and it was found that the four-hole generator increased heat transfer by up to 83% when using air and water.

This study investigates passive heat transfer enhancement using twisted metallic strips as turbulators in the inner pipe of a concentric double-pipe heat exchanger. The effects of various tape lengths (l/L = 0.25, 0.5, 0.75, 1) on thermal performance and pressure drop were experimentally examined under both parallel and counterflow configurations. The results revealed that twisted tapes significantly enhance heat transfer efficiency. A comprehensive evaluation of the heat gain-to-pressure loss ratio enabled the identification of optimal tape length and Reynolds number range. The study contributes to the development of energy-efficient designs for compact heat exchangers by promoting turbulence-induced thermal enhancement.

2 METHOD

The heat transferred from hot fluid to cold fluid in the concentric type heat exchangers [16];

$$Q = m_h Cp_h (T_{hi} - T_{ho}) \tag{1}$$

The heat received by the cold water is

$$Q = m_c Cp_c (T_{ci} - T_{co}) \tag{2}$$

The heat transfer for hot and cold fluids can be expressed using the following equations:

$$Q = m_h Cp_h (T_{hi} - T_{ho}) = m_c Cp_c (T_{ci} - T_{co}) \tag{3}$$

$$Q = h_h A (T_m - T_w) \tag{4}$$

The average fluid temperature, T_m, was computed as the arithmetic mean of the inlet and outlet temperatures of the fluid. This value was utilized to represent the fluid's physical properties at the average temperature. The wall temperature, T_w, was calculated as the arithmetic mean of the measurements taken by thermocouples affixed to the pipe wall.

The theoretical Nusselt number for the pipes was calculated using the Dittus-Boelter correlation, as provided in "Eq. 5" [17].

$$Nu = 0.023Re^{0.8}Pr^{0.4} \tag{5}$$

Additionally, the Nusselt number, representing the air-side average heat transfer coefficient, is expressed as

$$Nu = \frac{hD_e}{k} \tag{6}$$

$$Re = \frac{VD_e}{\nu} \tag{7}$$

$$D_e = \frac{4A_c}{P} \tag{8}$$

3 EXPERIMENTAL STUDY

The experimental setup used in the study is given in Figure 1a and the schematic representation of the system is in Figure 1b. A variety of twisted turbulators were designed and evaluated during the experiments. These turbulators were installed at the inlet section of the inner pipe within the concentric heat exchanger.

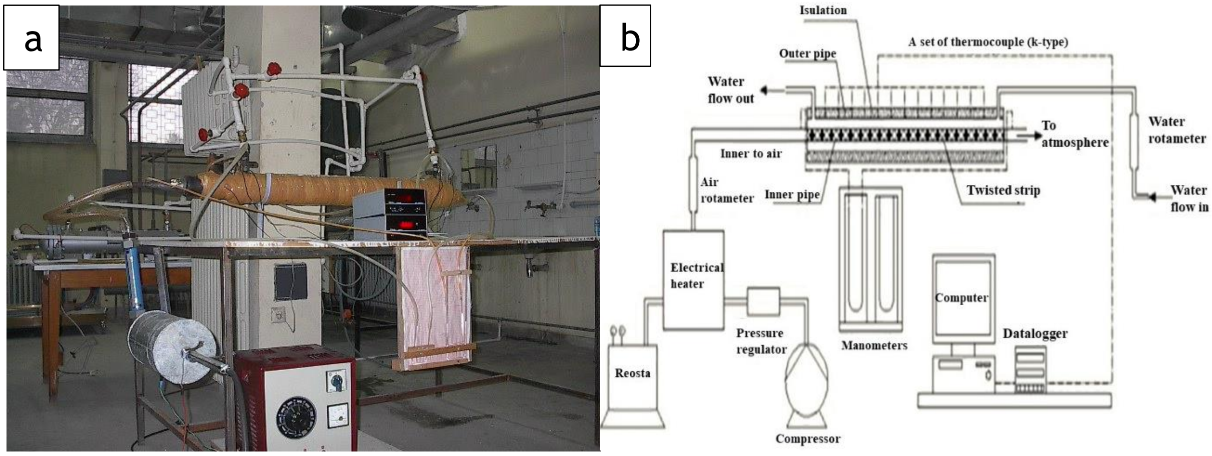

***Figure 1**. **a)** The experimental setup **b)** Schematic diagram of the experimental system.*

The design of the turbulator, formed by twisting galvanized iron strips with dimensions of 60 mm in width (H) and 2 mm in thickness (t), is illustrated in Figure 2. The overall lengths of the turbulators were 275 mm, 550 mm, 582 mm, and 1100 mm. The pitch, defined as the distance traveled by the strip after one full rotation around its axis, is illustrated in the figure. All turbulators tested were constructed with a consistent pitch of 100 mm. The inner diameters of the inner and outer pipes were 50 mm and 70 mm, respectively, with an overall pipe length of 1100 mm.

