HEAT TRANSFER ENHANCEMENT IN A ROUND TUBE WITH MODIFIED TWISTED TAPE INSERTS

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A THESIS SUBMITTED IN PARTIAL FULFILLMENT
OF THE REQUIREMENT FOR THE DEGREE OF
DOCTOR OF ENGINEERING IN MECHANICAL ENGINEERING
FACULTY OF ENGINEERING
KING MONGKUT'S INSTITUTE OF TECHNOLOGY LADKRABANG
2013

KMITL-2013-EN-D-058-186

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บทคัดย่อ

การเพิ่มการถ่ายเทความร้อนของเครื่องแลกเปลี่ยนความร้อนได้มีการนำมาประยุกต์และพัฒนาอย่างกว้างขวางในอุตสาหกรรมของเครื่องแลกเปลี่ยนความร้อนโดยมีความต้องการให้เครื่องแลกเปลี่ยนความร้อนมีขนาดเล็กลงกว่าเดิมร่วมถึงความต้องการจัดการลดต้นทุนและลดการใช้พลังงานเพื่อเป็นประโยชน์ต่อสิ่งแวดล้อม ในเทคนิควิธีที่หลากหลายในการเพิ่มการถ่ายเทความร้อนวิธีการใช้แผ่นบิด เป็นวิธีที่ได้รับความนิยมในการนำมาใช้ จากวิธีดังกล่าวพบว่าไม่ได้มีผลแต่เพียงการเพิ่มการแลกเปลี่ยนการถ่ายเทความร้อนเท่านั้นแต่ยังมีผลต่อการลดต้นทุนที่ดีขึ้นร่วมถึงง่ายต่อการติดตั้ง ในวิธีการใช้แผ่นบิดเพื่อเป็นอุปกรณ์เพิ่มการแลกเปลี่ยนความร้อนในเครื่องแลกเปลี่ยนความร้อนได้ส่งผลที่สำคัญจากการใช้แผ่นบิดคือ 1) การทำให้เกิดการไหลหมุนควงที่ช่วยเพิ่มการผสมของของไหลมากขึ้น และ 2) เหนี่ยวนำให้ของไหลเกิดการบิดเกลียวมากขึ้นและมีช่วงการไหลที่ระยะทางที่มากขึ้น จากผลดังกล่าวข้างต้นส่งผลโดยตรงต่อการปรับเพิ่มขึ้นของการถ่ายเทความร้อนต่อเครื่องแลกเปลี่ยนความร้อน

วิทยานิพนธ์นี้ได้ศึกษาวิจัยถึงการเพิ่มการแลกเปลี่ยนความร้อนและคุณลักษณะความดันที่สูญเสียโดยการไหลปั่นป่วนในท่อแลกเปลี่ยนความร้อนที่ได้มีการติดตั้งแผ่นบิดที่รูปร่างลักษณะหลากหลายรูปแบบเพื่อเป็นอุปกรณ์ในการสร้างการไหลหมุนควง ซึ่งมีรูปแบบดังนี้คือ 1) แผ่นบิดที่มีการเจาะรูบริเวณผนังท่อ (perforated twisted tape (*PT*)) 2) แผ่นบิดที่มีปีกผนังท่อ (delta-winglet twisted tape (*DWT*)) 3) แผ่นบิดที่มีปีกสี่เหลี่ยมบริเวณผนังท่อ (twisted tapes with rectangular wing (*T-Recs*)) 4) แผ่นบิดที่มีการเจาะรูกลางแผ่นร่วมกับปีกผนังท่อ (perforated twisted tape with parallel wing (*PTT*)) 5) แผ่นบิดคู่ที่มีการจัดวางแบบตามและทวน (twin twisted tapes in co/counter arrangement (*CoTs/CTs*)) และ 6) แผ่นบิดที่มีปีกกลางท่อและมีการปรับแกนแผ่น (wing twisted tape with alternate axis (*WT-A*)) และรวมถึงต้องการศึกษาผลจากลักษณะรูปร่างของแผ่นบิดต่อพฤติกรรมของตัวประกอบสมรรถนะความร้อน กรอบความคิดในการออกแบบรูปร่างลักษณะของแผ่นบิดมีดังนี้ 1) แผ่นบิดที่มีปีกและที่จุดปีกของแผ่นบิดจะช่วยเหนี่ยวนำให้เกิดความปั่นป่วนอย่างหนาแน่นใกล้กับผนังท่อซึ่งทำให้เกิดผลกระทบที่รบกวนต่อขอบชั้นชิดผิวความร้อน 2) รูหรือช่องโหว่ที่มีตลอดความยาวแกนกลางหรือท่อผนังจะช่วยลดความดันสูญเสียของท่อ ในการทดลองทำการสอดใส่ติดตั้งแผ่นบิดที่ได้มีการปรับดัดแปลงรูปร่างตลอดความยาวของท่อหรือในช่วงทดสอบที่อยู่ภายใต้เงื่อนไขในการควบคุมฟลักซ์ความร้อนให้สม่ำเสมอ โดยมี

อัตราส่วนการบิดแผ่นบิดที่แตกต่างกัน การประเมินหาค่าการเพิ่มการแลกเปลี่ยนความร้อนจากการเปรียบเทียบระหว่างผลของการทดลองที่ได้กับผลของท่อเปล่าหรือไม่มีการหมุนควงในท่อและจากการวิเคราะห์สหสัมพันธ์มาตรฐาน และยังทำการประเมินทดสอบกับแผ่นบิดแบบดั่งเดิม (typical twisted tapes (*TT*)) ด้วยเช่นกัน ในการทดลองได้ทำการควบคุมช่วงย่านการไหลแบบปั่นป่วนที่เลขเรย์โนลด์ระหว่าง 3000 ถึง 27,000 โดยใช้น้ำเป็นของไหลทดสอบ

ผลที่ได้อย่างชัดเจนในการสอดใส่แผ่นบิดที่ได้มีการดัดแปลงรูปร่างคือการทำให้ค่าการถ่ายเทความร้อนและตัวประกอบความเสียดทานสูงกว่าแบบท่อเปล่าและท่อที่มีการสอดใส่แผ่นบิดแบบดั่งเดิม นอกจากนั้นแผ่นบิดที่มีอัตราการบิดที่น้อยที่สุดจะส่งผลให้เกิดค่าการถ่ายเทความร้อน ค่าตัวประกอบความเสียดทานและค่าตัวประกอบสมรรถนะความร้อนสูงกว่าอัตราส่วนการบิดที่มากกว่าเนื่องจากผลของบริเวณพื้นผิวสัมผัสที่ใหญ่กว่าและระยะเวลาในการถ่ายเทความร้อนที่ยาวนานขึ้นร่วมถึงมีการไหลหมุนควงที่มีความเข้มข้นมากกว่าและการผสมกันของของไหลที่ดีกว่าซึ่งเกิดจากขอบชั้นชิดผิวความร้อนที่บางกว่า ทั้งนี้แผ่นบิดแบบ *WT-A* ได้ให้การถ่ายเทความร้อนและตัวประกอบสมรรถนะความร้อนสูงกว่าแผ่นบิดแบบอื่นๆที่ดัดแปลง โดยให้ค่าตัวประกอบสมรรถนะความร้อนสูงสุดเท่ากับ 1.4 ที่เลขเรย์โนลด์ 5337 นอกจากนี้สหสัมพันธ์ที่ได้จากการทดลองของค่าการถ่ายเทความร้อน ค่าตัวประกอบความเสียดทานและค่าตัวประกอบสมรรถนะความร้อนสำหรับการไหลแบบปั่นป่วนในท่อที่มีการใส่แผ่นบิดแบบดัดแปลงได้ถูกพัฒนาขึ้น

***คำสำคัญ*:** การเพิ่มการถ่ายเทความร้อน, เครื่องแลกเปลี่ยนความร้อน, การไหลหมุนควง, การไหลปั่นป่วน, แผ่นบิด

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Year	2013
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ABSTRACT

Enhancement of heat transfer in a heat exchanger is widely applied in industries due to the need of a compact heat exchanger, a lower operating cost, energy saving and ecological benefit. Among many heat transfer enhancement techniques, utilization of twisted tape vortex generators is a promising method. The approach possesses not only an effective heat transfer enhancement but also the advantage of a low cost and an ease of installation. Twisted tapes have been extensively used as heat transfer enhancing devices in heat exchangers. The important effects induced by the tapes are (1) swirl flow which improves fluid mixing, (2) helically twisting fluid motion which offers an effectively longer flow path, and (3) partitioning and blockage of the tube flow cross section which leads to a higher flow velocity. All the effects mentioned above are directly responsible for the improvement of heat transfer within heat exchanger.

This thesis deals with a preliminary study on heat transfer enhancement and pressure loss characteristics of turbulent flow in a heating tube equipped with various modified twisted tape swirl generators: (1) perforated twisted tape (*PT*), (2) delta-winglet twisted tape (*DWT*), (3) twisted tapes with rectangular wing (*T-Recs*), (4) perforated twisted tape with parallel wing (*PTT*), (5) twin twisted tapes in co/counter arrangement (*CoTs/CTs*), and (6) wing twisted tape with alternate axis (*WT-A*)). The effect the tape geometries on the thermal performance factor behaviors is also examined. The design of modified twisted-tapes involves the following concepts: (1) wings induce an extra turbulence near tube wall and thus efficiently disrupt a thermal boundary layer (2) holes existing along a core or wall tube, diminish pressure loss within the tube. In the experiments, modified twisted-tapes were inserted along the test section under uniform heat flux conditions at different twist ratios. Heat transfer enhancement is evaluated by comparing the present experimental results with the results of present plain tube (non-swirl flow) and also those obtained from standard correlations. The typical twisted tapes (*TT*) were also tested for evaluation.

The experiments were conducted in a turbulence region with Reynolds numbers ranging from 3000 to 27,000 using water as the test fluid.

Evidently, the tube with all of modified twisted-tapes consistently possesses higher heat transfer, friction factor and thermal performance than the plain tube and typical twisted tapes (*TT*). The tape with the smaller twist ratio gives higher heat transfer rate, friction factor as well as thermal performance factor than the one with larger twist ratio as a result of a larger contact surface area, longer residence time, stronger swirl intensity and thus better fluid mixing which leads to a thinner thermal boundary layer. In addition, the *WT-A* provide better heat transfer and thermal performance factor than those other modified tapes. Maximum thermal performance factor achieved by the use of *WT-A* at Re=5337 is 1.4, which is higher than achieved by the use of other modified tape inserts. Empirical correlations of the heat transfer, friction factor and thermal performance in turbulent regime for tubes with modified twisted-tapes were also developed.

Keywords: Heat transfer enhancement, Heat exchanger, Swirl flow, Turbulent flow, Twisted tape

To My Family

ACKNOWLEDGMENTS

On this very special occasion, May I take the privilege of expressing my deepest appreciation and my warmest heartfelt thanks for the invaluably distinctive generous advice and the instruction of these honorable benefactors:

My thesis advisor:	Assoc. Prof. Dr. Chinaruk Thianpong
My thesis co advisor:	Assoc. Prof. Dr. Smith Eiamsa-ard
My thesis counselor's staff:	Assoc. Prof. Dr. Pongjet Promvonge
	Assoc. Prof. Dr. Jaruwat Charoensuk
	Col. Assoc. Prof. Dr. Anotai Suksangpanomrung
	Asst. Prof. Dr. Nuthvipa Jayranaiwachira

With their benevolent deeds, my Doctoral degree research work studies can meet accomplishments. I hereby respectfully express my greatest gratitude's for their immeasurably incredible instruction extensively enlightening my wisdom, knowledge, conceptual viewpoint, vision, creative intelligence, the engineering science and worldwide experiences beneficial to my career work studies in the ages of the speedy changing Hi-technology world. My most appreciative thanks for every of their efforts fulfilling my work studies and my life exists in my happy heart and remembrance permanently.

I always realize that my qualification developed by their benevolent deeds enables creating my potentialities at full capacity of being one of the successful participants serving Thailand our beloved country reaching the hope of the nation: happiness, prosperity, progress, wealth, improvements and the future development of our country.

Also I would like to express my warmest thanks and appreciation my friends' help and everyone of my University Doctoral and Master Degree program friends who gave me a lot of will power, suggestion and encouragement on doing this thesis.

More importantly, I also would like to express my warmest thanks and appreciation the Energy Policy and Planning Office, Ministry of Energy, Thailand (EPPO) for financial support of this research.

May the Triple Gems and all of the Holy things in the Universe bless everyone of them with excellent life, good health, Wealth, Wisdom, success through their lives.

CONTENTS

CONTENTS (continued)

LIST OF TABLES

LIST OF FIGURES

LIST OF FIGURES (continued)

Figure **Page**

LIST OF FIGURES (continued)

LIST OF FIGURES (continued)

LIST OF FIGURES (continued)

NOMENCLATURE

A	Heat transfer surface area, m^2
C_p	Specific heat of fluid, J kg^{-1} K^{-1}
D	Inner diameter of the test tube, m
d	Diameter of hole at the core tape, mm
f	Friction factor $=\Delta P/((L/D)(\rho U^2/2))$
h	Heat transfer coefficient, W m^{-2} K^{-1}
I	Current, A
k	Thermal conductivity of fluid, W m^{-1} K^{-1}
L	Test length, m
M	Mass flow rate, kg s^{-1}
Nu	Nusselt number = hD/k
ΔP	Pressure loss, Pa
Pr	Prandtl number = $\mu C_p/k$
Q	Heat transfer rate, Watt
Re	Reynolds number = $\rho UD/\mu$
t	Thickness of the test tube, m
T	Temperature, K
$\tilde{T}$	Mean temperature, K
U	Average velocity, m s^{-1}
V	Voltage, V
W	Twisted tape width, m
w	Wing depth, mm
y	Twisted tape pitch, m

Greek Symbols

ρ	Fluid density, kg m^{-3}
δ	Twisted tape thickness, m
μ	Fluid dynamic viscosity, kg s^{-1} m^{-1}
η	Thermal performance factor

Subscripts

b	Bulk
conv	Convection
in	Inlet
out	Outlet

Subscripts

p Plain tube

Abbreviations

PT	perforated twisted tape
TT	typical twisted tape
DWT	delta-winglet twisted tape
T-Recs	twisted tapes with rectangular wing
PTT	perforated twisted tape with parallel wing
CoTs/CTs	twin twisted tapes in co/counter arrangement
WT-A	wing twisted tape with alternate axis

CHAPTER 1

Introduction

1.1 Heat transfer enhancement

A high cost of energy and material has resulted in an increased effort aimed at producing more efficient heat exchange equipment. The methods of improving convective heat transfer in the tubes of heat exchangers have been widely investigated by many researchers. Heat transfer enhancement techniques can be classified into two groups: (1) active technique that needs external power source and (2) passive technique which does not need external power source. Both active and passive techniques have been applied to improve heat transfer in several areas such as nuclear reactors, chemical reactors and general heat exchangers. The principle of the passive technique involves either surface treatment, such as coated surface, rough surface and extended surface or flow manipulation such as swirl flow and modified flow. One of the most favorable passive techniques is generating swirl flow by insertion of a twisted tape (Figure 1.1) because the tape is inexpensive and can be easily employed to the existing system. The effects of twisted tape insertion have been widely studied for both experimental and numerical simulation works. The presence of twisted tape directs toward reducing the hydrodynamic or thermal boundary layer thickness, leading to greater convective heat transfer. However, in the process, pumping power may increase significantly and ultimately the pumping cost becomes high. Therefore, to achieve a desired heat transfer rate in an existing heat exchanger at an economic pumping power, the design of twisted tape with a proper geometry is necessary.

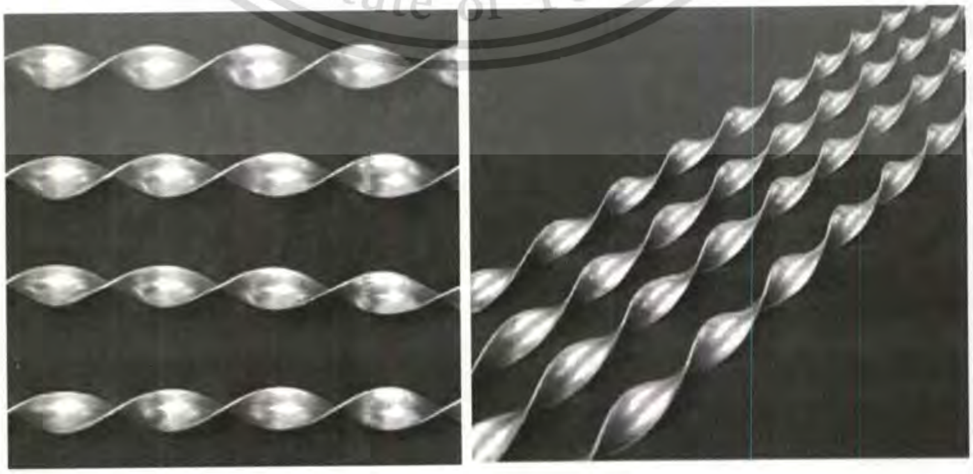


Figure 1.1: Pictorial view of twisted tape swirl generators.

1.2 Swirl flow devices

For almost a century, the swirl flow devices have been used for enhancing heat transfer rate in heat exchangers. It is regarded that the swirl tangential velocity increases the composite velocity, thins the boundary layer, enhances the tangential and radial turbulent fluctuation, and therefore causes the increase in pressure drop and heat transfer inside a tube. Generally, the swirl flow in pipes can be classified into two major types: the continuous swirl flow and the decaying swirl flow. In continuous swirl flow, the swirling motion persists over the whole length of the pipe (Figure 1.2) while in decaying swirl flow, the swirl is generated at the entry section of the pipe and decays along the flow path (Figure 1.3). It is well known that the twisted tape is one of the most widely used devices for producing compact heat exchangers and upgrading the thermal performance of the existing heat exchanger due to its low cost and ease of manufacture installation. Twisted tape brings several mechanisms for heat transfer enhancement including: (1) augmenting flow velocities caused by partial blockage of the tube flow cross section (2) promoting heat transfer coefficient as a result of hydraulic diameter reduction (3) lengthening flow path in consequence of a helically twisting fluid motion (4) improving fluid mixing by secondary fluid motion and (5) possible fin effect by a metallic tape with a good contact with a tube wall. Regarding to its promising characteristics, twisted tape inserts have been applied in several researches for heat transfer improvement. The performances of twisted tape swirl generator have been intensively investigated by many researchers with different tape geometries (*e.g.* broken, serrated, perforated, notched and jagged), working fluid types (*e.g.* water, air, oil, servotherm medium oil, turbine oil, ethylene glycol, nitrogen, and R134a), wall conditions (uniform heat flux and constant wall temperature) and flow regimes (laminar, transition and turbulent flows). Regarding to the results reported in the mentioned.

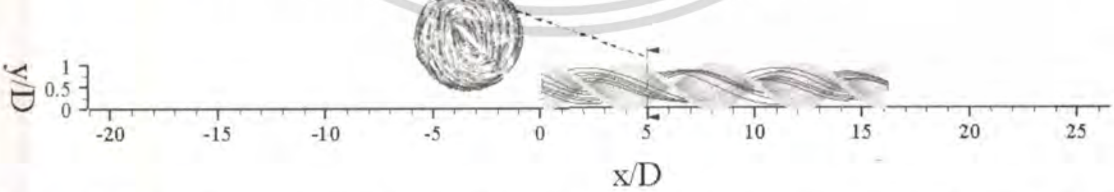


Figure 1.2: Continuous swirl flow produced by twisted tape fitted along the test tube (Seemawute and Eiamsa-ard; 2012).

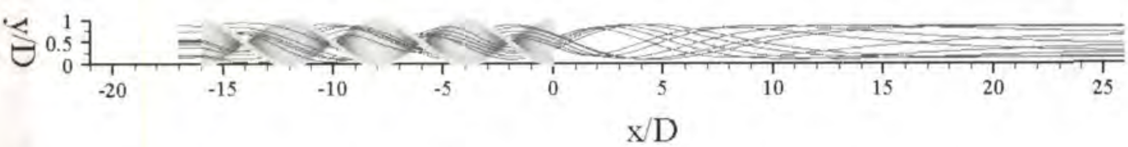


Figure 1.3: Decaying swirl flow produced by twisted tape fitted at the entry test tube (Seemawute and Eiamsa-ard; 2012).

1.3 Objectives and Scopes

1.3.1 Effects of the *typical twisted tape* on heat transfer, friction factor and thermal performance factor characteristics in a uniform wall heat flux tube are presented in this thesis. In the experiments, typical twisted tapes (*TTs*) at three different twist ratios (y/W = 3, 4 and 5) were inserted along the test section using water as the working fluid.

1.3.2 Effects of the *perforated twisted tapes* on heat transfer, friction factor and thermal performance factor characteristics are proposed. In the experiments, the holes are generated with prospect to promote fluid turbulence near the tube wall and the tape edge. The geometrical characteristics of perforated twisted tapes subjected to the present study are (1) twist ratios, y/W = 3, 4 and 5, (2) perforation hole diameter ratio, d/W = 0.11, 0.14 and 0.17, and (3) space-length ratio, s/W = 0.4, 0.6 and 0.8.

1.3.3 Effects of the *delta-winglet twisted tape* on heat transfer, friction factor and thermal performance factor characteristics are reported. In the experiments, the oblique delta-winglet twisted-tape and straight delta-winglet twisted tape arrangements with different (1) twist ratios, y/W = 3, 4 and 5, and (2) depth of wing cut ratios, d/W = 0.11, 0.21 and 0.32, are performed.

1.3.4 Effects of the *twisted tape with rectangular-wings and alternate-axes* on heat transfer, friction factor and thermal performance factor characteristics are studied. In the experiments, the twisted tapes with rectangular-wings for different wing depth ratios, d/W = 0.1, 0.2 and 0.3, are examined.

1.3.5 Effects of the *perforated twisted tape with parallel wing* on heat transfer, friction factor and thermal performance factor characteristics are conducted. In the experiments, the *perforated twisted tape with parallel wing* with different hole diameter ratios, d/W = 0.11, 0.33 and 0.55, and wing depth ratios, w/W = 0.11, 0.22 and 0.33, are performed.

1.3.6 Effects of the *twin counter/co twisted tapes* (counter/co-swirl tape) on heat transfer, friction factor and thermal performance factor characteristics are examined. In the present work, the twin counter twisted tapes (*CTs*) are used as the counter-swirl flow generators while twin co twisted tapes (*CoTs*) are used as co-swirl flow generators. Both twin counter/co twisted tapes are prepared with four different twist ratios, y/W = 2.5, 3.0, 3.5 and 4.0.

1.3.7 Effects of the *twisted tape consisting of centre wings and alternate-axes* on heat transfer, friction factor and thermal performance factor characteristics are reported. In the present work, the twisted tape with wings with different attack angles of the wings are varied at 43°, 53° and 74°, respectively.

1.4 The expected results

To improve the heat transfer rate in a heat exchanger with swirl generator device which can reduce the cost and size of the heat exchanger than can increasing the thermal performance and save the energy.

1.5 Outlines of thesis

The content of the thesis is consists of five chapters. The introduction, objective, hypothesis and thesis outlines are presented in chapter 1. In chapter 2, a reviewed of the past investigated with swirl flow by swirl flow by twisted tape combined with several enhancement devices and swirl flow by .modified twisted tape devices is reported. Chapter 3 describes the experimental facility. Chapter 4 presents the results and discussion of the heat transfer, friction factor and thermal performance behaviors in a heat exchanger tube with modified twisted tape devices. Finally, chapter 5 describes the main conclusions of the results obtained in the present thesis study and the future work.

1.6 Publications by the author relating to the work contained in this thesis

1.6.1 International journals

1.6.1.1 Chinaruk Thianpong, **Petpices Eiamsa-ard** and Smith Eiamsa-ard, Heat transfer and thermal performance characteristics of heat exchanger tube fitted with perforated twisted-tapes, ***Heat and Mass Transfer, Wärme-und Stoffübertragung***, Volume 48, Number 6, pp. 881-892, 2012.

1.6.1.2 Chinaruk Thianpong, **Petpices Eiamsa-ard**, Pongjet Promvonge and Smith Eiamsa-ard, Effect of perforated twisted-tapes with parallel wings on heat transfer enhancement in a heat exchanger tube, ***Energy Procedia***, Volume 14, pp. 1117-1123, 2012.

1.6.1.3 Smith Eiamsa-ard, Khwanchit Wongcharee, **Petpices Eiamsa-ard** and Chinaruk Thianpong, Heat transfer enhancement in a tube using delta-winglet twisted tape inserts, ***Applied Thermal Engineering***, Volume 30, Number 4, pp. 310-318, 2010.

1.6.1.4 Smith Eiamsa-ard, Chinaruk Thianpong and **Petpices Eiamsa-ard**, Turbulent heat transfer enhancement by counter/co-swirling flow in a tube fitted with twin twisted tapes, ***Experimental Thermal and Fluid Science***, Volume 34, Number 1, pp. 53-62, 2010.

1.6.1.5 Smith Eiamsa-ard, Khwanchit Wongcharee, **Petpices Eiamsa-ard** and Chinaruk Thianpong, Thermohydraulic investigation of turbulent flow through a round tube equipped with twisted tapes consisting of centre wings and alternate-axes, ***Experimental Thermal and Fluid Science***, Volume 34, Number 8, pp. 1151-1161, 2010.

1.6.2 International conferences

1.6.2.1 **Petpices Eiamsa-ard**, Chinaruk Thianpong and Smith Eiamsa-ard, Turbulent heat transfer in a circular tube fitted with twisted-tape swirl generator, ***International Symposium on Technology for Sustainability (ISTS)***, January 26-29, 2012, Bangkok, Thailand.

1.6.2.2 **Petpices Eiamsa-ard**, Chinaruk Thianpong and Smith Eiamsa-ard, Heat transfer enhancement in a heat exchanger tube fitted with twisted-tape consisting of triangular-wing and alternate-axis, ***International Symposium on Technology for Sustainability (ISTS)***, January 26-29, 2012, Bangkok, Thailand.

1.6.2.3 Chinaruk Thianpong, **Petpices Eiamsa-ard**, Pongjet Promvonge and Smith Eiamsa-ard, Effect of perforated twisted-tapes with parallel wings on heat transfer enhancement in a heat exchanger tube, ***2^{nd} International Conference on Advances in Energy Engineering (ICAEE 2011)***, December 27-28, 2011, Bangkok, Thailand.

1.6.2.4 **Petpices Eiamsa-ard**, Chinaruk Thianpong and Smith Eiamsa-ard, Influences of twisted-tape with parallel rectangular-wing on thermal performance of a heat exchanger, ***International Conference & Utility Exhibition on Power and Energy Systems: Issues and Prospects for Asia (ICUE 2011)***, September 28-30, 2011, Pattaya, Thailand.