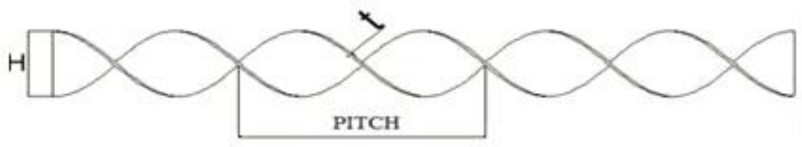

Figure 1*. Geometrical structure of turbulator.*

Tubes equipped with twisted tapes of different lengths and tape-length ratios (l/L=0.25, 0.5, 0.75, 1) were investigated. Additionally, for comparative purposes, a full-length tape was used in certain tests.

During the experiments, a twisted tape was positioned at the inlet of the test section to function as a swirl flow generator, promoting enhanced heat transfer within the tube. Hot air flowed through the inner pipe, while cold water moved through the surrounding annular space. The air entering the system was heated via an electrical heater, with its power output controlled using a variable-output voltage transformer.

The inlet and outlet temperatures of air and water, as well as those at 10 evenly distributed points along the inner pipe, were measured using a multichannel temperature monitoring system equipped with Fe-constantan thermocouples.

Air and water flow rates were adjusted using rotameters and valves positioned at the inlets. Pressure drops across the heat exchanger pipes were determined with the help of two inclined water manometers connected to the inlet and outlet sections.

Experiments were carried out under both parallel and counter flow arrangements, testing a variety of turbulent elements over a wide range of Reynolds numbers.

4 RESULTS

Before assessing the impact of the turbulators, an equivalent number of experiments was conducted with an empty inner pipe across Reynolds numbers ranging from 3400 to 6900. This step was undertaken to validate the methodology against the Dittus-Boelter correlation.

Based on the experimental data collected from the air side, the relationship between Nusselt numbers and Reynolds numbers was analyzed and plotted for both parallel and countercurrent flow configurations, as shown in Figures 3 and 4, respectively.

The Nusselt numbers showed a significant increase when turbulators were employed in the tubes. For countercurrent flow, the improvement in Nu numbers reached up to twice the values observed in the empty tube. The effect of turbulators on heat transfer was more pronounced at higher Reynolds numbers. While a similar trend was noted for parallel flow, as shown in Figure 3, the enhancement was 10-20% lower than that observed for countercurrent flow, as shown in Figure 4.

The overall findings confirm that, in all configurations, the use of turbulators resulted in higher Nusselt numbers compared to a plain tube. The relationship between pressure drops and Reynolds numbers for tubes equipped with twisted tape inserts of varying lengths is shown in Figure 5.