CHAPTER 2
Literature review

2.1 Introduction

Heat exchangers are essential facilities in chemical plants for heat transfer processes such as heating cold feedstock with hot products (regenerative heat exchanger), cooling chemical solutions to remove reaction heat, maintaining optimal temperatures in wastewater treatment units, etc. In common heat exchanger, the limitation of mass and heat transfer coefficients is a major problem that leads to a large consumption of materials for manufacturing and also high operating cost. For more compact and economic heat exchanger, heat transfer enhancement becomes an important issue.

Swirl flow is a rotational flow around an axis; its characteristic feature is adopted in many engineering devices such as cyclone separators, vortex tubes, agricultural spraying machines, gasoline engines, diesel engines, gas turbines, furnaces, vortex dust collectors and many other practical heating devices. Several studies have been reported on heat transfer enhancement in both active and passive methods. Passive method can improve the heat transfer rate by using the enhancement devices that do not need any extra power source which the swirl device is one of the most important of this group.

Twisted tapes belonging to one important group of swirl generators are mostly applied in heat transfer improvement. The mechanisms of heat transfer enhancement due to a twisted tape include (1) a reduction of a hydraulic diameter causing resulting in an increase of flow velocity and curvature which in turn increases the shear stress at the wall and drives secondary motions, (2) an increase of fluid mixing between the core and the near-wall flow regions and (3) an extra heat transfer through thermal contact, probably. Twisted tape insert is one of the favorable devices for heat transfer enhancement due to its good performance, installing simplicity and low cost. Like other enhancing devices, the insert involves both heat transfer enhancement and an increase in pressure drop which is directly related to energy consumption. The net effectiveness of twisted tape insert can be improved by optimizing these two effects with proper modifications of the insert. Numerous researches on heat transfer enhancement using modified twisted tape inserts have been reported. In general, the performances of modified twisted tapes were evaluated by comparing their results with those of an unmodified (typical) one.

2.2 Twisted tape

In general, heat transfer enhancement associated with the uses of typical twisted tapes (*TTs*), is accompanied by an increase of friction penalty. The evaluation of overall performance of twisted tape is usually reported in term of thermal performance factor which is based on the trade off between both increased heat transfer and friction factor. To reduce friction (or pressure drop) in the heat exchanger as compared to those given by typical twisted tapes, several modified twisted tapes have been proposed. However, most of them brought the heat transfer rate down significantly resulting in even poorer tradeoff between the increased heat transfer and friction factor than those provided by the typical one.

Al-Fahed *et al.* (1998) studied the heat transfer characteristics in tube fitted with tight-fit tapes (typical twisted tapes) and loose-fit tapes (modified twisted tapes) at various twisted ratios. Their results revealed that the tight-fit tape gave a better performance over the loose-fit tape. However, the loose-fit tape with small twist ratio was recommended due to its low pressure drop penalty as well as its ease of installation and removal for cleaning purposes. Patil (2000) examined the heat transfer and pressure drop characteristics in a tube fitted with twisted tape at varying tape-width. Saha *et al.* (2001) numerically studied the heat transfer and flow friction characteristics of laminar swirl flow through a round tube fitted with regularly spaced twisted-tape elements (Figure 2.1).

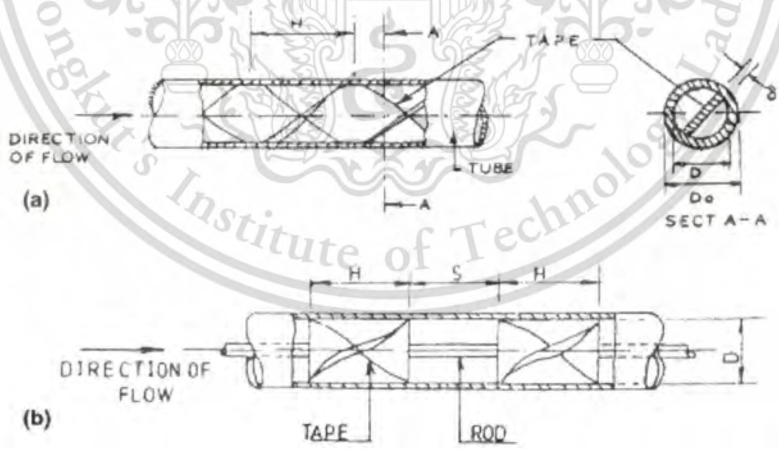


Figure 2.1: Pictorial view of circular tube containing (a) full-length twisted-tape and (b) regularly spaced twisted-tape elements (Saha *et al.*; 2001).

Eiamsa-ard *et al.* (2006) also reported the effect of the regularly spaced twisted tape elements on the heat transfer enhancement and friction factor in a heat exchanger (Figure 2.2). They found that the heat transfer coefficient decreased with twist ratio (y). Whereas the decrease in the free space ratio would improve both the heat transfer coefficient and friction factor.

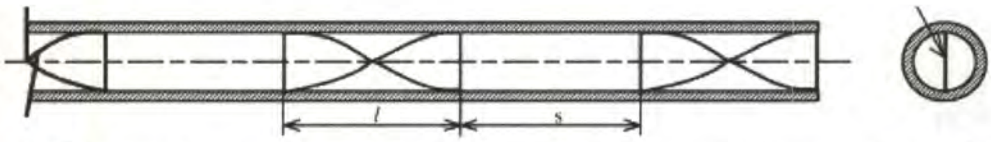
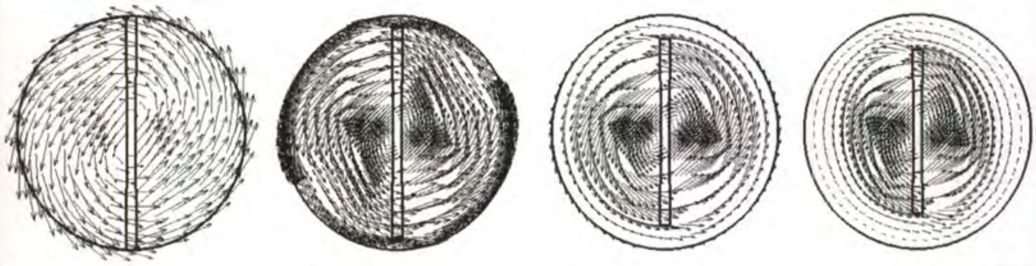


Figure 2.2: Pictorial view of regularly spaced twisted tape (Eiamsa-ard *et al.*; 2006).

Jaisankar *et al.* (2009) reported the heat transfer and friction factor characteristics of thermosyphon solar water heater system with full-length twist, twist fitted with rod and spacer fitted at the trailing edge for lengths of 100, 200 and 300 mm for twist ratio 3 and 5 (Figure 2.3). They found that the decrease in heat transfer for full length helical twist compared to twist fitted with rod is minimum and is quite significant for twist with spacer. But the decrease in friction factor is maximum in twist fitted with spacer compared to twist fitted with rod. The over all performance for twist fitted with rod was found to be better than twist fitted with spacer.

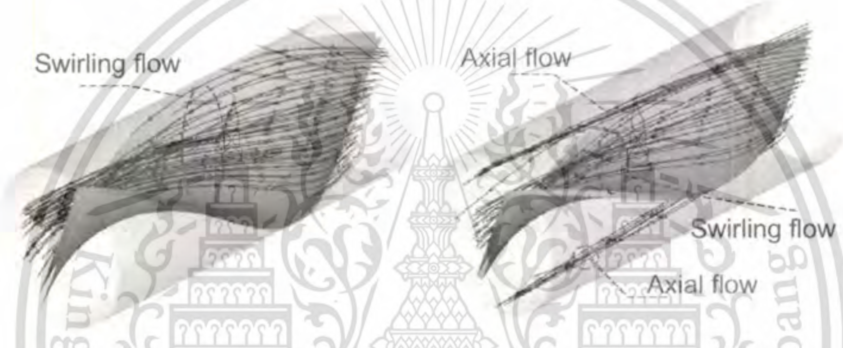
Figure 2.3: Pictorial view of helical inserts with various rod & spacer length (Jaisankar *et al.*; 2009).

Eiamsa-ard *et al.* (2009a) reported the effect loose-fit twisted tape insertion (Figures 2.4 and 2.5) at difference clearance ratios (c/D = 0.0 (tight-fit), 0.1, 0.2 and 0.3) on the heat transfer enhancement, friction factor and thermal performance factor. They found that the twisted tape inserts for y/w = 2.5 with CR = 0.0 (tight-fit), 0.1, 0.2 and 0.3 can enhance heat transfer rates respectively, up to 73.6%, 46.6%, 17.5% and 20% and increase friction factors respectively, up to 330%, 262%, 189%, and 160%, in comparison with those of the plain tube. The tube with loose-fit twisted tape insert for CR = 0.1, 0.2 and 0.3 provide heat transfer enhancement around 15.6%, 33.3% and 31.6% lower than those for CR = 0.0 (the tight-fit twisted tape). The heat transfer augmentation is expected to involve the swirl flow formation between the tape and a tube wall. In addition, the simulation for thermal performance factor of a tube with the loose-fit twisted tape and the tight-fit twisted tape under the same pumping power was also conducted, for comparison.



(a) CR = 0.0 (b) CR = 0.1 (c) CR = 0.2 (d) CR = 0.3

Figure 2.4: Vector plots of velocity at different *CR*s (Eiamsa-ard *et al.*; 2009a).



(a) CR = 0.0 (b) CR = 0.3

Figure 2.5: Contour plots of streamline at different clearance ratios (Eiamsa-ard *et al.*; 2009a).

Eiamsa-ard *et al.* (2009b) investigated the mean Nusselt number, friction factor and thermal performance factor characteristics in a round tube with short-length twisted tape insert (Figure 2.6). The full-length twisted tape was inserted into the tested tube at a single twist ratio of y/w = 4.0 while the short-length tapes mounted at the entry test section were used at several tape length ratios ($LR = l_s / l_f$) of 0.29, 0.43, 0.57 and 1.0 (full-length tape). The short-length tape is introduced as a swirling flow device for generating a strong swirl flow at the tube entry before decaying along the tube. On the other hand, the full-length tape (LR =1.0) is expected to produce a strongly swirling flow over the whole tube. They indicated that the short-length tapes of LR = 0.29, 0.43 and 0.57 perform lower heat transfer and friction factor values than the full-length tape around 14%, 9.5% and 6.7%; and 21%, 15.3% and 10.5%, respectively. In addition, it is apparent that the thermal performance factor of the tube with the short-length tape insert was found to be lower than that with the full-length one.

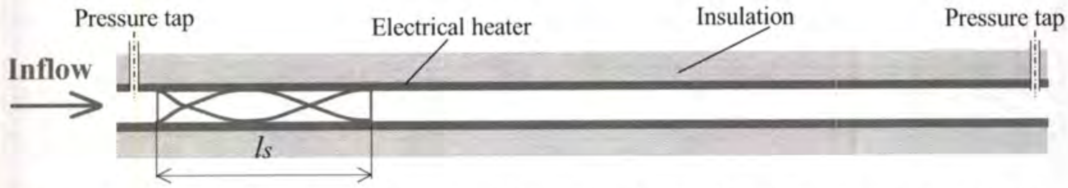


Figure 2.6: Pictorial view of short-length twisted tape (Eiamsa-ard *et al.*; 2009b).

Wang *et al.* (2011) presented the influence of the regularly spaced short-length twisted tape with effects of twist ratio and rotated angle, (Figure 2.7) on turbulent heat transfer using computational fluid dynamic (CFD) technique. The tape with larger rotated angle provided higher heat transfer and flow friction. However, the good overall performance was achieved with the use of the one with small rotated angle, small twist ratio and low Reynolds number.

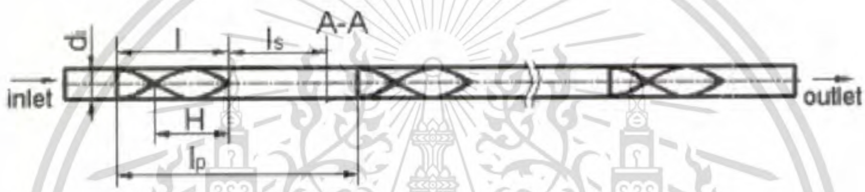


Figure 2.7: Pictorial view of regularly spaced short-length twisted tape (Wang *et al.*; 2011).

Guo *et al.* (2011) numerically studied the effect of the center-cleared/short-width twisted tape (Figure 2.8) on heat transfer enhancement and thermal performance behaviors in laminar region. They found that the heat transfer and thermal performance were weakened by cutting off the tape edge (tubes with short-width twisted tape inserts). On the other hand, the heat transfer and thermal performance could be effectively enhanced using the center-cleared twisted tape with suitable central clearance ratio. The thermal performance factor of the tube with center-cleared twisted tape was increased up to 20% higher than that of the one with typical twisted tape (TT).

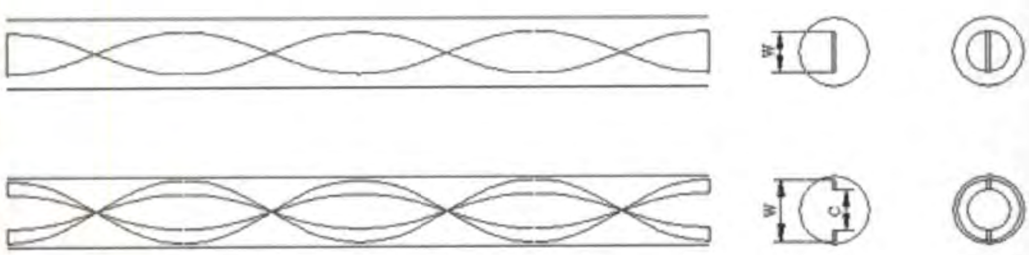


Figure 2.8: Pictorial view of short-width and center-cleared twisted tapes (Guo *et al.*; 2011).

Eiamsa-ard and Seemawute (2012a) presented the local heat transfer coefficient and flow characteristics of decaying turbulent swirl flow generated by short-length twisted tapes (Figure 2.9). They found that the local Nusselt numbers decrease with increasing axial distance (*x/D*) due to the decaying effect. Although, short-length twisted tapes consistently provide poorer heat transfer than *TTs*, the short-length twisted tapes with *y/W* = 4 and 5 yield superior thermal performance factors to the *TTs* at the same twist ratios, for Reynolds numbers beyond 10,000 due to the prominent effect of heat transfer improvement over that of the increase of friction factor.

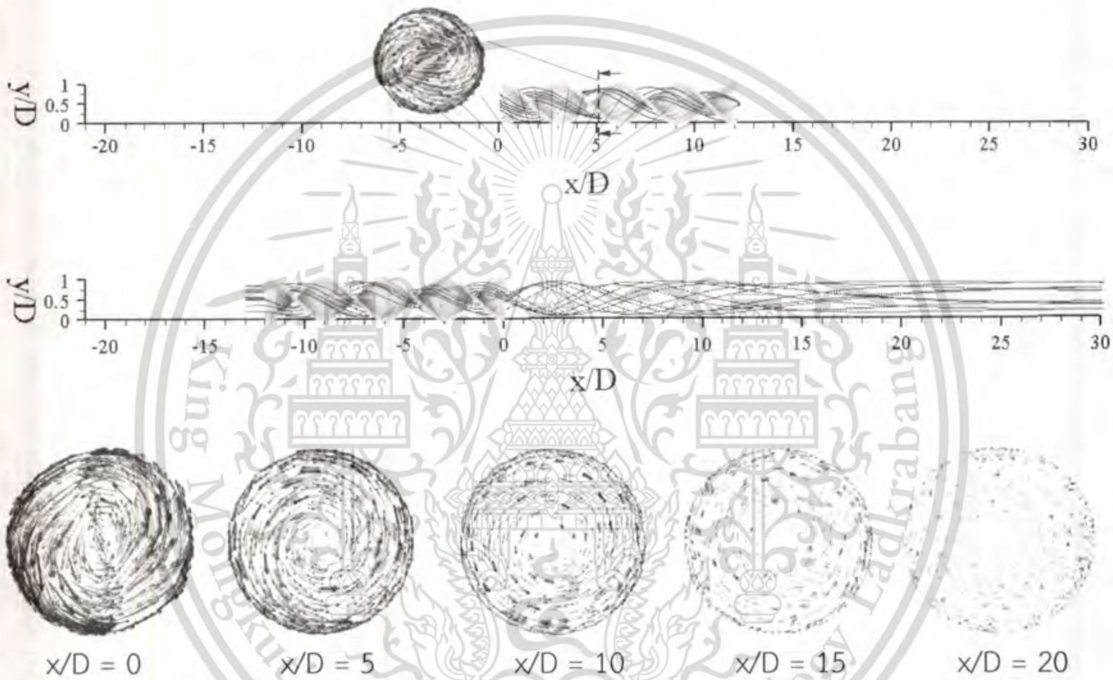


Figure 2.9: Vector plots of decaying velocity at different stations (Eiamsa-ard and Seemawute; 2012a)

Although all modified twisted tapes mentioned above generated lower pressure drop than typical twisted tape, they yielded considerably lower heat transfer rate, resulting in diminishment of overall heat transfer performance. The results are responsible by a lower swirl intensity/smaller swirling number produced by the modified twisted tapes as compared to those generated by typical ones.

2.3 Modified twisted tape

There have been attempts to improve overall heat transfer enhancement performance by different design of twisted tape, with aspect to promote fluid mixing and thus heat transfer rate. Several alternative approaches have been made to induce stronger turbulence intensity in the vicinity of tube wall or the core of tube by modifications of edges or center of twisted tapes. Chang *et al.* (2007a) examined the heat transfer and friction factor and thermal performance factor characteristics in a heat exchanger tube fitted with serrated twisted tape (Figure 2.10). They observed that heat transfer attributed to the serrated twisted tape was around 1.25 to 1.67 times of tube fitted with typical twisted tape. However, the thermal performance of the tube with serrated twisted tape was lower than the one with the typical twisted tape due to the high pressure loss penalty. Chang *et al.* (2007b) studied the effect of the broken twisted tape (Figure 2.11) on the heat transfer and pressure drop characteristics in a circular tube. In spite of its superior heat transfer rate, the broken twisted tape gave higher thermal performance factor than the typical one.

Figure 2.10: Pictorial view of serrated twisted tape (Chang *et al.*; 2007a).

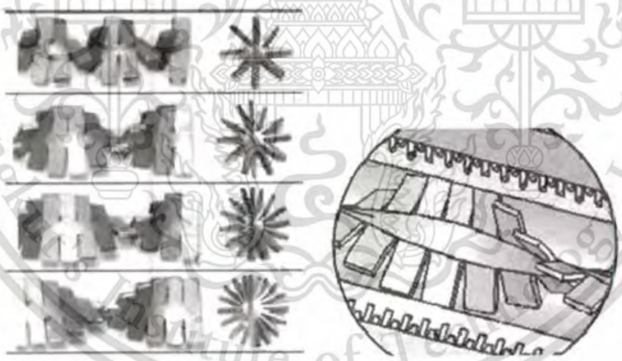


Figure 2.11: Pictorial view of broken twisted tape (Chang *et al.*; 2007b).

Sivashanmugam and Nagarajan (2007) studied the heat transfer enhancement in a heat exchanger tube fitted with right-left helical screw tape in laminar flow region. He found that the tube with helical screw tape provided higher heat transfer rate and pressure drop than the plain tube for all Reynolds number studied. Again, Sivashanmugam and Suresh (2007) carried out the heat transfer, friction factor and thermal performance behaviors in a heat exchanger tube equipped with right-left helical screws of equal and unequal length of different twist ratios. They observed that the heat transfer enhancement for right-left helical screw inserts is higher than those the straight helical twist tape and plain tube.

Rahimi *et al.* (2009) presented the effect of the jagged/perforated/notch twisted tape (Figure 2.12) on the heat transfer, friction factor and thermal performance factor. They results showed that the jagged twisted tape offered superior heat transfer rate and thermal performance factor to other tapes including other tape the typical one. The enhanced heat transfer and thermal performance by the jagged twisted tapes were 31% and 22% compared to those of the typical one.

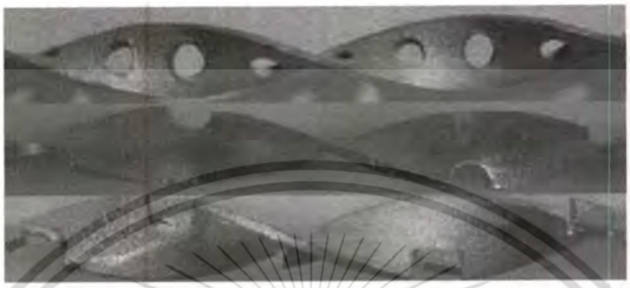


Figure 2.12: Pictorial view of perforated/notched/jagged twisted tape (Rahimi *et al.*; 2009).

Jaisankar *et al.* (2009) evaluated the feasibility of using typical twisted tapes at various twist ratios (Figure 2.13) in a solar water heater by determining heat transfer, friction factor and thermal performance. Results conclude that, heat transfer and pressure drop are higher in twisted tape collector compared to the plain one. Among the various twist ratios, the minimum twist ratio 3 was found to enhance the heat transfer and pressure drop due to swirl generation. As the twist ratio increases, the swirl generation decreases and minimizes the heat transfer and friction factor.

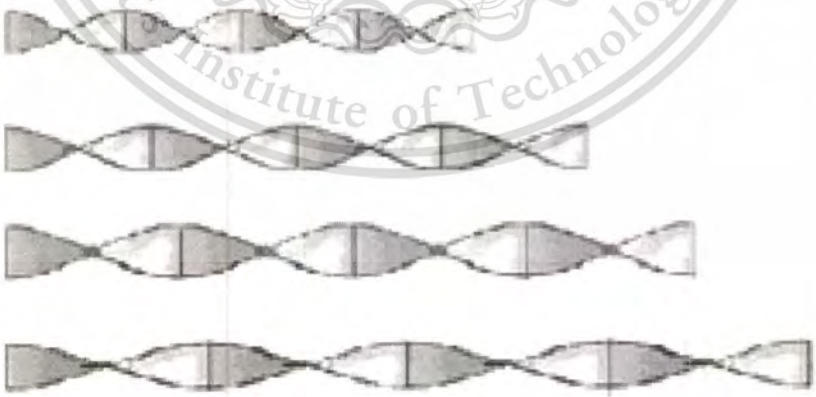


Figure 2.13: Pictorial view of typical twisted tapes at various twist ratios (Jaisankar *et al.*; 2009).

Eiamsa-ard and Promvonge (2010a) investigated the turbulent heat transfer and flow friction characteristics in a circular tube equipped with alternate clockwise and counterclockwise twisted tapes (C-CC twisted tapes) with three twist angles of 30°,

60° and 90° (Figure 2.14). They found that the *C-CC* twisted-tapes provide higher heat transfer rate, friction factor and heat transfer enhancement index than the typical twisted-tapes at similar operating conditions. The results also show that the heat transfer rate of the *C-CC* tapes increases with the decrease of twist ratio and the increase of twist angle values. Depending on Reynolds number, twist ratio and twist angle values, the mean Nusselt numbers in the tube fitted with the C-CC twisted tapes are higher than those with the *TTs* around 12.8 to 41.9%.

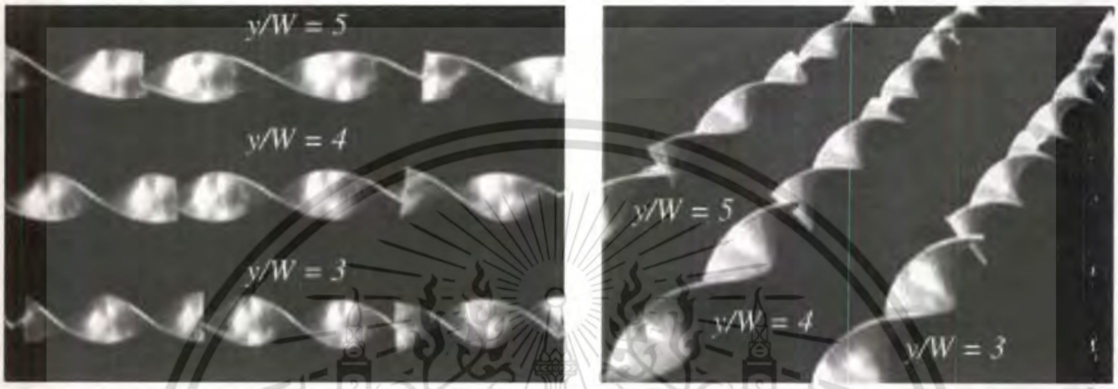


Figure 2.14: Pictorial view of alternate clockwise and counterclockwise twisted tape (Eiamsa-ard and Promvonge; 2010a).

Yong-Zhang and Mao-Cheng (2010) presented the numerical and experimental heat transfer and pressure drop characteristics of a tube fitted with edgefold twisted tape (*ETT*). As found, the tube fitted with *ETT* provided higher heat transfer rate and friction factor than that typical twisted tape (*TT*) due to the higher tangential and asymmetrical velocities. the Nusselt number of the tube with *ETT* inserts was found to be 3.9%-9.2% higher than that with *TT* inserts, and the friction factor of the tube with *ETT* inserts was 8.7%-74% higher than that of *TT* inserts.

Murugesan *et al.* (2010a) examined the effect of the twisted tape consisting wire nails (*WN-TT*) on heat transfer rate, friction factor and thermal performance factor characteristics in a double pipe heat exchanger. They observed that the tube equipped with *WN-TT* gave greater heat transfer, pressure drop and thermal performance that the typical twisted tape (*TT*) due to combined effects of the common swirling flow generated by the *TT* and additional turbulence offered by the wire nails. Over the range considered Nusselt number, friction factor and thermal enhancement factor in a tube with *WN-TT* are respectively, 1.08 to 1.31, 1.1 to 1.75 and 1.05 to 1.13 times of those in tube with *TT*. Again, Murugesan *et al.* (2010b, 2011) also investigated the heat transfer, friction factor and thermal enhancement of a double pipe heat exchanger with V/square-cut twisted tape inserts (*VTT/STT*) (Figures 2.15 and 2.16). They reported that a common swirl flow coupled with an extra fluid disturbance and secondary flow in the vicinity of the tube wall induced by the *VTT/STT* resulted in superior heat transfer enhancement as compared to only common swirl flow associated by the use of the typical twisted tape (*TT*). Over the range considered, the Nusselt number, friction factor and thermal performance factor in a tube with *STT* were respectively, 1.03 to 1.14, 1.05 to 1.25 and 1.02 to 1.06 times of those in tube with *TT*.

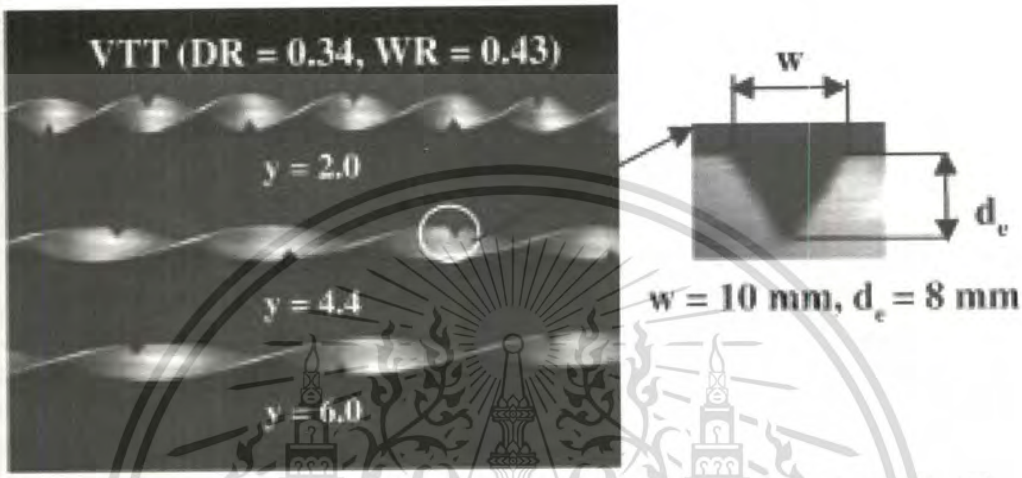


Figure 2.15: Pictorial view of V-cut twisted tapes (Murugesan *et al.*; 2010b).

Figure 2.16: Pictorial view of square-cut twisted tapes (Murugesan *et al.*; 2011).

Eiamsa-ard and Promvonge (2010b) investigated of the effect of twisted tape with serrated edge insert (Figure 2.17) on heat transfer and pressure loss behaviors. Two geometry parameters of the *STT* were the serration width ratio and the serration depth ratio. They found that the heat transfer rate in terms of Nusselt number, *Nu* increases with the rise in the depth ratio but decreases with raising the width ratio. The heat transfer rate is up to 72.2% and 27% relative to the plain tube and the *TT* inserted tube, respectively. The use of the *STT* leads to higher heat transfer rate and friction factor than that of the *TT* for all cases. The thermal performance factor of the *STT* tube under constant pumping power was evaluated and found to be above unity indicating that using the *STT* tube is advantageous over the *TT* tube or the plain tube.

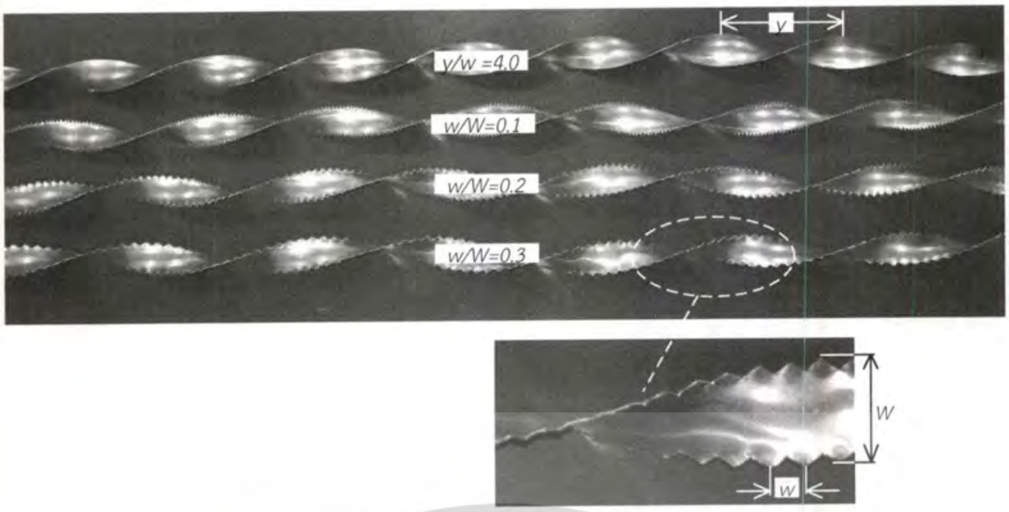


Figure 2.17: Pictorial view of serrated twisted tape (Eiamsa-ard and Promvonge; 2010b).

Eiamsa-ard *et al.* (2010c) performed the effects of peripherally-cut twisted tape insert (Figure 2.18) on heat transfer, friction loss and thermal performance factor characteristics. They found that both heat transfer rate and friction factor in the tube equipped with the peripherally-cut twisted tapes are significantly higher than those in the tube fitted with the typical twisted tape (*TT*) and plain tube, especially in the laminar flow regime. The obtained results also demonstrated that as the depth ratio increased and width ratio decreased, the heat transfer enhancement increased. Over the range investigated, the peripherally-cut twisted tape enhanced heat transfer rate up to 2.6 times (turbulent regime) and 12.8 times (laminar regime) of that in the plain tube. This corresponded to the maximum performance factors of 1.29 (turbulent regime) and 4.88 (laminar regime).

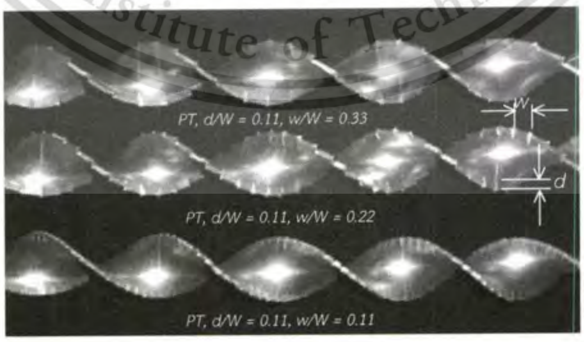


Figure 2.18: Pictorial view of peripherally-cut twisted tape (Eiamsa-ard *et al.*; 2010c).

Figure 2.19: Pictorial view of left-right twisted tape (Jaisankar *et al.*; 2011).

Jaisankar *et al.* (2011) studied the heat transfer, friction factor and thermal performance of thermosyphon solar water heater system fitted with helical and left-right twisted tape (Figure 2.19). It was discussed that the left-right twisted tape possessed bidirectional swirl flow, leading to superior heat transfer improvement over that associated by the helical twisted tape which produced single swirl flow.

Eiamsa-ard *et al.* (2012b) conducted the heat transfer and friction factor characteristics in a circular tube fitted by twisted tapes with non-uniform twist ratios (Figure 2.20). The non-uniform twisted tapes examined were (1) the one with sequentially increasing twist ratios (*SL*), (2) the one with sequentially decreasing twist ratios (*LS*), (3) the one with repeatedly increasing-decreasing twist ratios (*SL/SL*), (4) the one with repeatedly decreasing-increasing twist ratios (*LS/LS*), (5) the one with intermittently increasing-decreasing twist ratios (*SL*$\rightarrow$*LS*), (6) the one with intermittently decreasing-increasing twist ratios (*LS*$\rightarrow$*SL*). Apparently, *LS* yields monotonically increasing swirling intensity and SL or decreasing swirling intensity, resulting in lower heat transfer rate and friction factor than the other four non-uniform twisted tapes which possess extra fluid fluctuation. However, among the tapes tested, the *SL/SL* offers the maximum thermal performance factor of around 1.03 which corresponds to Nusselt number of around 36% and friction factor of 3.57 times, over those of the plain tube.

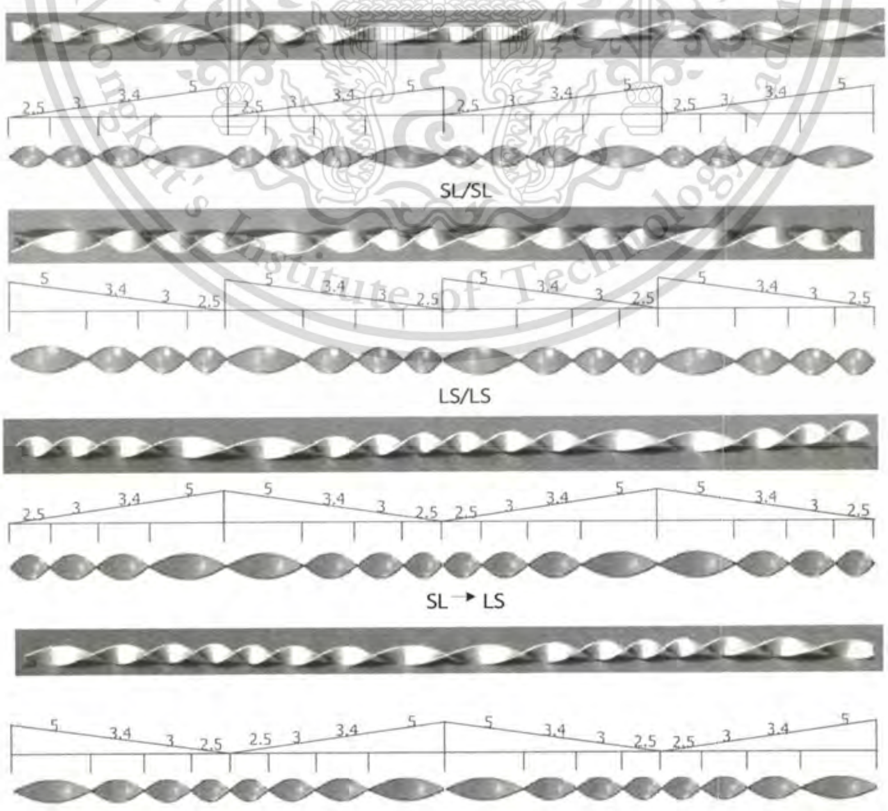


Figure 2.20: Pictorial view of non-uniform twisted tape (Eiamsa-ard *et al.*; 2012b).

Saha *et al.* (2012) performed the friction factor and heat transfer for laminar flow through a circular duct having integral helical-rib-roughness/axial-corrugation (Figures 2.21 and 2.22) and fitted with centre-cleared twisted-tape (Figure 2.23). They found that the centre-cleared twisted tapes in combination with integral helical rib roughness perform significantly better than the individual enhancement technique acting alone for laminar flow through a circular duct up to a certain amount of twisted-tape centre-clearance.

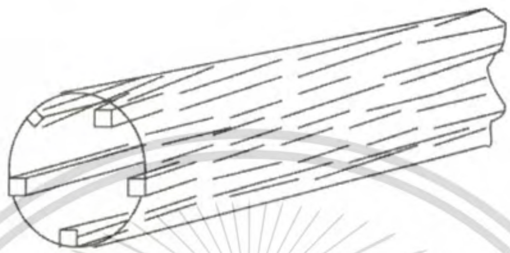


Figure 2.21: Axially corrugated circular duct (Saha *et al.*; 2012).

Figure 2.22: Axially corrugated circular duct (Saha *et al.*; 2012).

Figure 2.23: Pictorial view of center-cleared twisted-tape (Saha *et al.*; 2012).

Chang *et al.* (2012) investigated the heat transfer properties over developing and developed flow regimes, the pressure drop coefficients and the thermal performance factors of tubular flows with the continuous and spiky twist tapes (Figure 2.24) enhanced by perforated, jagged and notched winglets. Among these comparative groups, the present V-notched spiky twisted tape generally offers the highest HTE impacts with favorable thermal performance factors.

Figure 2.24: Pictorial view of *PJST* and *VST* twisted tapes (Chang *et al.*; 2012).

Zhang *et al.* (2012) performed the heat transfer enhancement in the tube fitted with triple and quadruple twisted tapes (Figure 2.25). The simulation was conducted in order to gain an understanding of physical behavior of the thermal and fluid flow in the tube fitted with triple and quadruple twisted tapes. They found that, a maximum increase of 171% and 182% were observed in the Nusselt number by using triple and quadruple twisted tapes. And the friction factors of the tube fitted with triple and quadruple twisted tapes were around 4.06–7.02 times as that of the plain tube. The thermal performance factor of the tubes varies from 1.64 to 2.46.

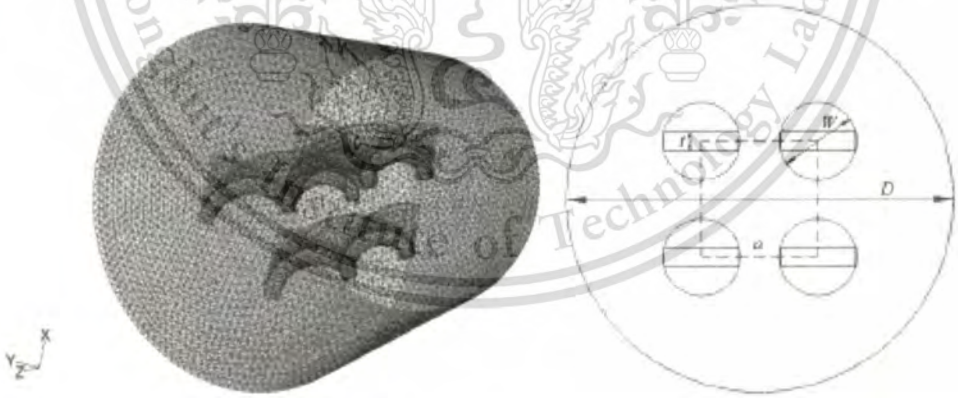


Figure 2.25: Pictorial view of tube mesh fitted with multiple twisted tapes (Zhang *et al.*; 2012).

Hong *et al.* (2012) presented a 3D numerical simulation of turbulent heat transfer and flow characteristics in converging-diverging tubes (*CDs*) and converging-diverging tubes equipped with twin counter-swirling twisted tapes (*CDTs*) (Figure 2.26). They found that all geometric parameters have important effects on the thermal performance of *CD* and *CDT*, and both *CD* and *CDT* show better thermal performance than plain tube at the constant pumping power. It is also found that the increases in the Nusselt number and friction factor for *CDT* were, respectively, up to 6.3%–35.7% and 1.75–5.3 times of the corresponding bare *CD*.

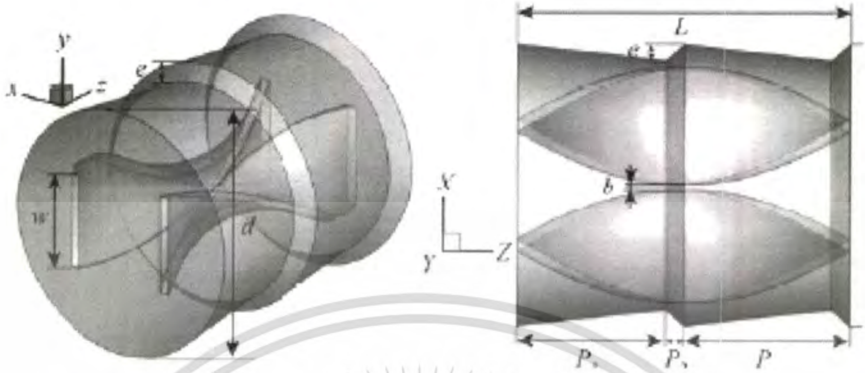
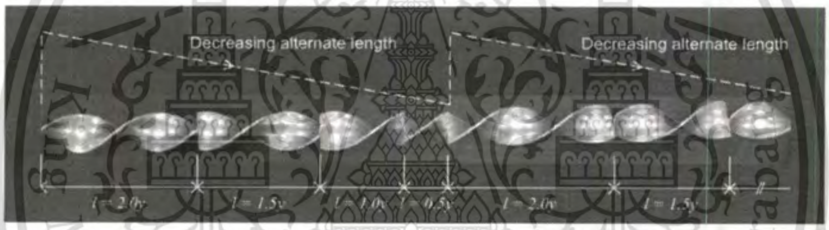
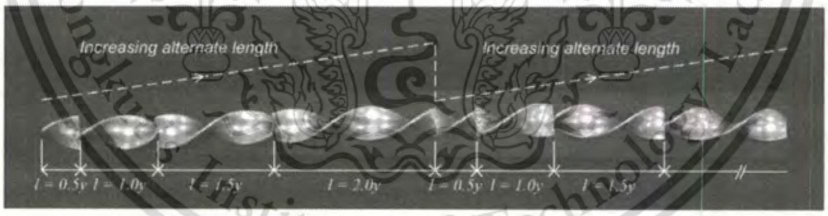


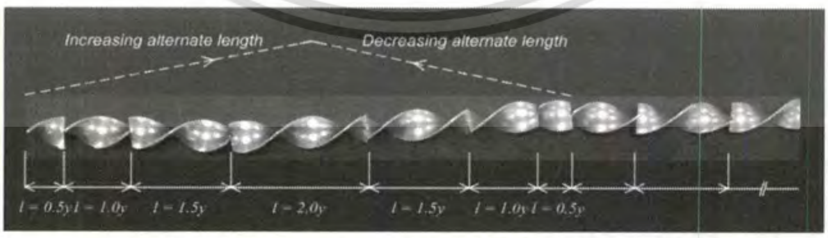
Figure 2.26: Pictorial view of converging-diverging tubes with twin twisted tapes (Hong *et al.*; 2012).



(a) N-TA with l/y = 2.0:1.5:1.0:0.5/2.0:1.5:1.0:0.5



(b) N-TA with l/y = 0.5:1.0:1.5:2.0/0.5:1.0:1.5:2.0



(c) N-TA with l/y = 0.5:1.0:1.5:2.0:1.5:1.0:0.5

Figure 2.27: Pictorial view of non-uniform twisted tape with alternate axis (Eiamsa-ard *et al.*; 2013a).

Eiamsa-ard *et al.* (2013a) reported the heat transfer enhancement in a tube using twisted tapes with alternate axes at different alternate lengths (Figure 2.27). They found that all the *TAs* and *N-TAs* yielded higher Nusselt number and friction factor than the *TT* and the Nusselt number and friction factor obtained were considerably increased with decreasing the alternate length (*l*). Heat transfer and friction factor by the *N-TAs* were found to be directly dependent of alternate length rather than the variation of the length. The optimum tradeoff between the enhanced heat transfer and increased friction was found for using the *TA* at *l/y* = 0.5, giving the highest heat transfer rate as well as the maximum thermal performance factor.

Eiamsa-ard *et al.* (2013b) investigated the effect of coupling twisted-tapes (Figure 2.28) on heat transfer enhancement in a heat exchanger. The effects of (1) twisted tape orientation: co twisted-tapes (co-*CTTs*) or counter twisted-tapes (counter-*CTTs*), (2) width ratio and (3) twist ratio, were examined. They found that the use of counter-*CTTs* resulted in higher heat transfer, friction loss and thermal performance factor than that of co-*CTTs* and *TT*. Thermal performance factor increased as twist ratio and Reynolds number decreased while width ratio increased. The highest performance factor of 1.08 was achieved by the utilizing counter-*CTTs* consisting of small twisted tape with the smallest twist ratio of 3.0 and largest width ratio of 0.5, where Nusselt number, friction factor increased by 77.7% and 595% over that of the plain tube.

Figure 2.28: Pictorial view of coupling twisted-tapes (Eiamsa-ard *et al.*; 2013b).

Eiamsa-ard *et al.* (2013c) studied the effect of twin delta-winged twisted tape insertion (Figure 2.29) on heat transfer, pressure drop and thermal performance characteristics of a heat exchanger tube. The twin delta-wings were formed by extrusion of the tape at the centre area at every twist length interval. For comparison, three different arrangements of the twin delta-wings were: (1) the wing tips pointing upstream of the flow (*TTW*-up, twin delta-winged twisted tape in counter-flow arrangement) (2) the wing tips pointing downstream of the flow (*TTW*-down, twin delta-winged twisted tape in co-flow arrangement) and (3) the wing tips pointing opposite direction (*TTW*-o, opposite winged twisted tape). Over the range investigated, the *TTW*-up with wing-tip angle of 20° gives the highest thermal performance factor of 1.26 along with the Nusselt number and friction factor of 2.57 and 8.55 times compared to those of the plain tube.

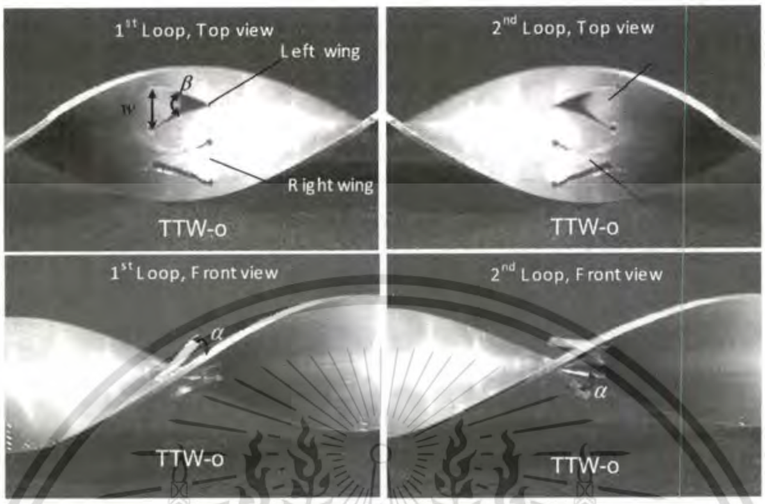


Figure 2.29: Pictorial view of twin delta-winged twisted-tape (Eiamsa-ard *et al.*; 2013c).

Due to the trade-off between the heat transfer and pressure drop mentioned above, only properly modified twisted tapes provide higher thermal enhancement factors over those of typical twisted tapes (*TTs*) at the same pumping power. Regarding to the previous works mentioned above, it can be observed that the modification of twisted tape by introducing parts which induce extra fluid disturbance, potentially increases heat transfer rate with reasonable increase of pressure drop, resulting in superior thermal enhancement factor over typical twisted tape.

2.4 Twisted tape with nanofluid

Recently, the utilizing of nanofluids in the tubes with twisted tape inserts was reported. In general, the combined techniques improved heat transfer rate with moderate increase of pressure drop as compared to those obtained from separated techniques. Nanofluids are introduced the fluids containing nanometer-sized particles or nanoparticles. Typical nanoparticles used in nanofluids are made of metals, oxides and carbides while common base fluids are water, ethylene glycol and oil. Nanofluids exhibit their great potentials for heat transfer applications in many areas, including microelectronics, fuel cells, pharmaceutical processes, and hybrid-powered engines. As revealed in several research works the effective conductivities of nanofluids are considerably enhanced as compared to those of their base fluids, leading to superior convective heat transfer coefficients.

Sharma *et al.* (2009) presented the heat transfer coefficient and friction factor in a tube with twisted tape inserts (Figure 2.30) in the transition range of flow by using Al_2O_3 nanofluid. They observed that the heat transfer coefficient of nanofluid flowing in a tube with 0.1% volume concentration was 23.7% higher when compared with water at number of 9000. The maximum friction factor with twisted tape at 0.1% nanofluid volume concentration was 1.21 times that of water flowing in a plain tube. Again, Syam Sundar and Sharma (2010) studied the convective heat transfer coefficient and friction factor in a tube fitted with twisted tape using water/Al_2O_3 nanofluid as the working fluid. The heat transfer coefficient and friction factor of 0.5% volume concentration of Al_2O_3 nanofluid with twist ratio of five was 33.51% and 1.096 times respectively higher compared to flow of water in a tube.

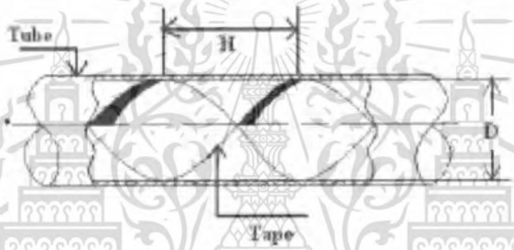


Figure 2.30: Pictorial view of typical twisted-tape (Sharma *et al.*; 2009).

Wongcharee and Eiamsa-ard (2011) conducted the heat transfer, friction and thermal performance characteristics of CuO/water nanofluid in a circular tube equipped with twisted tape with alternate axis (Figure 2.31). The concentration of nanofluid was varied from 0.3 to 0.7% by volume. The CuO nanoparticles were purchased from Nanostructured and Amorphous Material, Inc, USA. To prepare nanofluid, the nanoparticles with the required volume concentration of 0.3%, 0.5% and 0.7% volume were dispersed in water ultrasonic vibrator for 5 hours. The transmission electron microscopy (TEM) photograph of the CuO particles in water was taken using Hitachi Transmission Electron Microscope H-9500. As shown in Figure 2.32, the nanoparticles are in approximately spherical shape with diameter between 30 and 50 nm. Over the range investigated, the maximum Nusselt number, friction factor and thermal performance factor of 5.53 found with the simultaneous employment of the CuO/water nanofluid at 0.7% volume and the twisted tape with alternate axis at Reynolds number of 1990.

Wongcharee and Eiamsa-ard (2012) investigated the heat transfer rate by using CuO/water nanofluid in corrugated tube equipped with twisted tape they found that the maximum thermal performance factor 1.57 was found with the use of CuO/water nanofluid at concentration of 0.7% by volume. Again, Eiamsa-ard and Wongcharee (2012c) reported the effects of nanofluids and a micro-fin tube on the heat transfer rate, friction factor and thermal performance factor characteristics with the nanofluids consisting of CuO and water at CuO concentrations between 0.3% and 1.0% by volume. They found that the heat transfer rate increased with increasing nanofluid concentration.

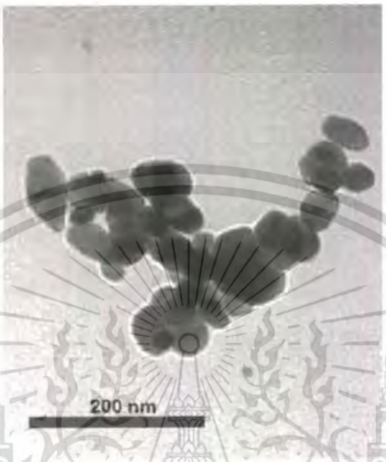


Figure 2.31: Pictorial view of TEM image of CuO nanoparticles (Wongcharee and Eiamsa-ard; 2011).

In addition, the twisted tapes were also applied in the refrigeration system for enhancing the heat transfer rate. Akhavan-Behabadi *et al.* (2009) carried out the heat transfer enhancement and pressure drop characteristics inside a horizontal evaporator fitted with twisted tape during flow boiling of R-134a. The experiments were performed for plain flow and four tubes with twisted tapes of 6, 9, 12 and 15 twist ratios and four refrigerant mass velocities of 54, 85, 114 and 136 kg/s for each tape. They found that the twisted tape inserts enhance the heat transfer coefficient on relatively higher pressure drop penalty, in comparison to that for the plain flow. Hejazi *et al.* (2010) investigated the augmentation of heat transfer coefficients and pressure drop during condensation of HFC-134a in horizontal tubes fitted with twisted tapes at different twist ratios. They found that the tube with twisted tape provided higher heat transfer rate and pressure drop than the plain tube by around 40% and 240%, respectively.