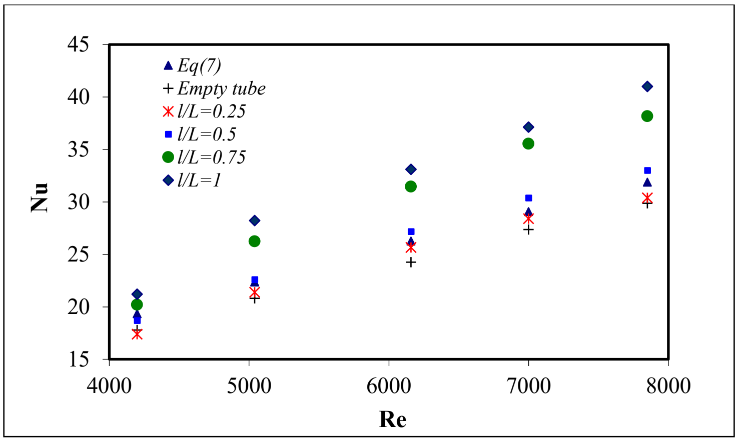

***Figure 2**. Nusselt numbers for parallel flow.*

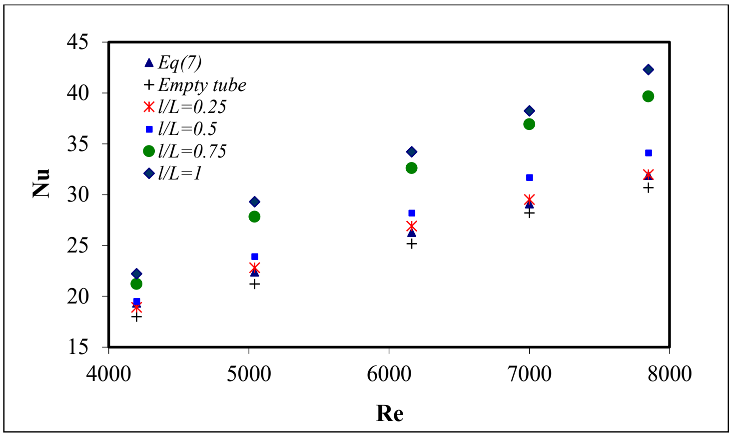

Figure 3. *Nusselt numbers for countercurrent flow.*

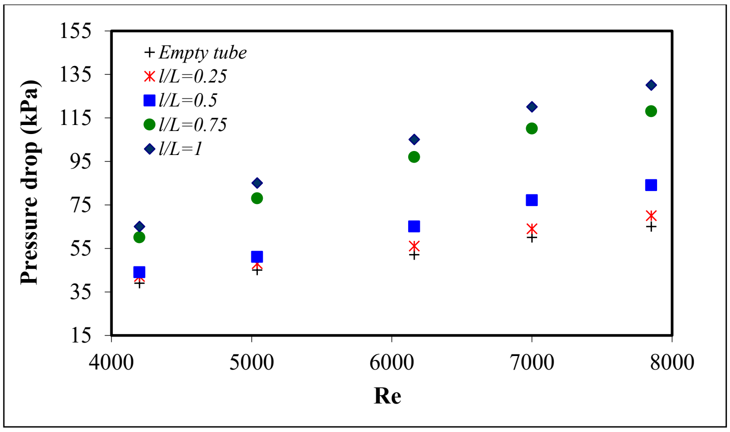

Figure 4. *Pressure drops along the heat exchanger tube equipped with helical turbulators.*

It is evident that the introduction of turbulators into the flow path causes a significant increase in pressure drop. For all turbulators with varying lengths of twisted tape, the pressure drop values were observed to be greater at higher Reynolds numbers.

5 DISCUSSION

In this study, an experimental investigation of heat transfer (Nusselt number, Nu) and pressure drops in a heat exchanger with a plain tube and twisted tapes of varying length ratios of (*l*/L= 0.25, 0.5, 0.75, 1) was conducted and the findings are presented. Several salient concluding remarks are revealed by the present study, which are summarized as follows:

• Compared to a conventional heat exchanger, the augmented model demonstrated a significant improvement in the heat transfer coefficient, with a 61% increase for twisted tape at l/L= 0.25 and a 78% increase for twisted tape at l/L=1.

• The Nu values for tubes with twisted tape inserts of varying lengths were found to be approximately 1.2-1.8 times higher than those for the plain tube, maintaining a consistent trend.

• It was observed that the smallest tape-length ratio (l/L=0.25) resulted in the lowest heat transfer rate.

• The twisted tape with the highest length ratio (l/L=1) provided a heat transfer coefficient approximately 1.5 times greater than that of the lower twist ratio (l/L=0.25).

• Pressure drops increased by up to 100% over the plain tube when using full-length twisted tapes.

• Pressure drops for tubes equipped with twisted tapes of l/L=0.25,0.5,0.75,1 rose with an increase in Reynolds number (Re).

• All tubes fitted with twisted tape inserts exhibited higher mean pressure drops compared to the plain tube.

• The pressure drop ratio showed a slight increase with the rise in Reynolds number.

• While turbulators caused a notable increase in pressure drop, the corresponding energy loss was negligible compared to the heat gain obtained from their use.

It was observed that an increase in heat gain corresponded with an increase in pressure drop. However, the energy loss resulting from the pressure drop was significantly lower than the heat gain achieved. The variation of the heat gain-to-pressure drop ratio as a function of the Reynolds number is illustrated in Figures 6 and 7 and can be calculated using the following equation [16], [18] :

$$\frac{Q_{NG}}{\Delta P_A} = \frac{Q_{ts} - Q}{\Delta P_{ts} - \Delta P}\ 1000.V \tag{9}$$

Where ΔP_{ts} (kPa) and ΔP (kPa) are the pressure decreases in the heat exchanger with and without the twisted strips, respectively, while Q_{ts} (W) and Q (W) are the heat energy amounts transferred in the heat exchanger with and without the twisted strips.