2.5 Compound devices

Another key device for heat transfer enhancement is swirl generator. Twisted tape insert as the common swirl generator is widely employed to generate swirl into the bulk flow for thinning the thermal/velocity boundary layer. The twisted tape also plays a vital role as the blockage of tube flow cross section which leads to a higher flow velocity and hence momentum transfers. In addition its helical geometry is useful for prolonging residence time of fluid flow within heat exchanger which directly associates heat transfer enhancement. It was reported that the performance of heat transfer enhancement of twisted tapes is strongly dependent on their geometries. Apart from the utilization of each enhancing heat transfer device, efforts have also been made by combining different types of enhancement devices known as compound devices, to further improve heat transfer. Twisted tapes were applied together with several modified tubes for improving the heat transfer rate such as tube with internal extended surfaces, single/three-start spirally corrugated tubes, converging-diverging tube, square ducts with internal transverse rib turbulator, spirally grooved tube, micro-fin tube, dimpled tube. In addition twisted tapes were also combined with other types of turbulators such as conical-ring, wire coils, interrupted ribs in square channels, duct with periodic transverse rib, and wire nails.

Promvonge and Eiamsa-ard (2007) examined the heat transfer, friction factor and thermal performance factor characteristics in a circular tube fitted with conical-ring turbulators and a twisted-tape (Figure 2.32). In the experiments, two enhancement heat transfer devices were applied. One is the conical-ring used as a turbulator and placed in the tested tube and the other is the twisted-tape swirl generator placed at the core of the conical-ring. They found that the tube fitted with the conical-ring and twisted-tape provides Nusselt number values of around 4 to 10% and thermal performance factor of 4 to 8% higher than that with the conical-ring alone. A maximum heat transfer rate of 367% and thermal performance factor of around 1.96 was found for using the conical-ring and the twisted-tape of twist ratio 3.75.

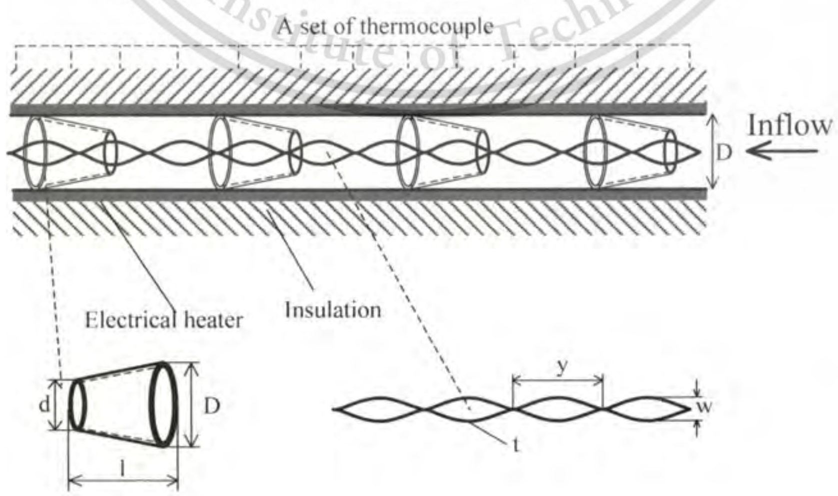


Figure 2.32: Pictorial view of conical-ring turbulator and twisted tape (Promvonge and Eiamsa-ard; 2007).

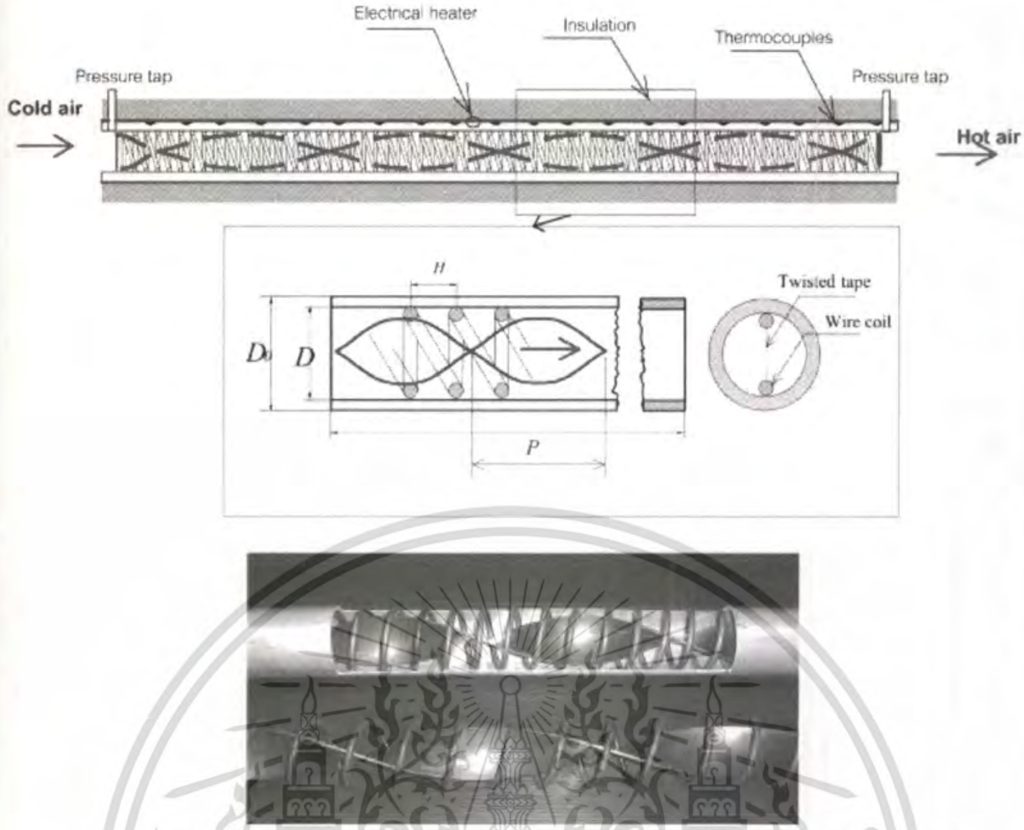


Figure 2.33: Pictorial view of wire coil and twisted tape (Promvonge; 2008).

Promvonge (2008) proposed the influences of insertion of wire coils in conjunction with twisted tapes (Figure 2.33) on heat transfer and friction characteristics. The wire coil used as a turbulator was placed inside the test tube while the twisted tape was inserted into the wire coil to create a continuous impinging swirl flow along the tube wall. The results indicate that the presence of wire coils together with twisted tapes leads to a double increase in heat transfer over the use of wire coil/twisted tape alone. The combined twisted tape and wire coil with smaller twist and coil pitch ratios provides higher heat transfer rate than those with larger twist and coil pitch ratios under the same conditions.

Thianpong *et al.* (2009) presented the heat transfer behaviors in a dimpled tube fitted with a twisted tape (Figure 2.34). The experiment were performed using two dimpled tubes with different pitch ratios of dimpled surface and three twisted tapes with three different twist ratios. They found that both heat transfer coefficient and friction factor in the dimpled tube fitted with the twisted tape, were higher than those in the dimple tube acting alone and plain tube. It was also found that the heat transfer coefficient and friction factor in the combined devices increase as the pitch ratio (*PR*) and twist ratio (*y/w*) decrease.

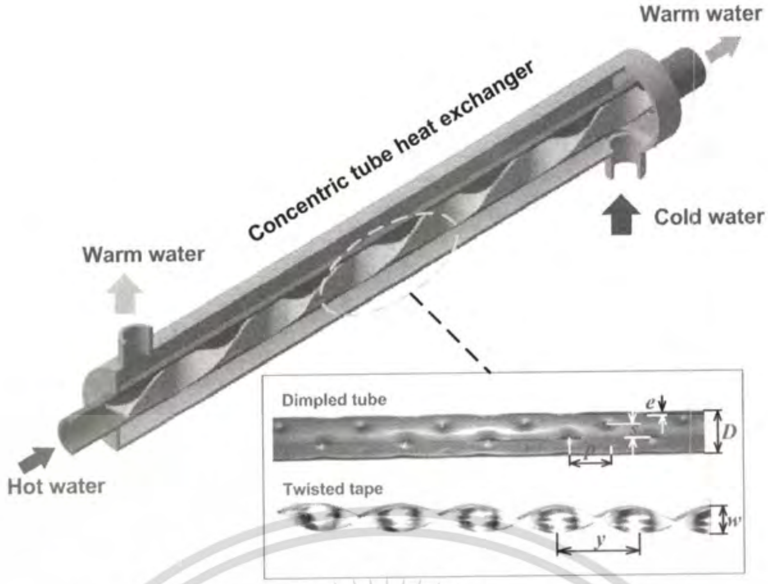


Figure 2.34: Pictorial view of dimpled tube fitted with a twisted tape (Thianpong *et al.*; 2009).

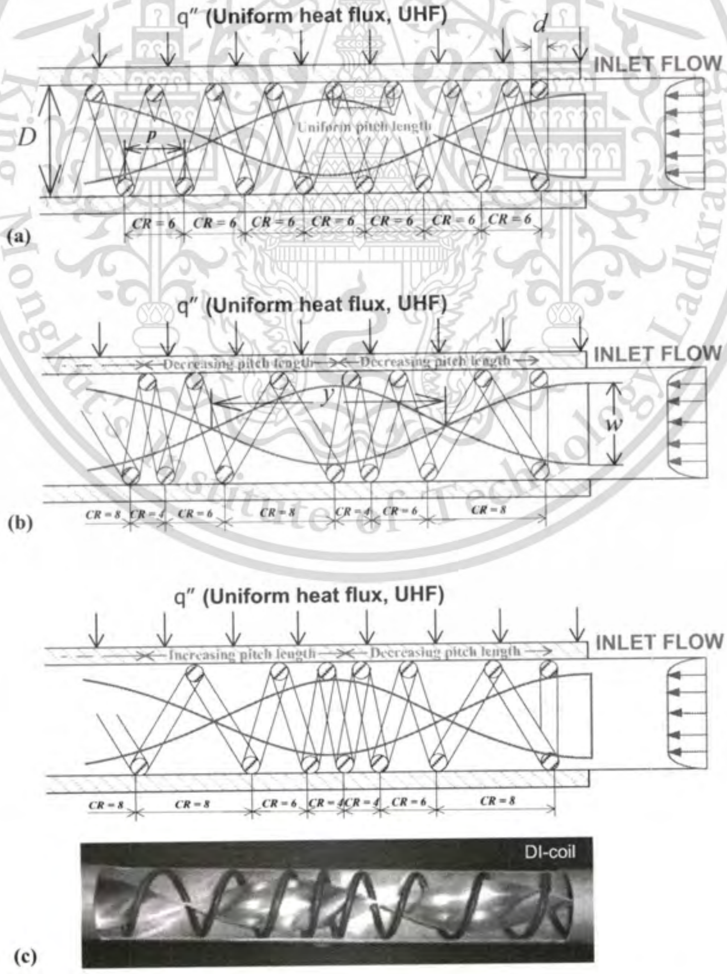


Figure 2.35: Pictorial view of combined devices between twisted tape and constant/periodically wire coil (Eiamsa-ard *et al.*; 2010d).

Eiamsa-ard *et al.* (2010d) examined the heat transfer, friction factor and thermal performance behaviors in a tube equipped with the combined devices between the typical twisted tape (*TT*) and constant/periodically varying wire coil pitch ratio (Figure 2.35). The periodically varying three coil pitch ratios were arranged into two different forms: (1) *D*-coil (decreasing coil pitch ratio arrangement) and (2) *DI*-coil (decreasing/increasing coil pitch ratio arrangement) while the twisted tapes were prepared with two different twist ratios. Compared to each enhancement device, the heat transfer rate is further augmented by the compound devices. Over the range investigated, the highest thermal performance factor of around 1.25 was found by using *DI*-coil in common with the *TT* at lower Reynolds number.

Saha (2010) reported the heat transfer and pressure drop characteristics in rectangular/square duct with internal axial corrugations (Figure 2.36) equipped by twisted-tape inserts with and without oblique teeth. At similar conditions, the twisted tapes with oblique teeth gave better performance than the ones without oblique teeth. In addition, the effects of the duct aspect ratio, corrugation angle, corrugation pitch, twist ratio, space ratio, length, tooth horizontal length and tooth angle of the twisted-tape, Reynolds number and Prandtl number on thermal performance were studied

Figure 2.36: Square duct with internal axial corrugation (Saha; 2010).

Wongcharee and Eiamsa-ard (2012) investigated the heat transfer, friction factor and thermal performance factor in corrugated tube equipped with twisted tape into two different arrangements of twisted direction of twisted tape relative to spiral direction of corrugated tube: parallel and counter arrangements. They shows that the twisted tape coupled with corrugated tube in counter pattern offer higher heat transfer performances than the ones in parallel pattern (Figure 2.37). They found that with the use of corrugated tube together with twisted tape at twist ratio of 2.7 (in counter arrangement) where heat transfer rate and friction factor increase to 2.67 times and 5.76 times of those in the plain corrugated tube.

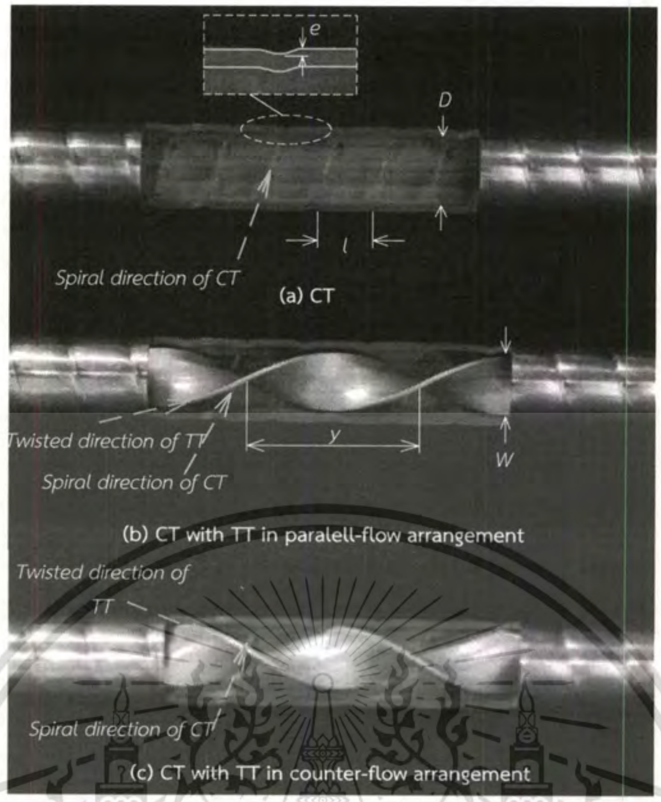


Figure 2.37: Pictorial view of corrugated tube equipped with twisted tape (Wongcharee and Eiamsa-ard; 2012).

Eiamsa-ard and Wongcharee (2012c) examined the heat transfer and frction factor characteristics in a micro-fin tube fitted with dual twisted-tapes (Figure 2.38). They observed that the micro-fin tube equipped with dual twisted-tapes consistently gave superior thermal performance factor to the one equipped with a single twisted-tape as well as the micro-fin tube alone.

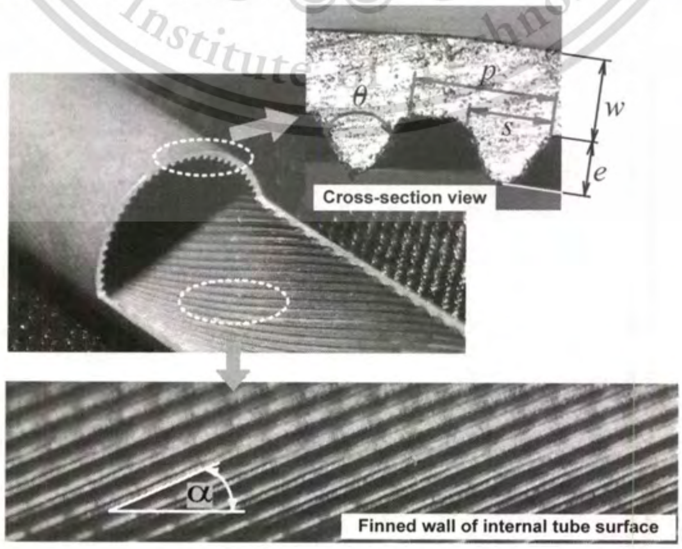


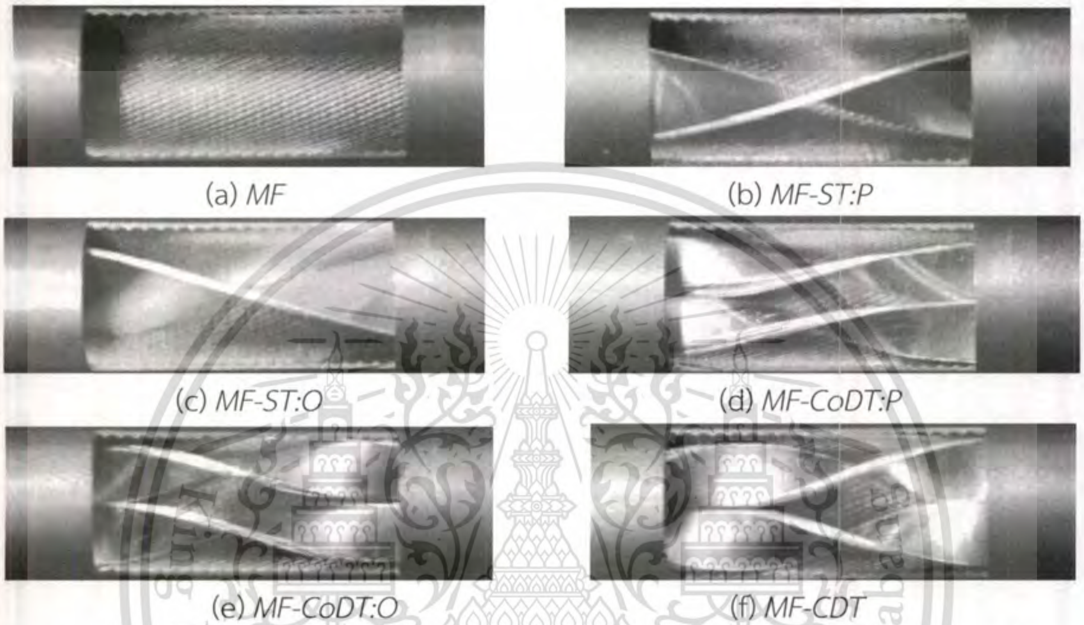
Figure 2.38: Pictorial view of micro-fin tube (Eiamsa-ard and Wongcharee; 2012c).

Eiamsa-ard *et al.* (2013d) reported the influence of co/counter swirl flow on the characteristics of friction, heat transfer and thermal performance factor. Each twisted tape was inserted into a helical screw tape and then these combined tapes (Figure 2.39) were subsequently equipped in a heat exchanger tube for generating the swirl flow. The combined tapes were arranged in two different forms, namely (1) co-swirl arrangement in which their twist were in the same direction, and (2) counter-swirl arrangement in which their twist directions were in the opposite directions. The helical screw tapes were used at three different tape-twist ratios ($PR = p/D$ = 1.0, 1.5 and 2.0), each with three different tape-width ratios ($WR = w/D$ = 0.15, 0.2 and 0.25), while twisted tapes were used at three different twist ratios ($YR = y/W$ = 3, 4 and 5). The results show the maximum thermal performance factor of 1.49 for helical tape alone with PR=2.0 and WR=0.2. At similar conditions, the heat transfer rate associated with the combined tapes in counter-swirl arrangement was 3.4% and 10% higher than those in co-swirl arrangement and helical tape alone.

Figure 2.39: Pictorial view of helical screw tape combined with twisted tape in co/counter swirl arrangements (Eiamsa-ard *et al.*; 2013d).

Eiamsa-ard and Wongcharee (2013e) carried out to investigate the influence of double twisted-tape inserts (*DTs*) in micro-fin tubes (*MFs*) (Figure 2.40) on heat transfer, friction factor and thermal performance factor characteristics. The compound devices in the following configurations: (1) twisted tapes acted in the same direction (for co-swirl) while *MF* and twisted tapes acted in the same (parallel) direction (*MF-CoDTs:P*), (2) twisted tapes acted in the same direction (for co-swirl) while micro-fin tube and twisted tapes acted in opposite directions (*MF-CoDTs:O*) and

(3) twisted tapes acted in opposite directions for counter swirl (*MF-CDTs*) were tested. They found that *MF-CDTs* induce stronger swirl/turbulence flow, resulting in higher heat transfer rate, friction factor and thermal performance factor than other combined devices. The thermal performance factors associated with the use of *MF-CDTs* were found to be higher than those associated with the uses of *MF-CoDTs:P*, *MF-CoDTs:O* and *MF* alone up to 9.3%, 6.5% and 56.4%, respectively.



(a) *MF* (b) *MF-ST:P*
(c) *MF-ST:O* (d) *MF-CoDT:P*
(e) *MF-CoDT:O* (f) *MF-CDT*

Figure 2.40: Pictorial view of micro-fin tube (*MF*) fitted with twisted tape (s) with different arrangements (Eiamsa-ard and Wongcharee; 2013e).

In general, the use of compound devices results in the increase in both heat transfer coefficient over that of the individual devices. However, other factors such as the overall performance indicated by thermal performance factor above unity, the ease of installation/operation, the durability as well as the strength of devices should be taken into consideration for selecting compound devices.

2.6 General conclusions

According to the above literature review, it can be concluded that the modified twisted tape which gives greater higher heat transfer rate usually generates larger pressure drop (Table 2.1). However, the increase of heat transfer will be prominent over that of pressure drop at a proper geometry and optimum conditions. This inspires us to propose "perforated twisted tape, delta-winglet twisted tape, twisted tapes with triangular, rectangular and trapezoidal wings, perforated twisted tape with parallel wing, twin twisted tapes, wing twisted tape with/without alternate axis" as another form of modified twisted tape.

Table 2.1:

Previous studies of modified twisted-tapes

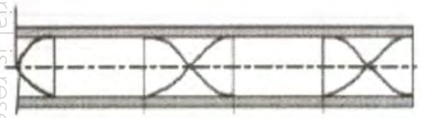
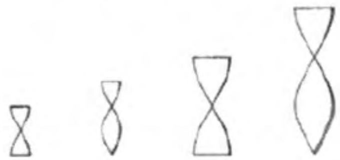
Author (Year)	Type	working fluid	Reynolds number	Nu/Nu_p	f/f_p	η
Eiamsa-ard *et al.* (2006) 	regularly spaced twisted tape	water	2000-12,000	2.2-2.0	4.4-2.5	-
Mengna *et al.* (2007) 	-	air	3400–20,000	1.21-0.85	8.0-9.5	1.6-2.3
Chang *et al.* (2007) 	broken twisted tape	water	1000–40,000	2.0-4.7	1.28-2.4	0.99-1.8
Chang *et al.* (2007) 	serrated twisted tape	water	5000–25,000	3.8-6.0	1.25-1.67	0.9-0.65

Table 2.1:

Previous studies of modified twisted-tapes *(continued)*

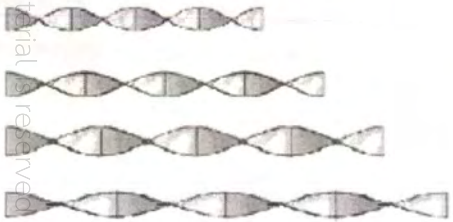
Author (Year)	Type	working fluid	Reynolds number	Nu/Nu_p	f/f_p	η
Jaisankar *et al.* (2009) 	typical twisted tapes at various twist ratios	water	3000-23,000	11-1.4	2.33-2.08	-
Akhavan-Behabadi *et al.* (2009) 	-	R-134a	41,000–96,000	1.43	-	1.7
Rahimi *et al.* (2009) 	jagged twisted tape	water	2950-11,800	2.49-1.96	8.7-6.5	1.21-1.05
Eiamsa-ard *et al.* (2009) 	-	air	4000-20,000	1.99-1.76	1.27-1.16	0.95 – 1.0

Table 2.1:

Previous studies of modified twisted-tapes *(continued)*

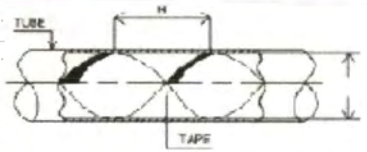
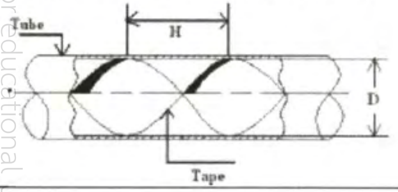
Author (Year)	Type	working fluid	Reynolds number	Nu/Nu_p	f/f_p	η
Sharma *et al.* (2009) 	-	Al_2O_3	3000-9000	1.33-1.0	0.11-0.04	-
Sundar and Sharma (2010) 	-	Al_2O_3 nanofluid	10,000–22,000	1.33 (Nu/Nu_{TT})	1.096 (f/f_{TT})	-
Eiamsa-ard *et al.* (2010) 	peripherally-cut twisted tape	water	1000-20,000	3.3–1.35	12.5–3.7	1.4–0.8
Seemawut and Eiamsa-ard (2010) 	-	water	5200- 22,000	2.83-1.85	8.4-6.5	1.39-0.98

Table 2.1:

Previous studies of modified twisted-tapes *(continued)*

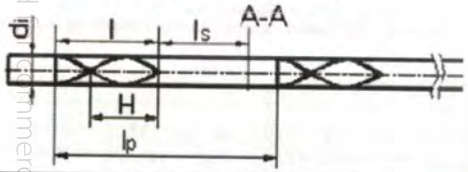
Author (Year)	Type	working fluid	Reynolds number	Nu/Nu_p	f/f_p	η
Eiamsa-ard and Promvonge (2010) 	twisted tape with serrated edge (*STT*)	air	4000-20,000	2.5-3.3	1.4-1.6	1.17-1.09
Murugesan *et al.* (2010) 	V cut twisted tape inserts (*VTT*)	water	3000-11,000	3.17–1.79	26.7–10.3	1.18–1.02
Eiamsa-ard and Promvonge (2010) 	-	water	3000-27,000	2.19-1.18	6.6–2.3	1.3-1.02
Yong-Zhang and Mao-Cheng (2010) 	-	air	2500-9500	1.04-1.09 (Nu/Nu_{TT})	1.09-1.74 (f/f_{TT})	1.24-1.4
Wang *et al.* (2011) 	regularly spaced short-length twisted tape	air	10,000-20,000	-	-	1.079

Table 2.1:

Previous studies of modified twisted-tapes *(continued)*

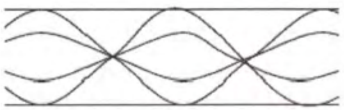
Author (Year)	Type	working fluid	Reynolds number	Nu/Nu_p	f/f_p	η
Guo *et al.* (2011) 	center-cleared/short-width twisted tape	water	500-1750	-	-	1.07-1.2
Murugesan *et al.* (2011) 	square-cut twisted tapes (*STT*)	water	2000-12,000	1.92–1.49	5.2–2.5	1.2 –1.09
Wongcharee and Eiamsa-ard (2011) 	-	water	830-1990	14-8.0	12-6.0	5.2-3.7
Saha (2012) 	centre-cleared twisted-tape	oil	3000-23,000	1.3-1.45 (Nu/Nu_{TT})	1.1-1.2 (f/f_{TT})	1.15-1.3

Table 2.1:

Previous studies of modified twisted-tapes *(continued)*

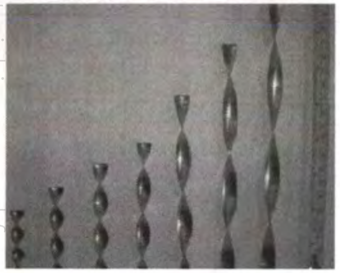
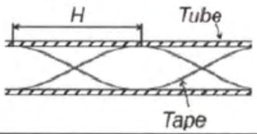
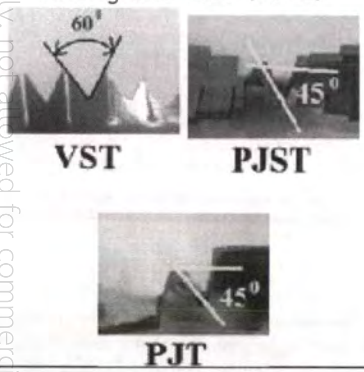
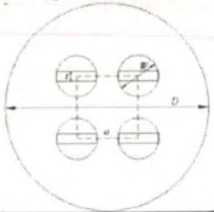
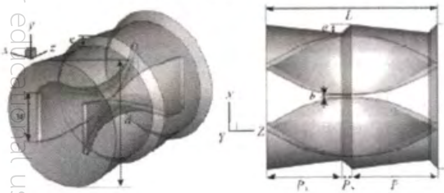
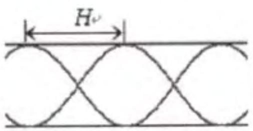
Author (Year)	Type	working fluid	Reynolds number	Nu/Nu_p	f/f_p	η
Ferroni *et al.* (2012) 	-	water	10,000-85,000	1.5-3.2	1.15	-
Mogaji *et al.* (2012) 	-	R134a	-	-	-	-
Chang and Guo (2012) 	V-notched spiky twisted tape (*VST*)	air	500-40,000	2.4-11.6	12.8-28	1.83-3.18

Table 2.1:

Previous studies of modified twisted-tapes *(continued)*

Author (Year)	Type	working fluid	Reynolds number	Nu/Nu_p	f/f_p	η
Zhang *et al.* (2012) 	triple and quadruple twisted tapes	air	300-1800	1.71-1.82	4.06-7.02	1.62-2.46
Hong *et al.* (2012) 	converging-diverging tubes (*CDs*) and converging-diverging tubes equipped with twin counter-swirling twisted tapes (*CDTs*)	water	10,000-20,000	1.06-1.36	1.75-5.3	1.23-1.59
Song *et al.* (2013) 	-	MPCM slurry	200-2200	-	-	3.3

CHAPTER 3

Experimental Facility

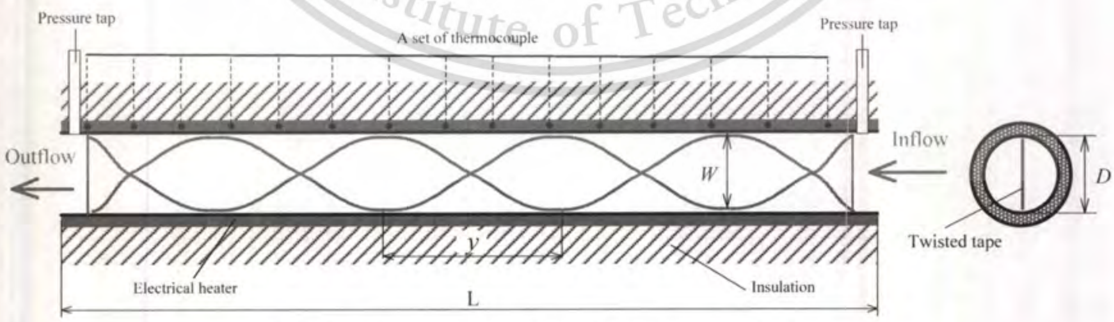
3.1 Open remarks

In this chapter, details of the experimental apparatus of the heat exchanger tube with twisted tapes inserts are presented. The experimental facility is including: (1) physical model of modified twisted tapes, (2) experimental setup and (3) data reduction. The experiments were conducted to investigate the effects of modified twisted tapes on the heat transfer rate, friction factor and thermal performance factor characteristics in a turbulent flow that considered under uniform wall heat flux conditions.

3.2 Physical model of twisted tape

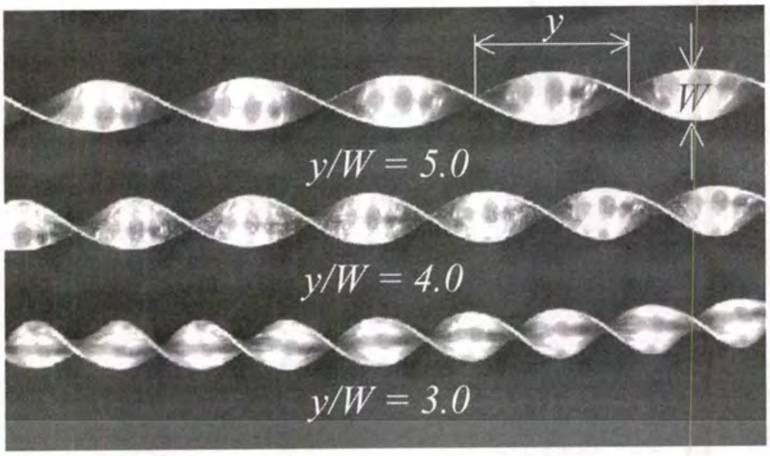
3.2.1 Typical twisted tape

Typical twisted tapes (*TTs*) were made of aluminium sheet with tape width (*W*) of 19 mm, tape thickness of 0.8 mm and tape length (*L*) of 1000 mm. Typical twisted tapes (*TTs*) were prepared with three different twist ratios, y/W = 3.0, 4.0 and 5.0 where twist ratio is defined as twist length (y: 180°/twist length) to tape width (*W*). They were fabricated by twisting straight tapes, about their longitudinal axis, while being held under tension. Figure 3.1(a-b) demonstrates the twisted tapes at different twist ratios (y/W) and their arrangements. In the experiments, the typical twisted tape (*TT*) was also subjected to the test, for comparison. All of the tapes were inserted at the core tube along the test section.



(a) diagram of tube fitted with twisted tape

Figure 3.1: Pictorial view of typical twisted tapes at different twist ratios (y/W).

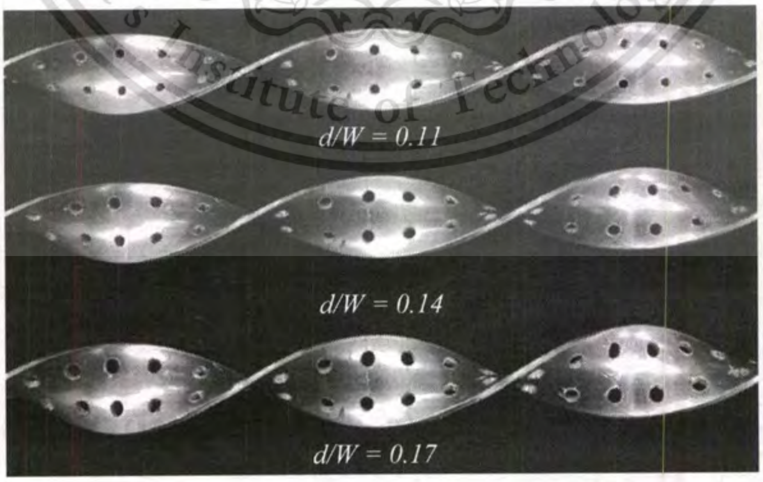


(b) typical twisted tape

Figure 3.1: Pictorial view of typical twisted tapes at different twist ratios (*y/W*). (*continued*)

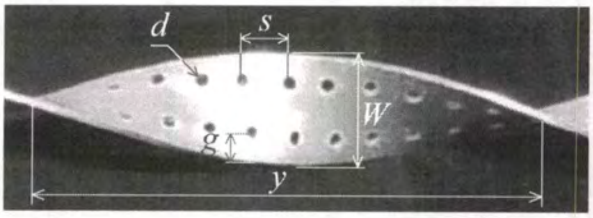
3.2.2 Perforated twisted tape

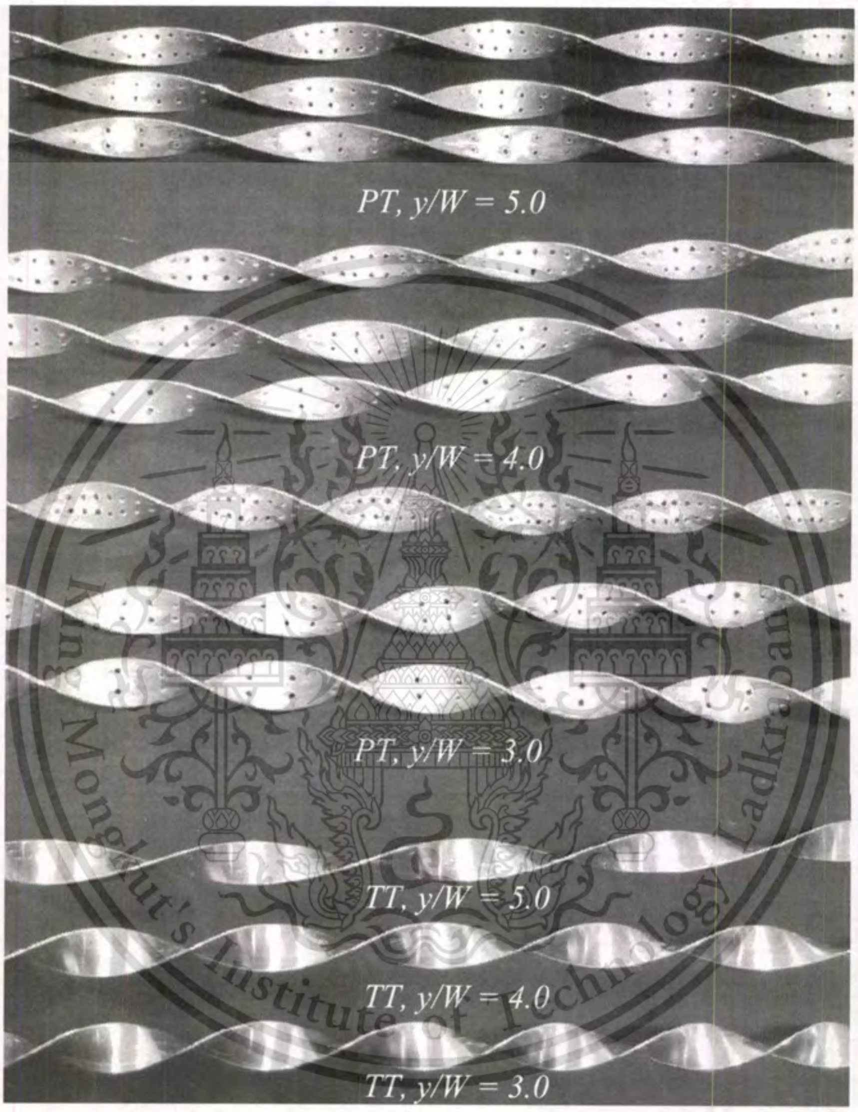
Twisted tapes were made from aluminum strip with a thickness of 0.8 mm and a width of 18 mm (*W*). The tapes were formulated in three different twist lengths of *y* = 54 mm (*y/W* = 3), 72 mm (*y/W* = 4), and 90 mm (*y/W* = 5). Perforated twisted tapes (*PTs*) were then modified from the typical twisted tape by drilling the small holes with three different hole diameters (*d*) of 2 mm (*d/W* = 0.11), 2.5 mm (*d/W* = 0.14) and 3 mm (*d/W* = 0.17) along the tape edge at three different space/pitch lengths of *s* = 7.5 mm (*s/W* = 0.4), 11 mm (*s/W* = 0.6), and 15 mm (*s/W* = 0.8) as shown in Figure 3.2(a-b).



(a) *PTs* with various perforation hole diameters

Figure 3.2: Pictorial view of perforated twisted tape (*PT*).





(b) *PTs* with various twist and pitch ratios in comparison with *TTs*

Figure 3.2: Pictorial view of perforated twisted tape (*PT*). (*continued*)

3.2.3 Delta-winglet twisted tape

All of delta-winglet twisted tape tapes (*DWTs*) that are used in the present work are made of aluminum strip with 0.8 mm thickness and 19 mm width (*W*). Firstly, aluminum strip was twisted to produce a typical twisted tape (*TT*). A tape was subsequently modified to obtain the *DWT* by cutting at the edge of the tape with oblique shape and straight shape to produce an oblique delta-winglet twisted-tape

(*O-DWT*) and a straight delta-winglet twisted-tape (*S-DWT*), respectively. Then the outer part of the cut was arranged to 90° (*degree*) relatively to the inner part, forming delta-winglet shape. The length between the two cuts was set equally to the twist length (*y*). The *DWTs* were prepared with three different twist ratios (*y/W*) = 3.0, 4.0 and 5.0, and three different depth of wing cut ratios (*d/W* = 0.11, 0.21 and 0.32). The geometrical configurations of the delta-winglet twisted tape (*DWT*) inserts are presented in Figure 3.3(a-c).

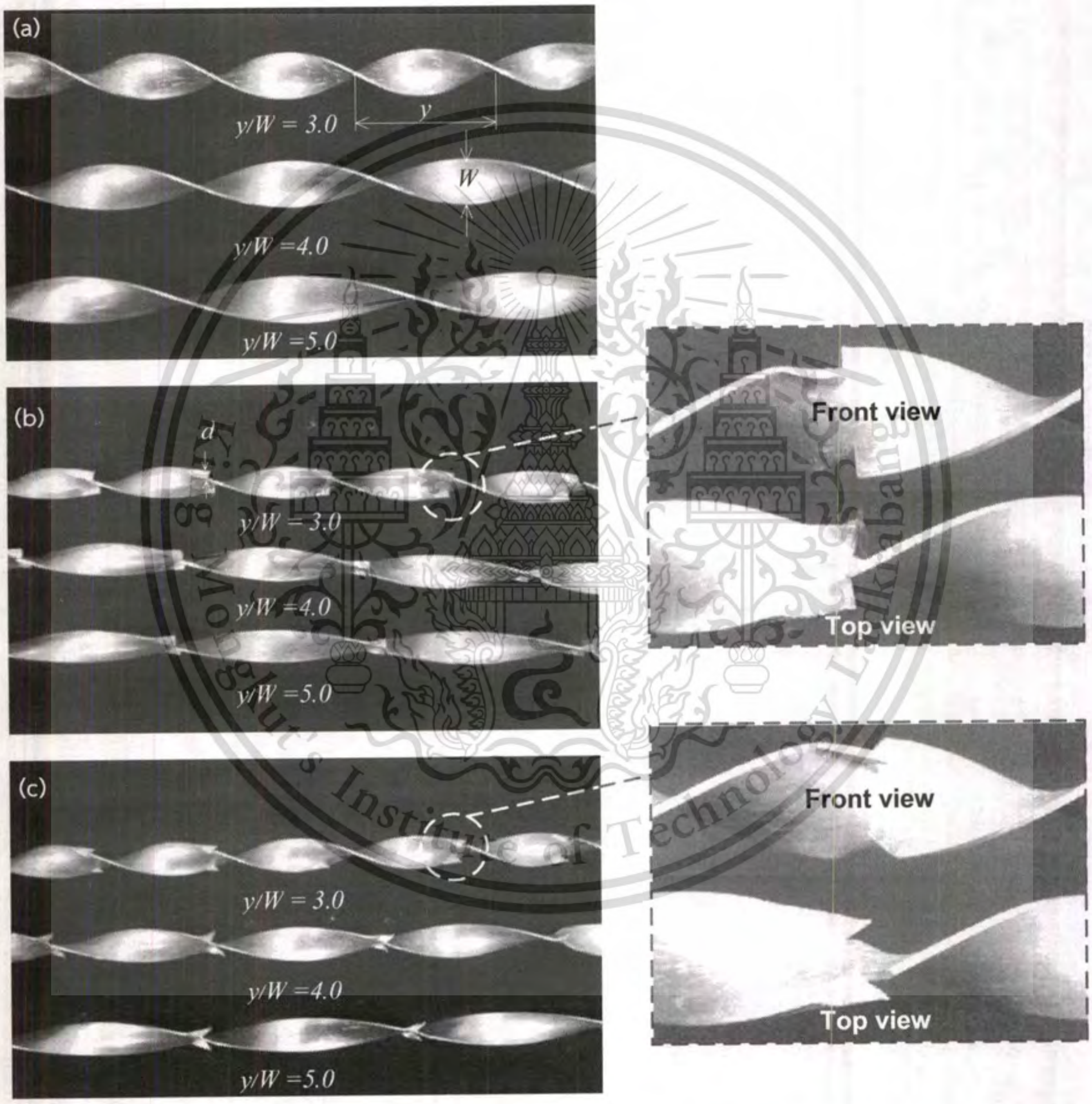
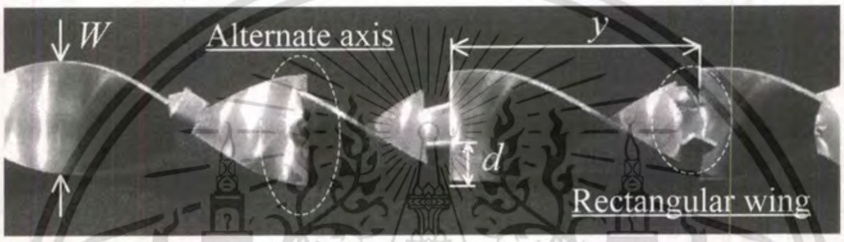


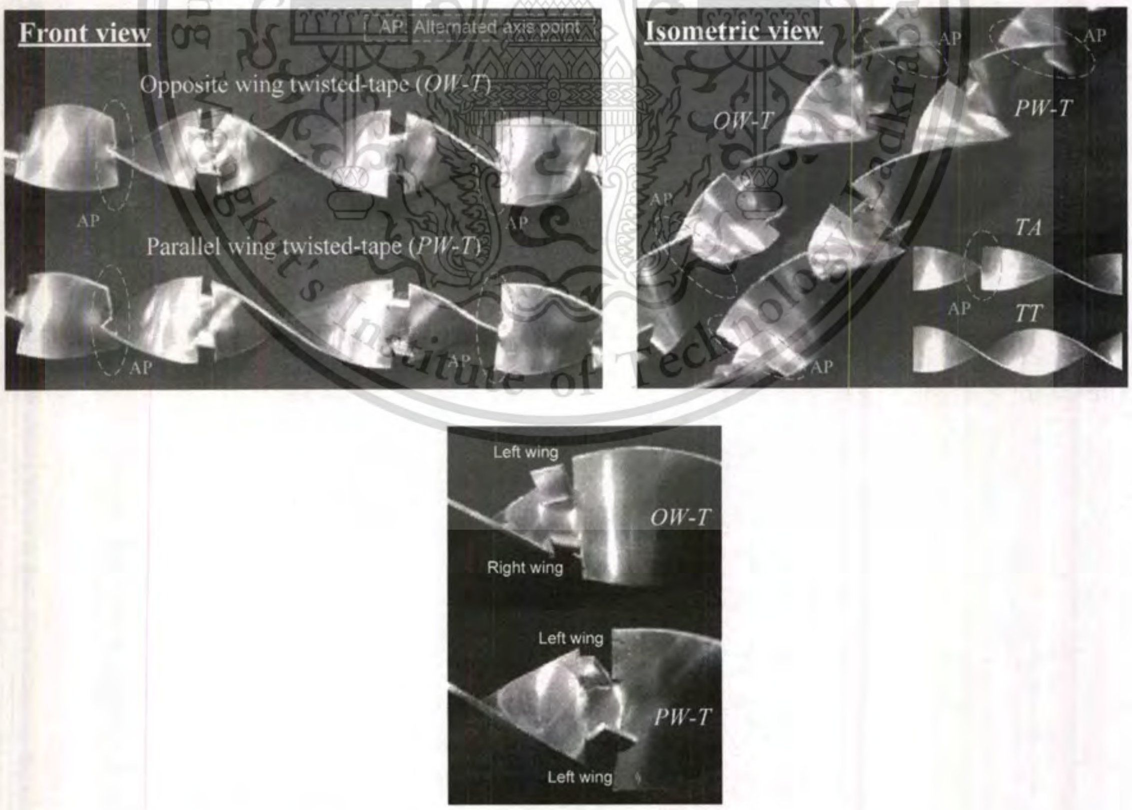
Figure 3.3: Pictorial view of twisted tape: (a) typical twisted tape (*TT*), (b) straight delta-winglet twisted tapes (*S-DWT*) and (c) oblique delta-winglet twisted tapes (*O-DWT*).

3.2.4 Twisted tape with rectangular wing

Twisted-tape with rectangular-wings (*T-Rec*) were made from the aluminum strip sheet and twisted at constant twist length of y = 80 mm or y/W = 4.0, where the tape has 20 mm in width (W) and 0.8 mm in thick as depicted in Figure 3.4. The *TT-RWs* were cut at the edge of the tape in straight line at every pitch length (y: 180°/twist length). Then each cut was bended in 45° to the main flow to form (1) opposite wing twisted-tape (*OW-T*) and (2) parallel-wing twisted-tape (*PW-T*). The wings were generated in three different depths of cut of d = 2, 4 and 6 mm, corresponding to depth ratios $DR = d/W$ = 0.1, 0.2 and 0.3, respectively. To form the alternate axis, the tape was cut on both sides at every double twist length (2y: 360 degree). Subsequently, both sides of the tape were twisted simultaneously to angle difference of 90 degree.



(a) details of twisted-tapes with rectangular-wings



(b) opposite/parallel-wing twisted-tape (*OW-T/PW-T*)

Figure 3.4: Pictorial view of twisted-tapes with rectangular-wings (*T-Rec*).

3.2.5 Perforated twisted tape with parallel wing

Perforated twisted-tapes with parallel wings were made from an aluminum strip sheet having width (*W*) of 18 mm and thickness of 1.0 mm. All tapes were twisted at constant twist length of y = 56 mm which corresponds to twist ratio (y/W) = 3.0 to formulate typical twisted tapes (*TTs*). Then, *TTs* were modified by periodic generating holes along a core tube in straight line at every twist length (y: 180°/twist length). Furthermore, each tape was cut at the edge on both sides between adjacent holes, each cut was subsequently arranged in 45° to the axial flow in the same direction, so called parallel-wings the tap. The modified twisted tape (*PTT*) is depicted in Figure 3.5. The parameters investigated were the hole diameter ratio (d/W = 0.11, 0.33 and 0.55) and wing depth ratio (w/W = 0.11, 0.22 and 0.33).

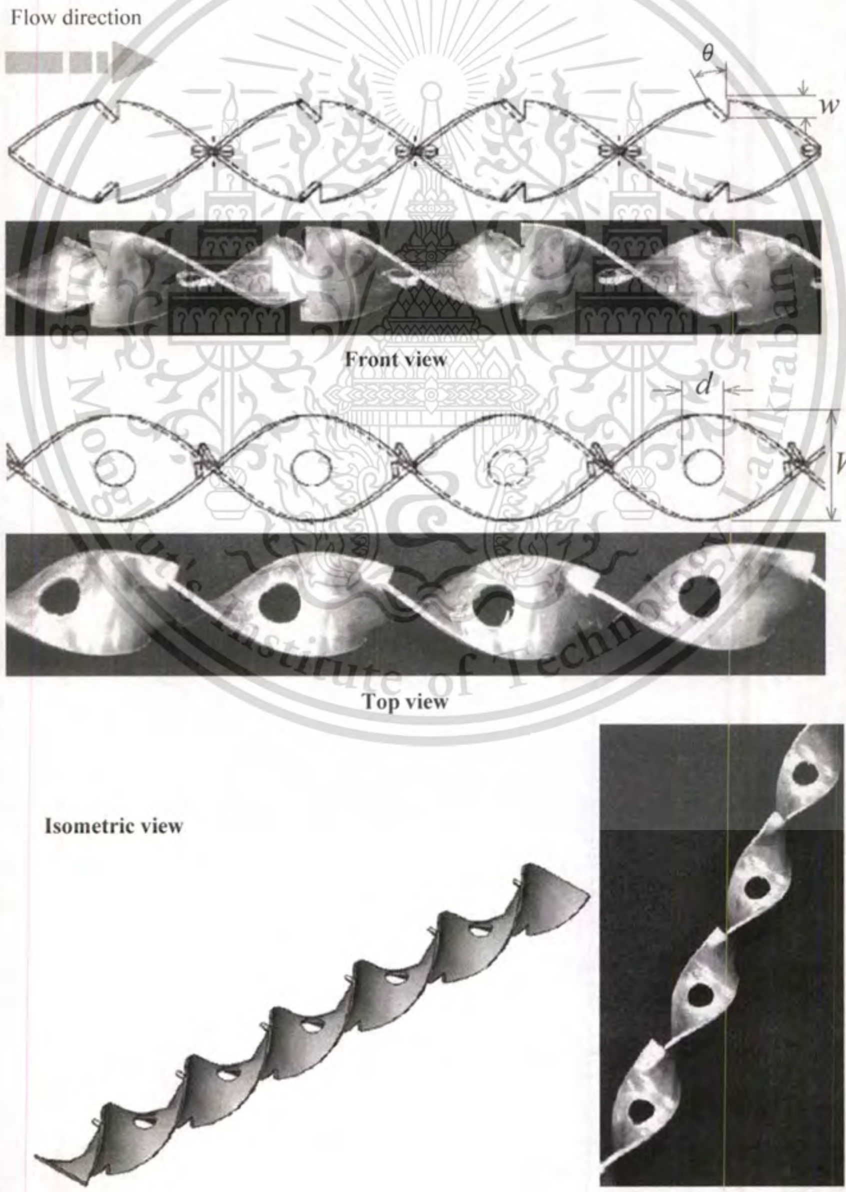
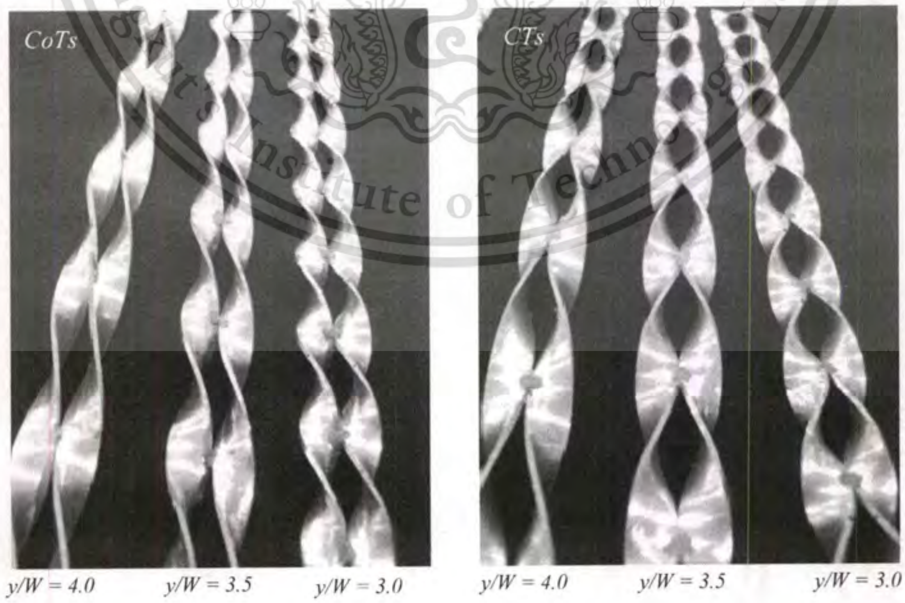


Figure 3.5: Pictorial view and sketch of *PTT* with parallel wings.