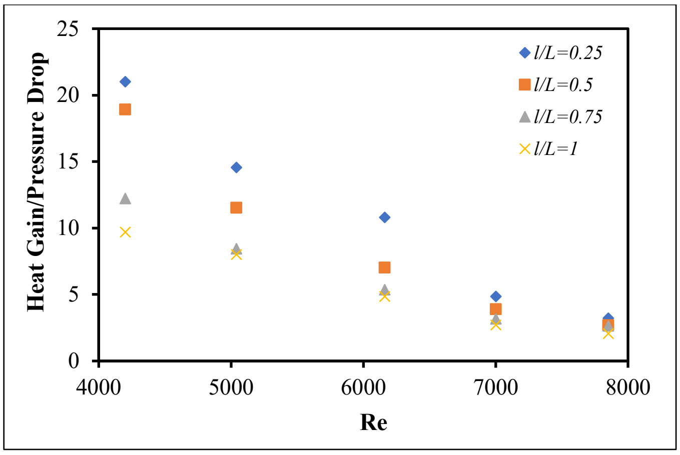

Figure 5. *Variation of (Heat Gain/Pressure Drop) for parallel flow.*

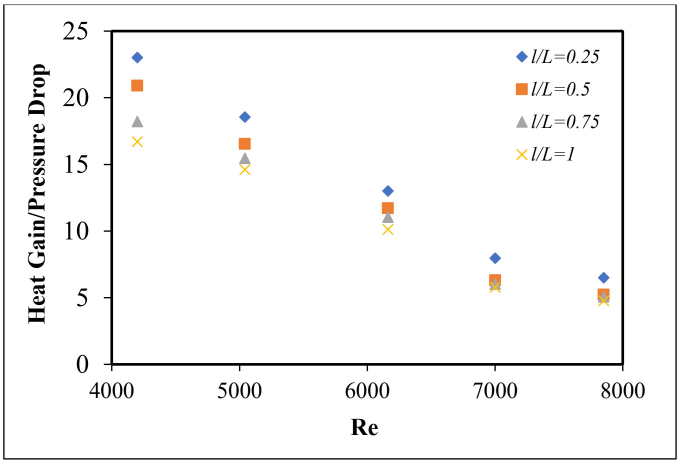

Figure 6. *Variation of (Heat Gain/Pressure Drop) for countercurrent flow.*

In Figure 6 and 7, the "heat gain/pressure drop" ratio decreases significantly for all l/L ratios as the Reynolds number increases. This trend shows that pressure losses increase significantly at high flow rates, whereas the increase in heat transfer is more limited. The highest ratio is observed for l/L = 0.25 at low Reynolds numbers. This indicates that the shorter the twisted strips offer a more favourable performance. At l/L = 0.5, 0.75, and 1, the efficiency value is lower, suggesting that longer the twisted strips lead to more pressure drop than heat gain due to increased friction losses.

Notations

A = Surface area for heat transfer (in square meters).
A_c = Surface area of the duct (in square meters)
C_p = Heat capacity per unit mass (in joules per kilogram per kelvin).
D_e = Equivalent diameter of the tube (in meters).
H = Width of twisted strips (in meters).
h = Coefficient of film heat transfer (in watts per meter-kelvin).
l= Twisted tape length (in meters).
L = Tubes length (in meters).
m = Flow rate of mass (in kilograms per second).
Nu = Dimensionless Nusselt number, representing convective heat transfer.
P = Perimeter per unit length of the duct (in meters)
Pr= Dimensionless Prandtl number, indicating fluid heat transfer characteristics.
Q = Heat transfer power (in watts).
Re = Reynolds number, expressing flow conditions in the system.
t = thickness of twisted strips (in meters)
T = Fluid temperature (in kelvin).
v = Kinematic viscosity of the fluid (in square meters per second).
V = Flow rate of fluid volume (in cubic meters per second).

Subscripts

i = inlet
c = cold
h = hot
o = outlet
w = wall

Conflict of Interest

There is no conflict of interest between the authors.

Authors Contributions

Zeki ARGUNHAN: Project administration, investigation, methodology, conceptualization, conduct practical experiments, visualization, writing – original draft, writing – review and editing.

Cengiz YILDIZ: Project administration, investigation, methodology, conceptualization, conduct practical experiments, visualization, writing – original draft, writing – review and editing.

Emin EL: Investigation, visualization, writing – original draft, writing – review and editing.

Statement of Research and Publication Ethics

The study is complied with research and publication ethics.

Artificial Intelligence (AI) Contribution Statement

This manuscript was entirely written, edited, analyzed, and prepared without the assistance of any artificial intelligence (AI) tools. All content, including text, data analysis, and figures, was solely generated by the authors.

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