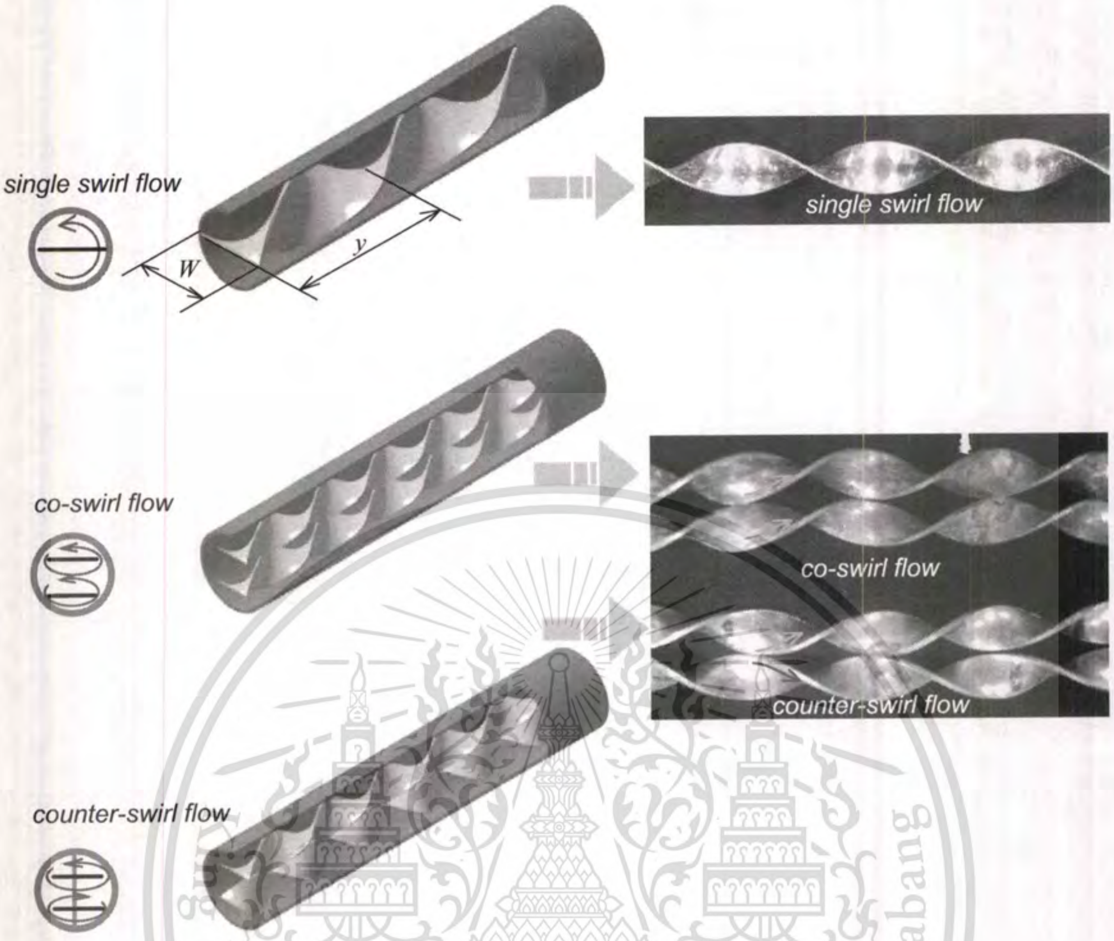
3.2.6 Twin twisted tapes

Twin twisted tapes are made of aluminium and have tape width (*W*) of 9.5 mm, tape thickness of 0.8 mm. and tape length (*L*) of 1000 mm. Dimensions of single twisted tape are same as those of the twin tapes, except tape width of 19 mm. Both twin counter/co twisted tapes were prepared with four different twist ratios, y/W = 2.5, 3.0, 3.5 and 4.0 where twist ratio is defined as twist length (*y*: 180°/twist length) to tape width (*W*). The single tapes *(ST)* or typical twisted tape (*TT*) were prepared with three twist ratios, y/W= 3.0, 3.5 and 4.0, for comparative investigation. Comparison of geometric details of the twin counter/co twisted tapes (*counter/co-swirl tapes*) and also a single twisted tape is shown in Figure 3.6(a-b). The tape thickness of 0.8 mm was chosen to avoid twin tape twisting difficulty of the thinner tape which was torn easily during the twisting operation. On the other hand, to avoid an additional friction in the system that might be caused the thicker tape. To produce the twisted tape, one end of a straight tape was clamped while another end was carefully twisted to ensure a desired twist length. For twin twisted tapes, two tapes were twisted separately and then welded together with a small aluminium wire. As shown in Figure 3.6(a-b), for the twin co twisted tapes (*CoTs*), both tapes were well aligned and positioned to be twisted in same direction to generate identical direction swirl called *co-swirl flow*. On the other hand, the twin counter twisted tapes (*CTs*), two tapes were aligned to be twisted in opposite directions to produce *counter-swirl flow*.



(a) photograph of twin twisted tapes

Figure 3.6: Pictorial view of twin twisted tape inserts: (a) twin twisted tapes and (b) tube fitted *TT/CoTs/CTs*.

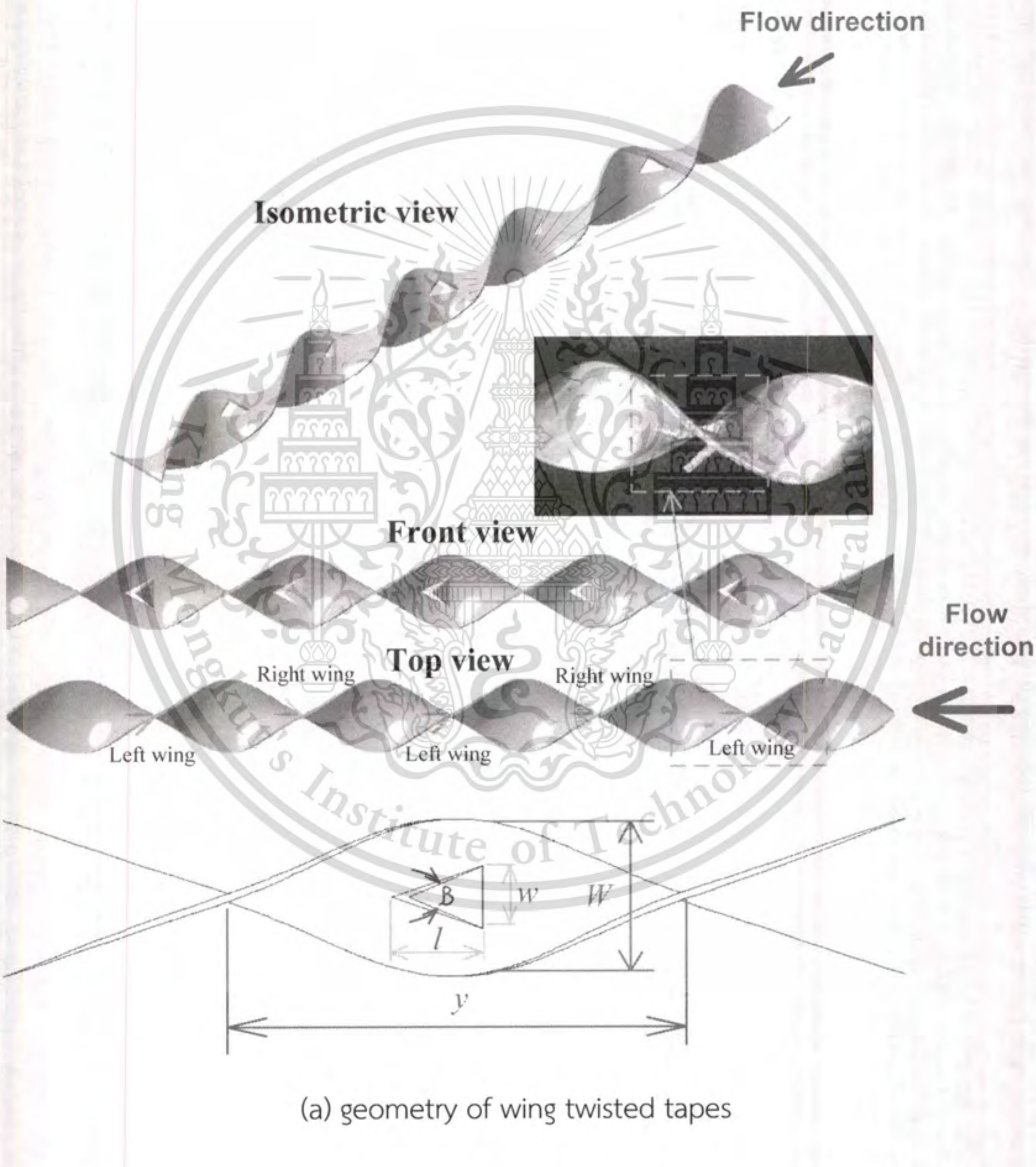


(b) tube fitted *CoTs/CTs*.

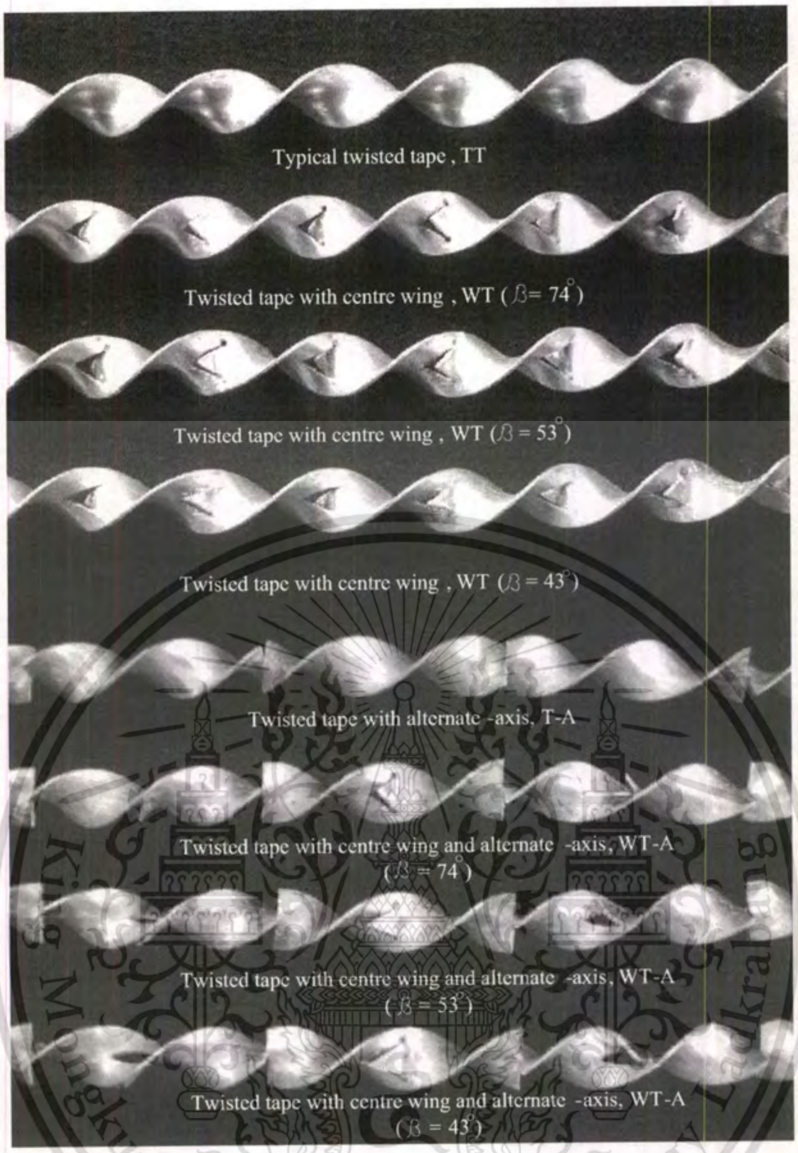
Figure 3.6: Pictorial view of twin twisted tape inserts: (a) twin twisted tapes and (b) tube fitted *TT/CoTs/CTs*. (*continued*)

3.2.7 Wing twisted tape with/without alternate axis

Wing twisted tape with/without alternate axis are made of aluminum strips with thickness of 0.8 mm, width (*W*) of 19 mm and length (*L*) of 1000 mm. The twist ratio *y/W* defined as ratio of twist length (*y*: 180°/twist length) to tape width of twisted tapes (*W*), was kept constant at 3.0. All tapes were initially formed as typical twisted tapes (*TT*). The twisted tape with centre wings (*WT*) was modified from the *TT* by punching delta-wings on the *TT* along the centre line (one delta-wing per twist length). The adjacent wings were open to the different sides of the twisted tape (left to right and vice versa) as shown in Figure 3.7(a-b). The attack angle of the wing was varied at 43°, 53° and 74°. Another modified twisted tape, twisted tape with alternate axes alone (*TA*) was formed by cutting on both sides of the *TT*, with the depth of cut of 4 mm on each side. Subsequently, the tape was twisted to angle difference of 180° to produce an alternate axis. The distance between each pair of alternate points was set at two twist lengths. The last modified twisted tape, the twisted tape with centre wings and alternate axes (*WT-A*) was formed by creating wings on the *TT* (one wing per two twist lengths) and then generating alternate axis in the middle between each pair of the wings. Note that the consequence of generating the alternate points on the *WT-A*, made wings appeared on four sides of the tape (left-right-top-bottom). The photograph of all tapes used in the present work, including one *TT*, one *TA*, three different *WT* and three different *WT-A*, is shown in Figure 3.7(b).



(a) geometry of wing twisted tapes



(b) photograph of various wing twisted tapes

Figure 3.7: Pictorial view of wing twisted tapes: (a) geometry of wing twisted tapes and (b) photograph of various wing twisted tapes. (*continued*)

3.3 Experimental set-up

3.3.1 Heat transfer set-up

The experiments were carried out using an experimental facility as shown in Figures 3.8-3.10. The test tube was made of copper with inner diameter of 19.5 mm (D), outside diameter of 21 mm (D_o), wall thickness of 1.5 mm, and length of 1000 mm (L). In the experiments, the twisted tapes were inserted at the core tube along the test section. The rather long tube provided sufficient contact surface between the tapes and tube wall for the firm attachment of the tapes to the tube without the need of any extra fitting. The tube was wound with electrical heating wire covered with ceramic beads. During the test, the tube was heated by continually winding flexible electrical wire providing a uniform heat flux condition. The electrical output power was controlled by a variac transformer to obtain a constant heat flux along the entire length of the test section and by keeping the current less than 9 Amp. The outer surface of the test tube was well insulated to minimize convective heat leak to surroundings. Moreover, necessary precautions were taken to prevent leakages from the system. In the experiment, the heat transfer losses from the test tube were found to be around 3 to 8% of the total heat input ($Q = IV$). The inner and outer temperatures of the water were measured at certain points with a multi-channel temperature measurement unit in conjunction with the resistance temperature detectors (RTDs). Fifteen thermocouples were tapped on the local wall of the tube and the thermocouples were placed round the tube to measure the circumferential temperature variation, which was found to be negligible. The mean local wall temperature was determined by means of calculations based on the reading of Copper-Constantan thermocouples.

The test loop consists of a water pump, data logger, pressure transmitter, thermocouple/RTD, rotameter, and heat transfer test section as depicted in Figure 3.8(a-b). In the apparatus setting above, the inlet water from a reservoir tank was discharged through the rotameter to measure the volumetric flow rate and passed to the heat transfer test section. For each test run, it was necessary to record the data of temperature, volumetric flow rate and pressure drop of the bulk water at steady state conditions in which the inlet water temperature was maintained constant at 27°C. The evaluations of Reynolds number and friction factor were based on bulk mean temperatures.

3.3.2 Dye injection set-up

The flow visualization was carried out using dye injection technique. The system consists of an acrylic tube fitted with a twisted tape, a water pump, a water filter, a settling chamber, four injection needles, and a digital camera as shown in Figure 3.11. A control valve and rotameter were respectively equipped downstream and upstream of the test section for controlling fluid flow rate. In the experiment, the water flow rate was maintained, corresponding to the Reynolds number of 2000. Four different dyes (red, blue, green and yellow) were injected at different position to indicate flow field associated by twisted tapes. The flow behaviors in the plain tube and the tube with the typical twisted tape, were also visualized for comparison.

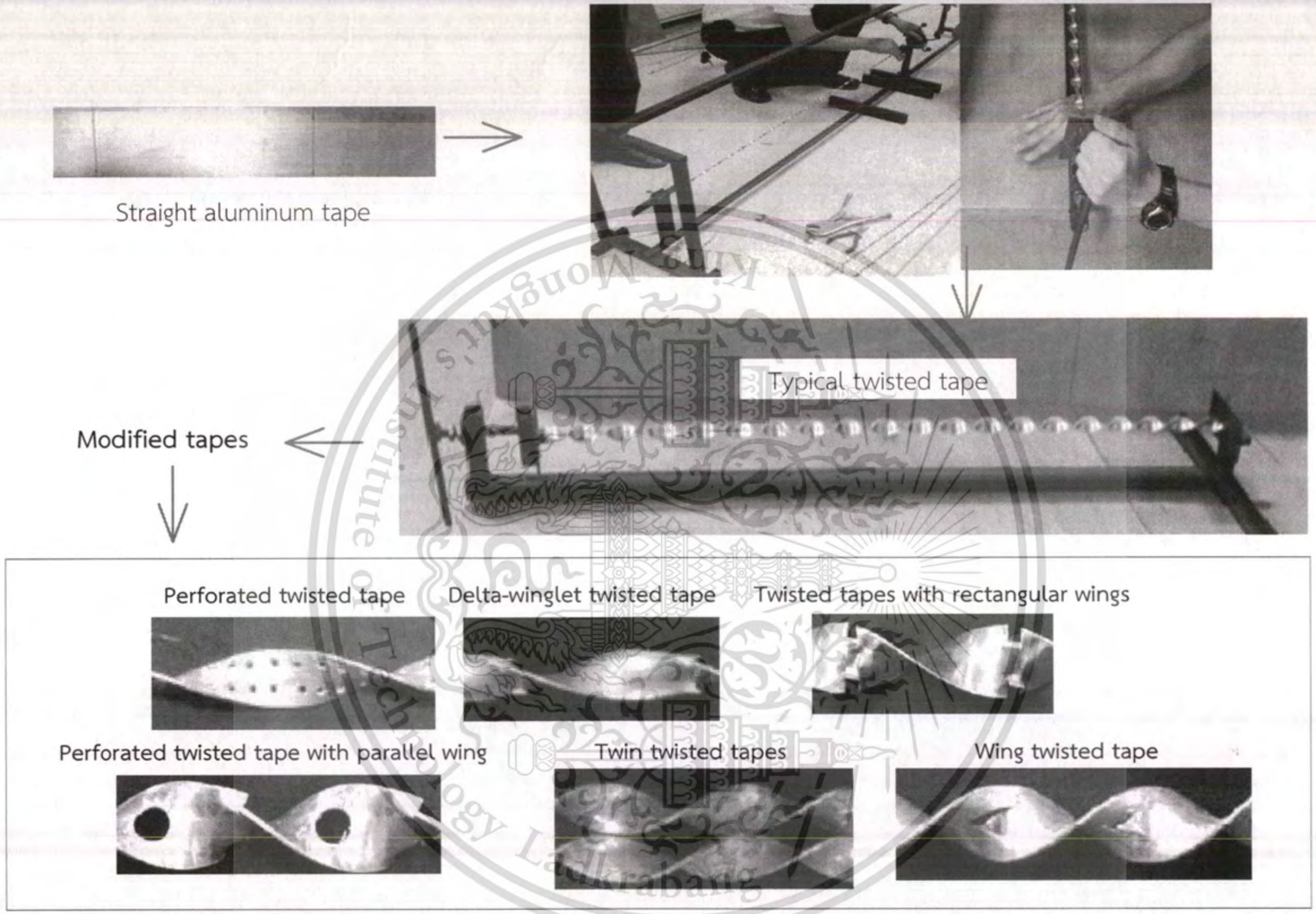


Figure 3.8: Production details of modified twisted tapes.

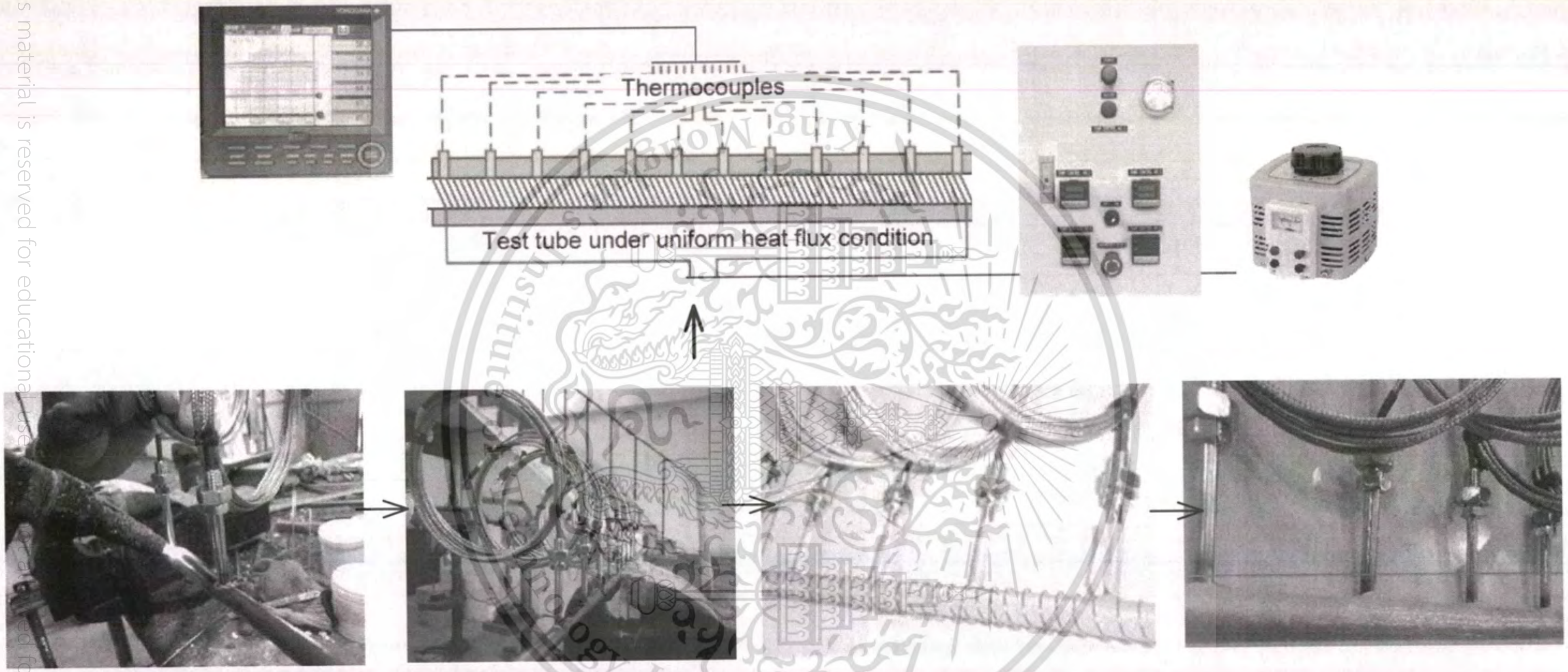
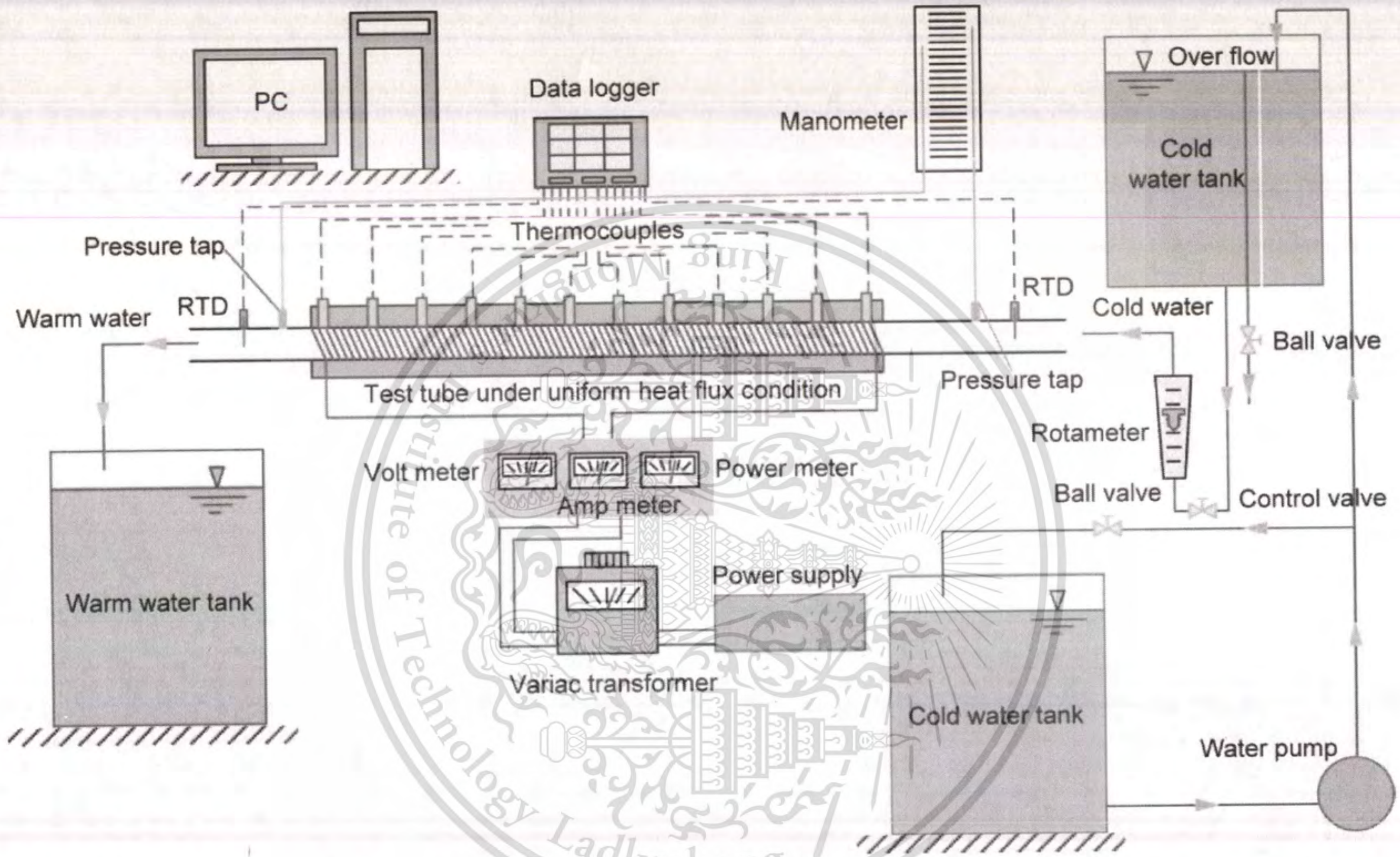
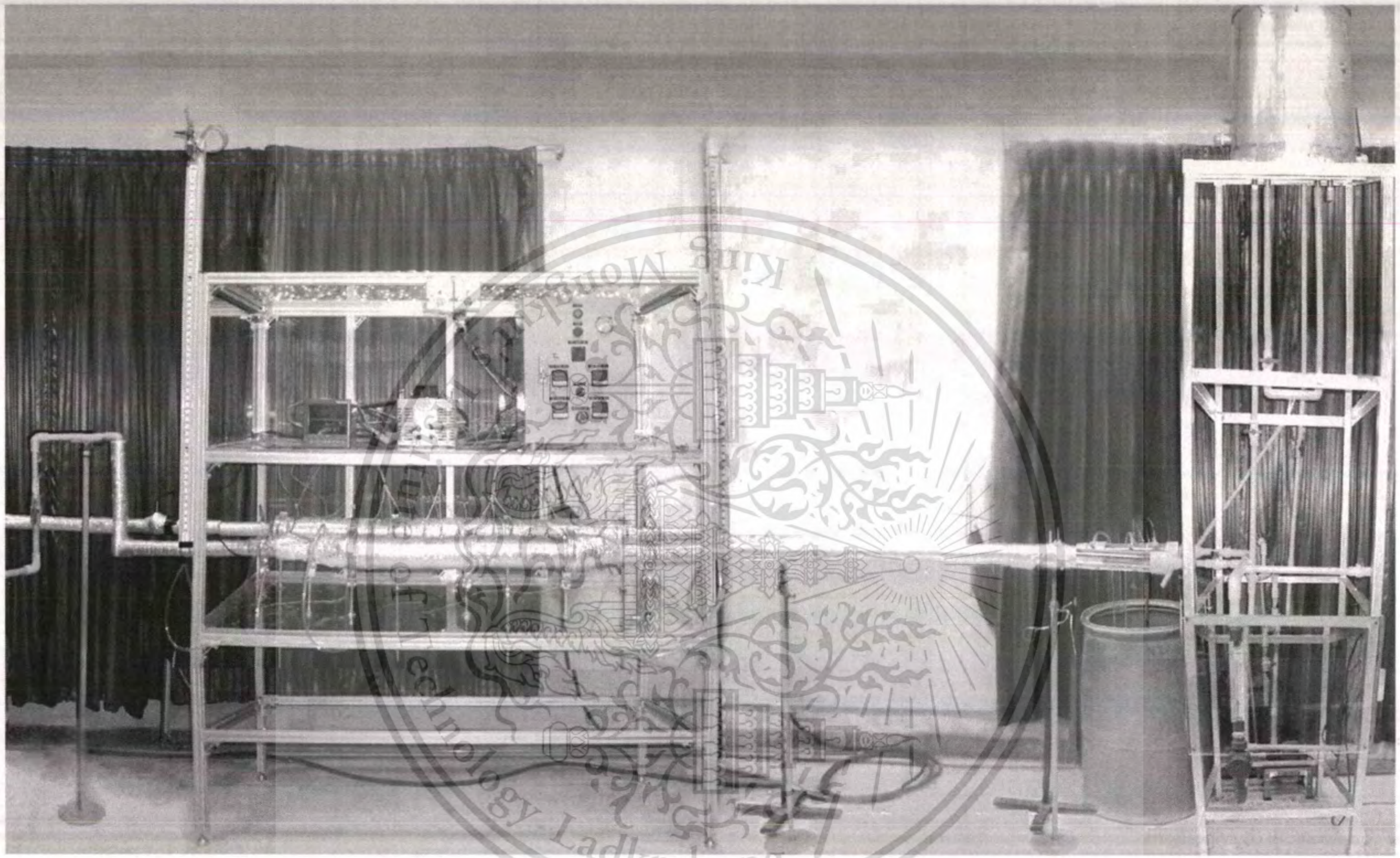


Figure 3.9: Production details of test section.



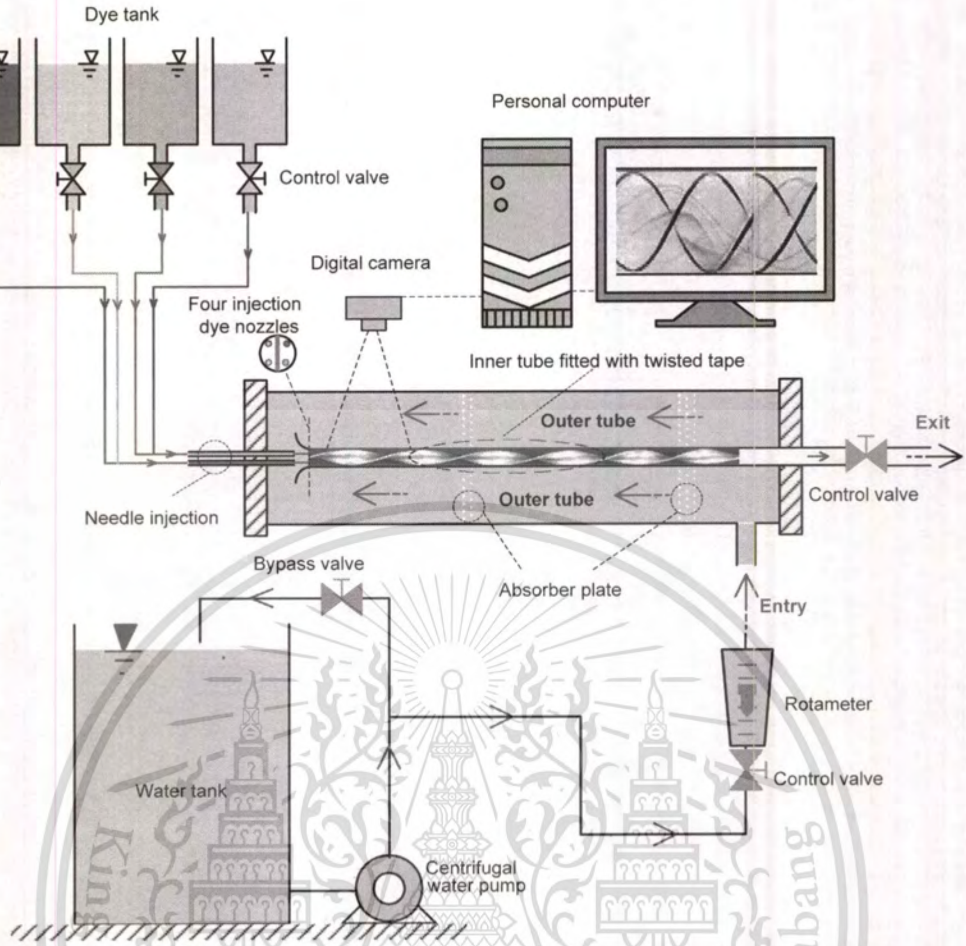
(a) schematic heat transfer set-up

Figure 3.10: Details of experimental set-up: (a) schematic heat transfer set-up and (b) photograph of experimental set-up.



(b) photograph of experimental set-up

Figure 3.10: Details of experimental set-up: (a) schematic heat transfer set-up and (b) photograph of experimental set-up. (continued)



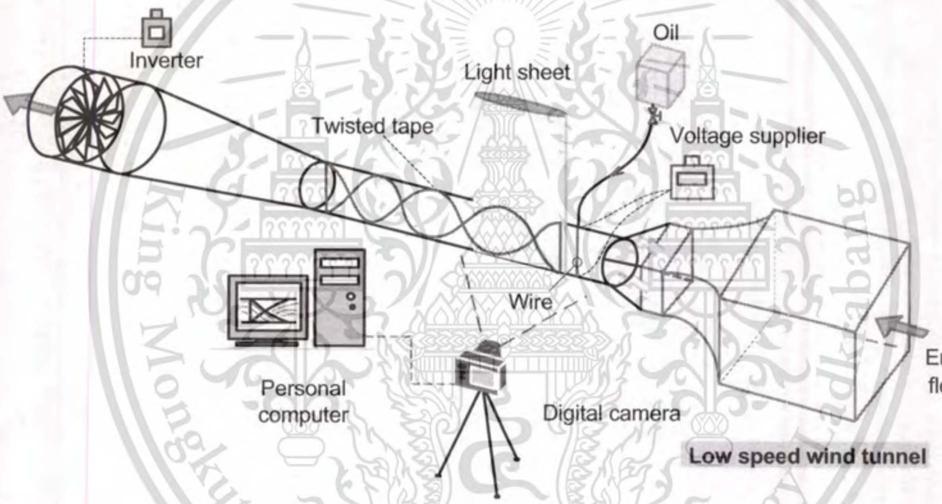
(a) schematic dye injection set-up

(b) photograph of dye injection set-up

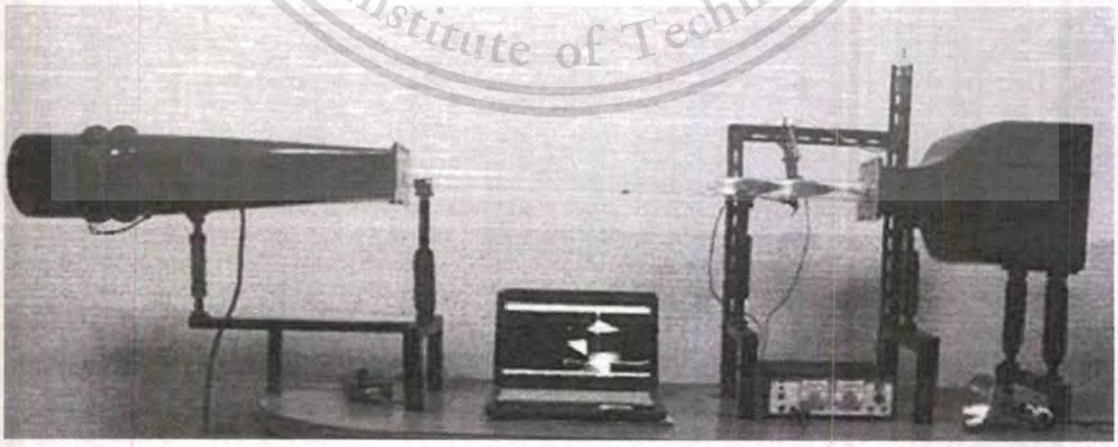
Figure 3.11: Details of experimental set-up: (a) schematic dye injection set-up and (b) photograph of dye injection set-up.

3.3.3 Smoke wire set-up

To gain a better understanding on flow phenomena induced by each twisted tape, the flow visualization was conducted. The smoke wire flow-visualization set-up is schematically illustrated in Figure 3.12(a-b). The facility consists of (1) a copper tungsten wire with diameter of 0.2 mm (2) a paraffin oil dropper (as smoke source), (3) a twisted tape insert, (4) a light bulb to lighten the test section, (5) a voltage supplier, (6) a digital camera (Nikon D80), (7) an inverter, (8) a hot wire anemometer, (9) a blower and test section made from acrylic tube, and (10) a personal computer. In the experiment, the paraffin oil was dropped on the copper tungsten wire. Then, heat was supplied to the oil through the voltage supplier to generate smoke and air was fed into the tunnel as a smoke carrier. The velocity of air in the tunnel was measured using a hot wire anemometer and air flow rate was kept constant, corresponding to the low Reynolds number.



(a) schematic smoke wire set-up



(b) photograph of smoke wire set-up

Figure 3.12: Details of experimental set-up: (a) schematic smoke wire set-up and (b) photograph of smoke wire set-up.

3.4 Data reduction

In experiments, water is used as a working fluid and flowed through a test tube under uniform heat flux condition. The heat transfer rate at steady state is assumed to be equal to the heat loss from the test section which may be expressed as:

$$Q_{water} = Q_{conv} \tag{3.1}$$

Then the heat transferred from the water flow can be drawn as,

$$Q_{water} = MC_p(T_{out} - T_{in}) \tag{3.2}$$

The heat transferred from the tube wall by convection may be also written as,

$$Q_{conv} = hA\left|\widetilde{T}_w - \left(\frac{T_{out} + T_{in}}{2}\right)\right| \tag{3.3}$$

where T_{out} is the outlet temperature of water flow, T_{in} is the inlet water temperature, T_w is the temperature of the locations in the centerline of the tube wall and A is the total surface area of the inside tube wall, which can be expressed as:

$$A = \pi DL \tag{3.4}$$

where D is the inner diameter of the test tube. The average heat transfer coefficient (h) for the test tube can be calculated by the combination of equations (3.2) and (3.3),

$$h = \frac{MC_p(T_{out} - T_{in})}{A\left|\widetilde{T}_w - \left(\frac{T_{out} + T_{in}}{2}\right)\right|} \tag{3.5}$$

In the experimental results, the average Nusselt number is defined as follow,

$$Nu = \frac{hD}{k} \tag{3.6}$$

where k is the local thermal conductivity of the fluid which calculated from the fluid properties at the local mean bulk fluid temperature ($T_b=(T_{out}+T_{in})/2$). The friction factor for tube with or without twisted tape can be calculated using pressure loss, ΔP, across the test length, L, via following equation:

$$f = \frac{1}{2}\frac{\Delta P}{L}\frac{D}{\rho U^2} \tag{3.7}$$

where ρ is the density at the mean bulk temperature, and U is the average velocity based on the inner diameter.

The Nusselt number, Prantl number, Reynolds number, and all of thermo-physical properties of the fluid were calculated of the basis of water properties corresponding to the bulk fluid temperature (T_b). The Reynolds number based on the total flow rate at the inlet of the test section is expressed as:

$$Re = \rho UD / \mu \tag{3.8}$$

where μ is the dynamic viscosity of the working fluid.

It is very important to validate the present plain tube data of Nusselt number and friction factor in fully developed straight/axial flow with the correlations from the previous studies. In the present work, Nusselt number and friction factor from the experimental data were compared with the correlations recommended by Dittus-Boelter and Blasius.

Heat transfer enhancement index or thermal performance factor is one of the key parameters in design of heat exchangers. A comparison between swirl flow (in tube with twisted tape or swirl generator) and straight flow (in plain tube) is usually made by comparing heat transfer coefficients at identical pumping power, since this is relevant to the operation cost. For a constant pumping power,

$$(\dot{V}\Delta P)_p = (\dot{V}\Delta P)_s \tag{3.9}$$

and the relationship between friction and Reynolds number can be expressed as:

$$(f\,Re^3)_p = (f\,Re^3)_s \tag{3.10}$$

$$Re_p = Re_s\left(f_s/f_p\right)^{1/3} \tag{3.11}$$

The thermal performance criteria of the test tube fitted with the modified twisted tape is defined as the ratio of the heat transfer coefficient for the inserted tube to that of the plain tube at the same level of pumping power and can be expressed as follows:

$$\eta = \left.\frac{h_s}{h_p}\right|_{pp} = (Nu/Nu_p)/(f/f_p)^{1/3} \tag{3.12}$$

where h_p, and h_s are the heat transfer coefficient for the plain tube and the plain tube with modified twisted tapes.

The objective of this investigation is to determine the fully developed heat transfer in a circular tube with modified twisted tape inserts (swirl generator). The independent parameters are Reynolds number (*Re*) and tape geometries depending on each modified twisted tape.

CHAPTER 4
Experimental Results

4.1 Open remarks

In this chapter, the experimental results including heat transfer (Nu), friction factor (f) and thermal performance factor behaviors in a circular tube heat exchanger fitted with different modified twisted tapes (perforated twisted tape, delta-winglet twisted tape, twisted tapes with triangular, rectangular and trapezoidal wings, perforated twisted tape with parallel wing, twin twisted tapes, wing twisted tape with/without alternate axis) are presented and discussed in the following subsections. The results obtained by using plain tube and the tubes equipped with typical twisted tape are also presented for comparison.

4.2 Validated on test

4.2.1 Plain tube

Plain tube data serve as a qualification for the facility and the procedure used over the range of Reynolds number studied. Qualification of the heat transfer rate and friction factor of the plain tube is evaluated by comparing the experimental data with the previous correlations under similar conditions. The results shown in Figure 4.1(a-b), reveal that the data for present plain tube are in good agreement with previous reports for both the Nusselt number and the friction factor correlations. As found, the mean absolute percentage deviation of the present experimental Nusselt number data is ±6% from the values predicted by Dittus-Boelter correlation, while the average absolute percentage deviation for friction factor data is ±10.4% from the values predicted by Blasius correlation. Comparison between the present results and correlations by Manglik and Bergles using the tube fitted with typical twisted tapes at three different twist ratios, *y/W* = 3.0, 3.5 and 4.0 is demonstrated in Fig. 4.2(a-b) for Nusselt number and Figure 4.3(a-b) for friction factor. Compared to the results obtained from the past investigation, the present Nusselt numbers are found to be slightly lower while the present friction factors are noticeably higher at the low Reynolds number region and then become comparable at the high Reynolds number. In addition, it is obvious that the present Nusselt numbers and friction factors reasonably agree well with the available correlations within ±15 to ±20% for both Nusselt number and friction factor.

4.2.2 Typical twisted tape

Figure 4.4 demonstrates the variation of the heat transfer rate with the Reynolds numbers in the tube with/without typical twisted tape (*TT*) swirl generators. For all cases, Nusselt number (or heat transfer rate) tends to increase with increasing Reynolds number, as a result of increased turbulence intensity. At given Reynolds number, the tubes with *TTs*, consistently possess superior heat transfer to the one without *TT* (the plain one). The augmented heat transfer is directly responsible by the continuous swirl flow generated by a twisted tape. The swirl flow repeatedly disrupts thermal boundary layer along the whole test length, resulting in consistent thermal boundary thinning and thus more efficient heat transfer through the tube wall. It is also found that Nusselt number increases as twist ratio (*y/W*) decreases (or twist number increases). This primarily caused by the promoted swirl violation, enlargement of fluid travel distance and also tangential contact area between a fluid and tube wall. The experimental results reveal that the mean Nusselt numbers attained by the uses of the tubes with *TTs* at *y/W* = 3, 4 and 5 are respectively 37.3%, 27.5% and 19.9%, higher than that of the plain tube. In the other word, the *TT* with smaller (*y/W* = 3) yields higher heat transfer rate than the ones with greater twist ratios (*y/W* = 4 and 5) by around 7.7% and 14.5%, respectively. It is noteworthy that for the range investigated, the increased percentage of Nusselt number is larger at higher Reynolds number.

Figure 4.5 presents the relationship between the friction factor and Reynolds number for tubes with/without typical twisted tape (*TT*) insert. For the tubes with all *TTs,* friction factor is diminished as Reynolds number increases, which is in manner similar to that of the plain tube. This is basically due to the dissipation of dynamic pressure caused by tube blockage and promoted interaction between the pressure forces and inertial forces in the boundary layer. It is also found that *TT* with small twist ratio induces higher friction factor than the one with large twist ratio. For the present range, friction factors associated by the *TTs* at *y/W* = 3, 4 and 5 are respectively 230%, 194% and 170%, higher than that of the plain tube.

Figure 4.6 presents the thermal performance factor for water flowing in the heat exchanger tube with typical twisted tape inserts with different twist ratios. Obviously, the effect of Reynolds number on thermal performance factor is primarily governed by that of friction factor since thermal performance factor tends to decrease with the rise of Reynolds number. In contrast, the effect of twist ratio is influential by that of Nusselt number. The *TT* with the smallest twist ratio *y/W* = 3 offers the highest thermal performance factor while that with the largest twist ratio *y/W* = 5 yields the lowest one. The maximum thermal performance factors provided by the tapes with twist ratios, *y/W* = 3, 4 and 5, are found to be 0.99, 0.96 and 0.92, respectively.

The present plain tube work of Nusselt number and friction factor correlations can be written as follows:

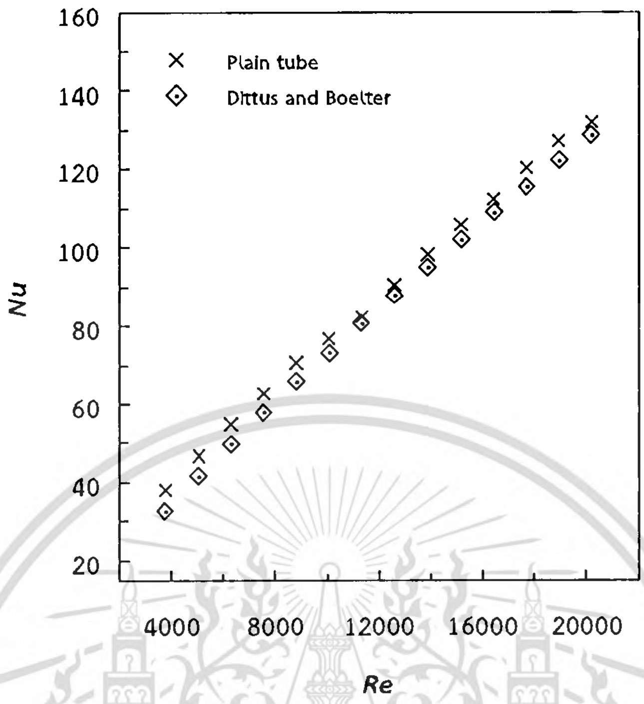
$$Nu = 0.0225Re^{0.802}Pr^{0.4} \tag{4.1}$$

$$f = 0.512Re^{-0.3} \tag{4.2}$$

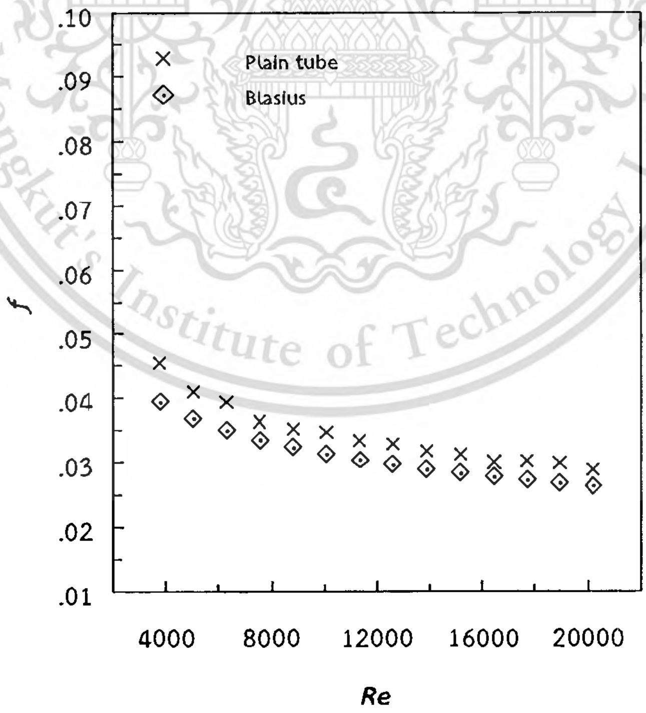
$$Nu = 0.126Re^{0.686}Pr^{0.4}(y/W)^{-0.265} \tag{4.3}$$

$$f = 11.72Re^{-0.454}(y/W)^{-0.391} \tag{4.4}$$

The present plain tube fitted with typical twisted tape work of Nusselt number and friction factor correlations can be written as above. The Prandtl number is varied between 3.0 and 5.0 depending on the bulk fluid temperature.

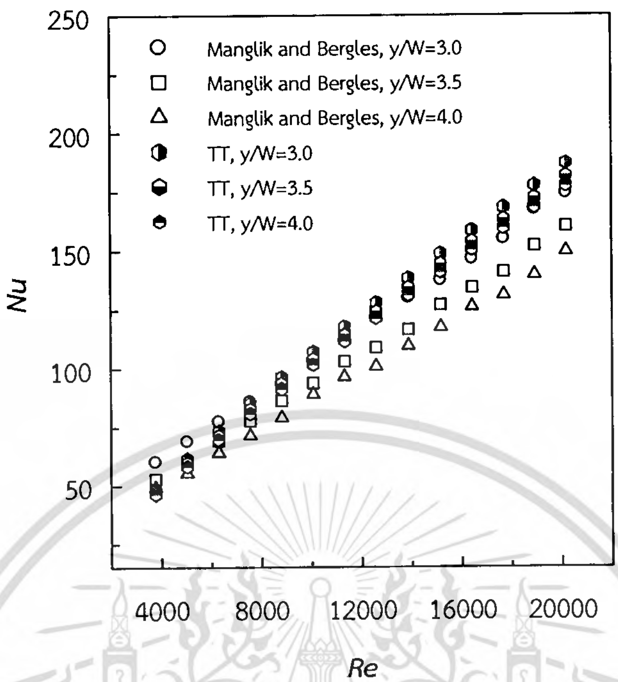


(a) Nusselt number

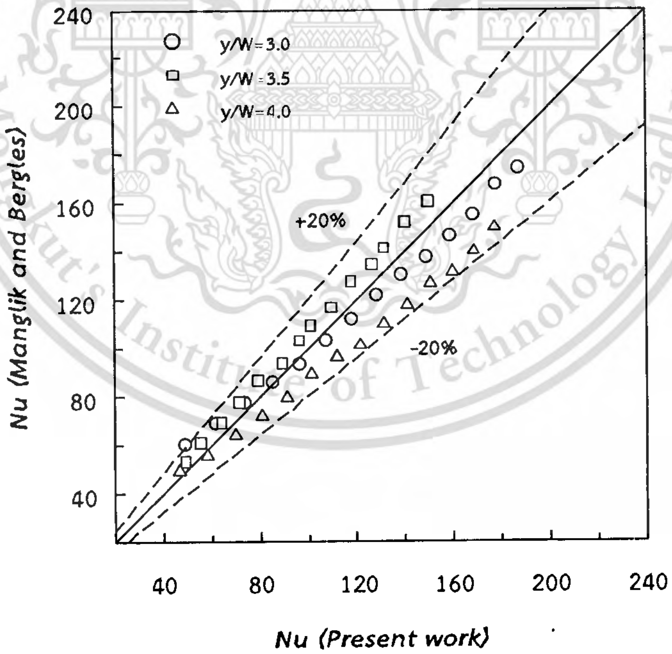


(b) friction

Figure 4.1: Verification of Nusselt number/friction factor for plain tube.



(a) Nusselt number



(b) Nusselt number

Figure 4.2: Verification of Nusselt number for plain tube fitted with twisted tape.

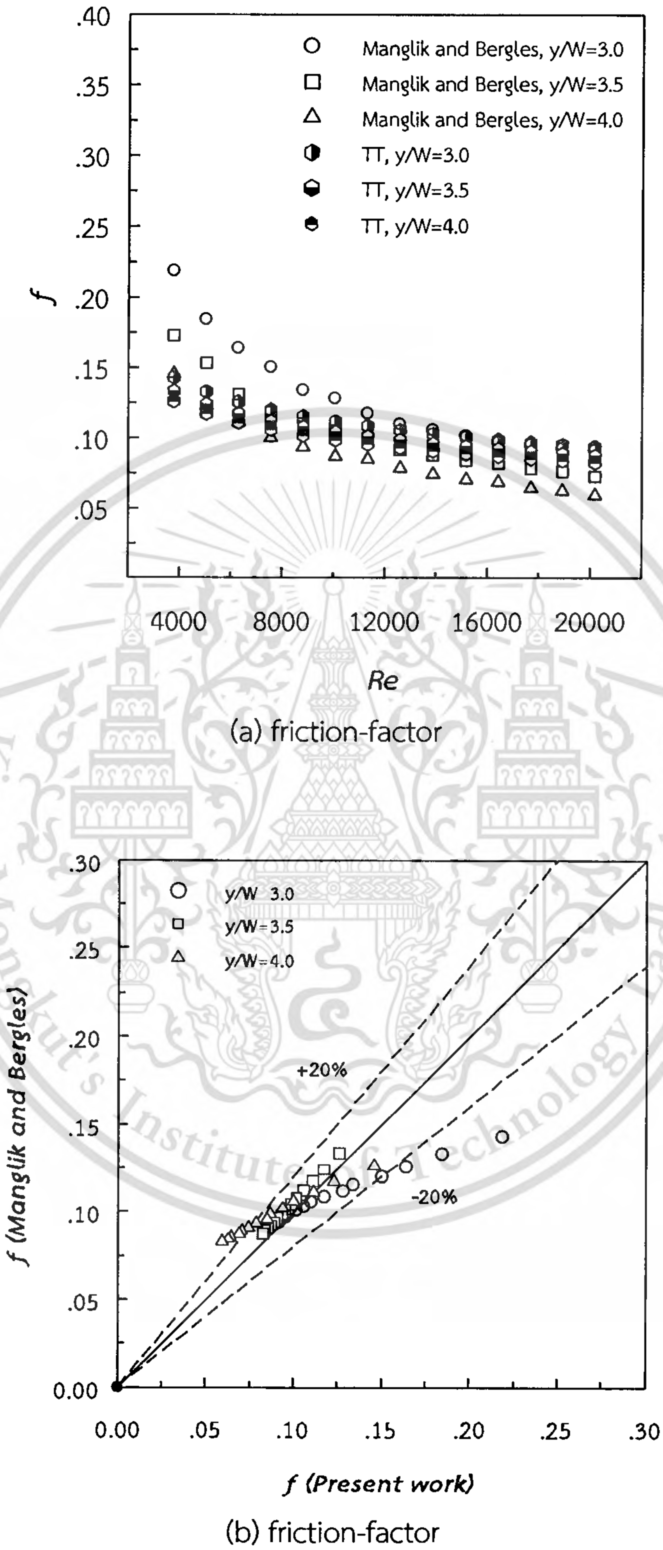


Figure 4.3: Verification of friction factor for plain tube fitted with twisted tape.

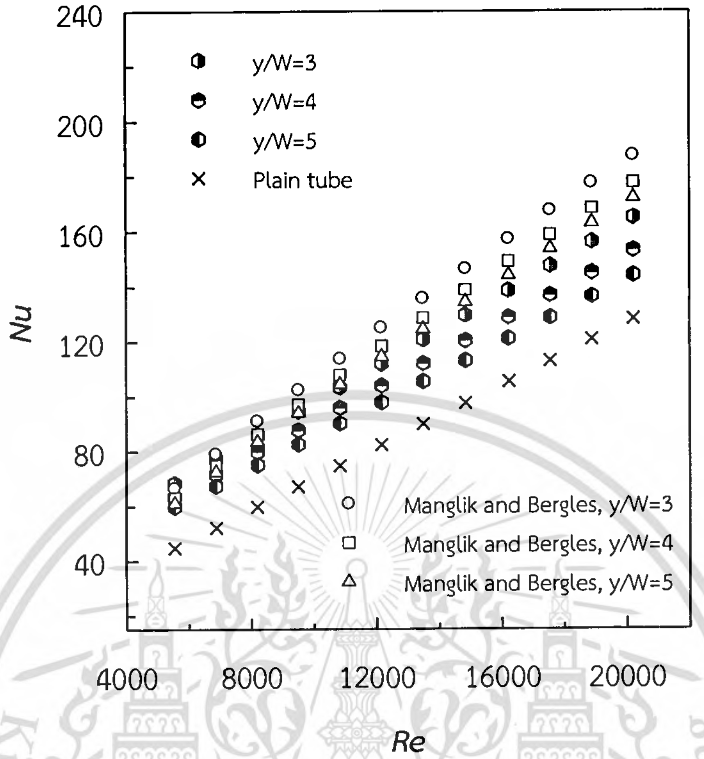


Figure 4.4: Effect of twist ratio on heat transfer rate.

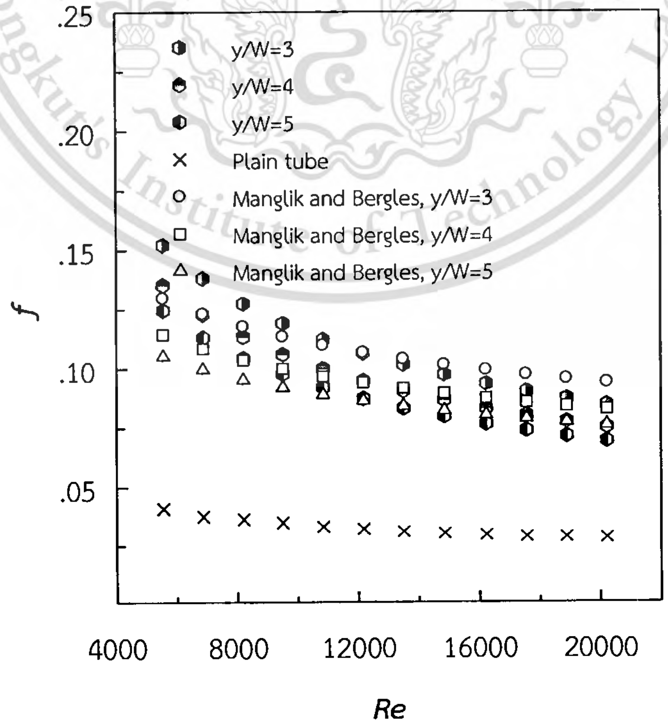


Figure 4.5: Effect of twist ratio on friction factor.

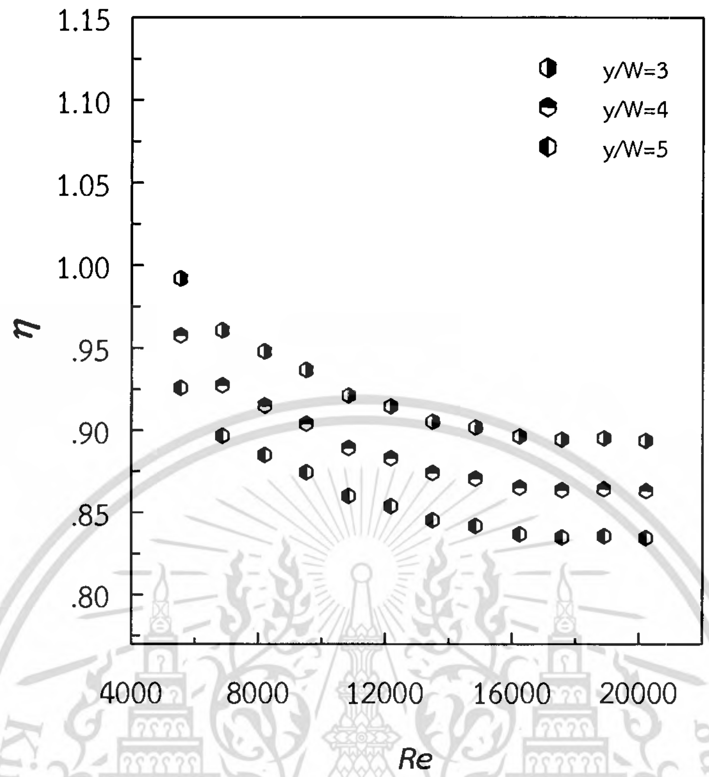


Figure 4.6: Effect of twist ratio on thermal performance factor.

4.3 Perforated twisted tape

Results of heat transfer in term of Nusselt number in a heat exchanger equipped with perforated twisted tape (*PT*) inserts at various pitch ratios (*s/W* = 0.4, 0.6 and 0.8) and tape-twist ratios (*y/W* = 3, 4 and 5) for *d/W* = 0.11, are demonstrated in Figures 4.7 and 4.8. The results show that Nusselt number increases with decreasing the twist ratio and space ratio of tapes. This high heat transfer rate at small twist ratio and pitch ratio is attributed to a stronger swirl flow and higher turbulent intensity imparted to the flow. Thus, a more efficient disturbance of a thermal boundary layer is induced along the pipe wall, resulting in a thinner thermal boundary layer which is more favourable for heat transfer across the layer. In addition, the tape with twist ratio and space ratio prolongs resident time of fluid flow which is directly responsible for the increase of overall heat transfer. For the range studied, the twisted tape with pitch ratios (*s/W*) of 0.4, 0.6 and 0.8 respectively enhance heat transfer rate up to 77.5%, 53% and 48% over that of the plain tube. The results on the effect of twist ratio (*y/W*) reveal that the Nusselt number provided by *PT* and *TT* with *y/W* = 3 are found to be around 10% and 17% higher than those given by the ones with *y/W* = 4 and 5.

The comparison between heat transfer augmentations offered by *PTs* and *TTs* is also presented in Figures 4.7 and 4.8. At similar conditions, *PTs* consistently provide higher Nusselt numbers than *TTs*. The superior heat transfer offered by *PTs* caused by an extra fluid turbulence around tube wall due to the presence of holes on the tapes, apart from a common swirl flow. Over the range investigated, *PTs* with *s/W* = 0.4, *d/W* = 0.11 and *y/W* = 3 give Nusselt number up to 25% higher than *TT*s at similar conditions (Reynolds number, inlet temperature and twist ratio, etc).

Effects of the *PTs* on the heat transfer enhancement with different perforation hole diameter ratios (*d/W* = 0.11, 0.14 and 0.17) at pitch ratios of *s/W* = 0.4, 0.6 and 0.8, are reported in Figures 4.9 and 4.10 The improvement of heat transfer may also be caused eddy motion and the strong turbulence between the tube wall and perforated tape. It is also observed that the heat transfer enhancement increases with the increase of the perforation hole diameter. This can be explained by the fact that PTs with a larger perforation hole diameter induce higher turbulent intensity imparted to the flow between the wall tube and edge of the tapes. The Nusselt numbers obtained by the use of the tape with the largest perforation hole diameter ratio (*d/W* = 0.17), are 65.6 to 94.9% than those of the plain tube. It should be noted that heat transfer rate in term of Nusselt number for *PTs* with *d/W* = 0.17 are respectively, 4.8 to 6.1% and 2.1 to 5.1% higher than those for the *PTs* with *d/W* = 0.11 and 0.14, depending on pitch ratio (*s/W*).

The variation of the pressure drop in terms of friction factor across the test with Reynolds number is shown in Figures 4.11 and 4.12. For all cases, friction factor decreases with increasing the Reynolds number. Similar to the results mentioned for Nuselt number, friction factor increases with decreasing the twist ratio and space ratio of tapes. At d/W = 0, friction factors generated by *PT* and *TT* with y/W = 3 are higher than those provided by the ones with twist ratios of y/W = 4 and 5 by 23% and 34%, respectively.

At similar operating conditions, *PTs* generate higher flow frictions than *TTs*. This is because of the extra dissipation of dynamic pressure of the fluid during act caused by the additional fluid disturbance due to the presence of holes on the tapes, resulting in an increase of interaction of the pressure force with inertial force around a velocity boundary layer. An average friction factor generated by *PT*s is found to be around 24% higher than that induced by *TTs*. In addition, the variation of the friction factor with Reynolds number for the *PTs* with different perforation hole diameter ratios (d/W) is demonstrated in Figures 4.13 and 4.14. At the given Reynolds number, the friction factors of *PT* with the largest perforation hole diameter ratio (d/W = 0.17) are found to be higher than that of the ones with d/W = 0.11 and 0.14 by around 15.5 to 24.3% and 7.8 to 10.4%, respectively, depending on pitch ratio.

The thermal performance factor above unity indicates an energy saving by using the tube with twisted tape relative to the plain tube. The variation between thermal performance factor and Reynolds number is presented in Figure 4.15. In general, thermal performance factor decreases with increasing Reynolds numbers. The results also reveal that the tapes with smaller pitch ratio and twist ratio offer higher thermal performance factor. At similar conditions, the factor is varied between 0.94 and 1.07, 0.91 and 1.05, and 0.9 and 1.03 for s/W = 0.4, 0.6 and 0.8, respectively. Effect of the perforation hole diameter ratio (d/W) on the thermal performance factor is given in Figure 4.16. Apparently, the thermal performance obtained by the use of the tape with a smaller perforation hole diameter ratio is considerably higher than that given by the one with a larger perforation hole diameter ratio for all Reynolds number value studied. This is principally influenced by a lower pressure drop penalty. The thermal performances obtained from the *PT* inserts with perforation hole diameter ratio of d/W = 0.11, are found to be around 1.7 to 2.1% and 0.8 to 1.3%, respectively, in comparison with those of the ones with d/W = 0.14 and 0.17. Although, the use of *PT* leads to a considerable increase in friction factor with respect to the use of *TT*, a reasonable increase in heat transfer rate results in slightly higher thermal performance. This indicates the beneficial gain by using *PT* over the use of *TT* in view point of energy saving.

The experimental results of the Nusselt number (Nu), friction factor (f) and thermal performance factor for plain tube equipped with perforated twisted tape (PT) with three different tape-twist ratios (y/W = 3, 4 and 5), spaced-pitch ratios (s/W = 0.4, 0.6 and 0.8) and perforation hole diameter ratios (d/W = 0.11, 0.14 and 0.17) are fitted by the following empirical equations:

$$Nu = 0.09Re^{0.768}Pr^{0.4}(y/W)^{-0.325}(s/W)^{-0.133}(d/W)^{0.114} \tag{4.5}$$

$$f = 9.03Re^{-0.272}(y/W)^{-0.631}(s/W)^{-0.204}(d/W)^{0.428} \tag{4.6}$$

$$\eta = 1.764Re^{-0.059}(y/W)^{-0.114}(s/W)^{-0.065}(d/W)^{-0.028} \tag{4.7}$$

Comparisons between the Nusselt number and the friction factor obtained from the present data with those calculated by the present correlations are portrayed in Figures 4.17-4.19. As found, the equations (4.6) to (4.8) are fitting the experimental data within ±3%, ±8% and ±2% for Nusselt number, friction factor and thermal performance, respectively.

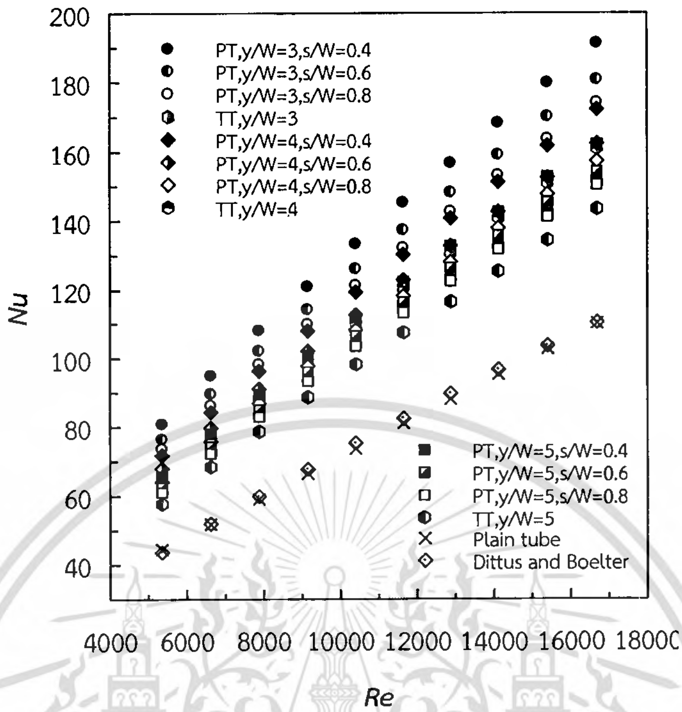


Figure 4.7: Effect of spaced pitch ratio on Nusselt number.

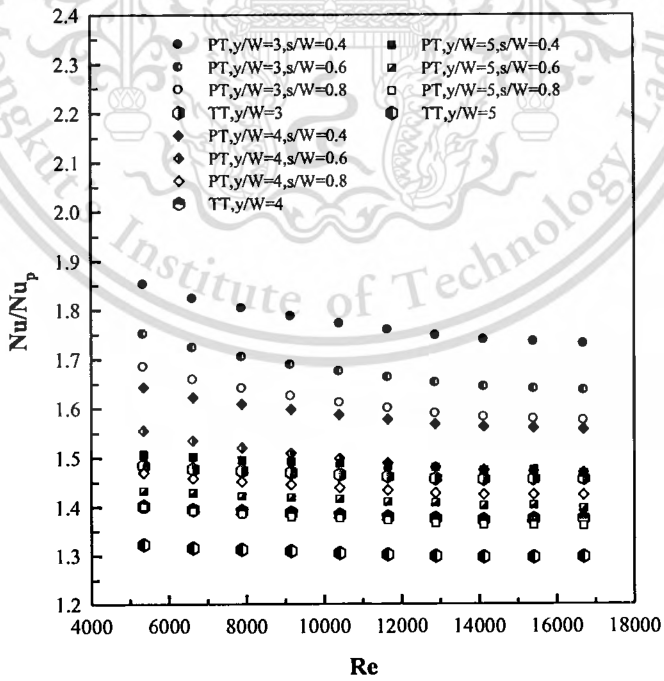


Figure 4.8: Effect of spaced pitch ratio on Nusselt number ratio.

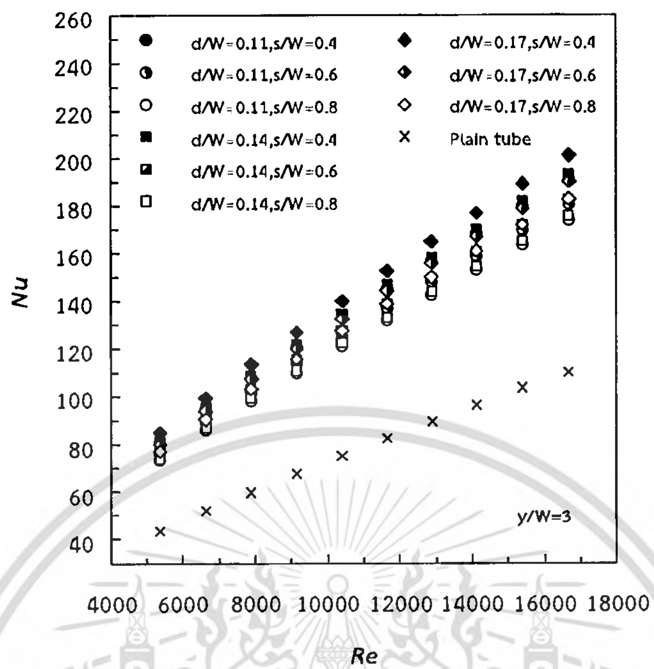


Figure 4.9: Effect of perforation hole diameter ratio (*d/W*) on Nusselt number.

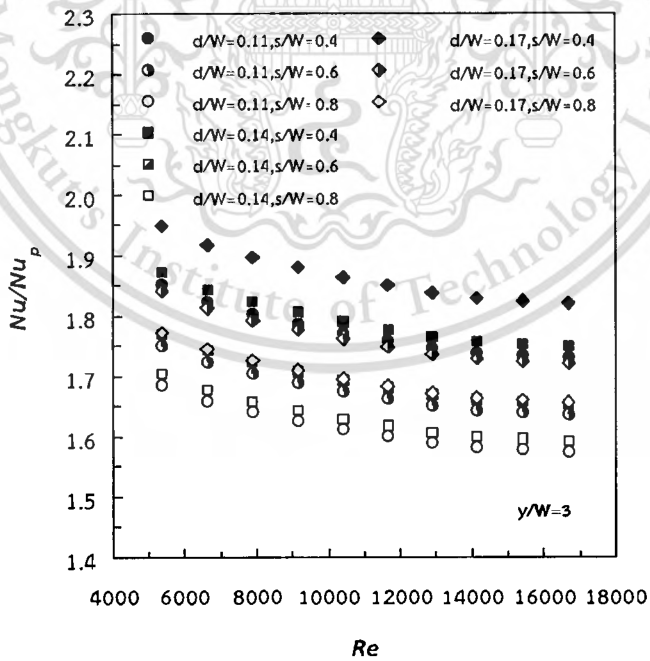


Figure 4.10: Effect of perforation hole diameter ratio (*d/W*) on Nusselt number ratio.

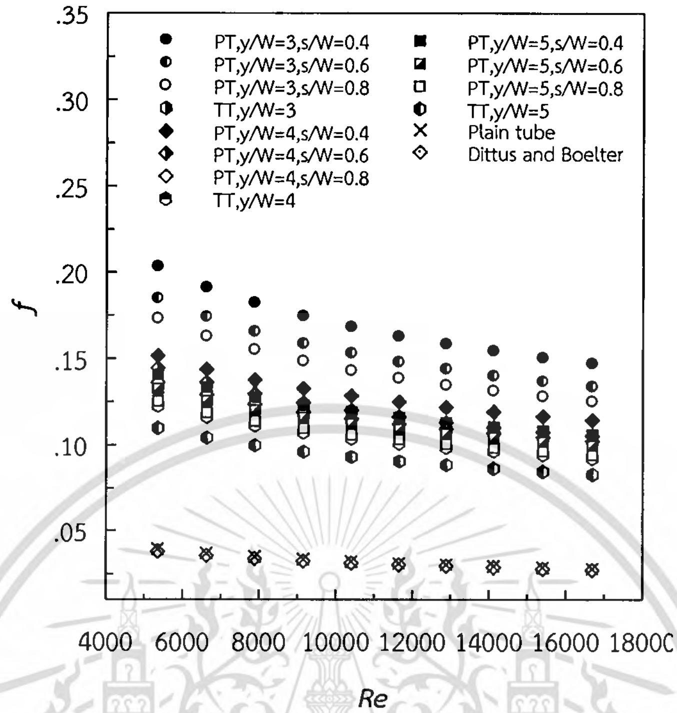


Figure 4.11: Effect of spaced pitch ratio on friction factor.

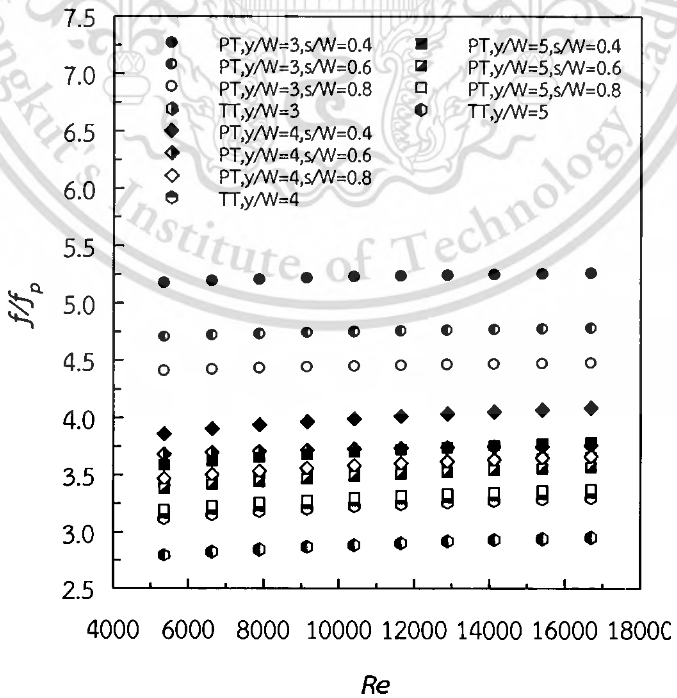


Figure 4.12: Effect of spaced pitch ratio on friction factor ratio.

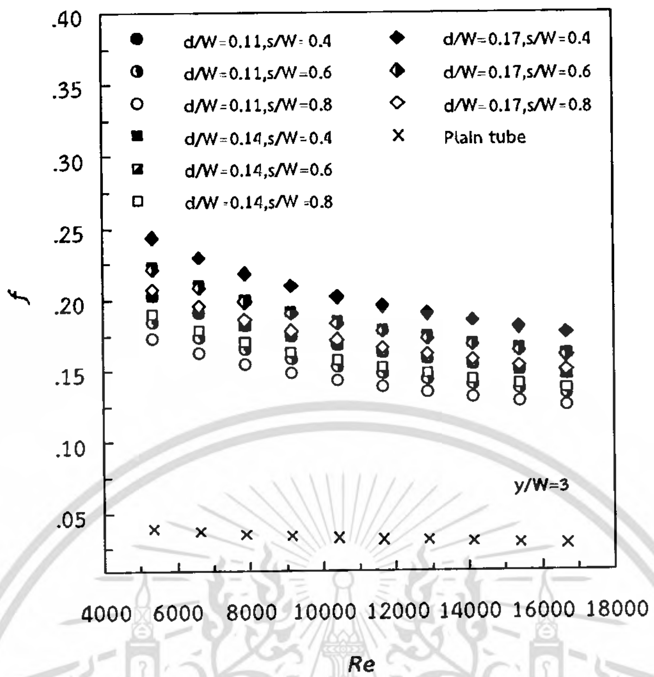


Figure 4.13: Effect of perforation hole diameter ratio (*d/W*) on friction factor.

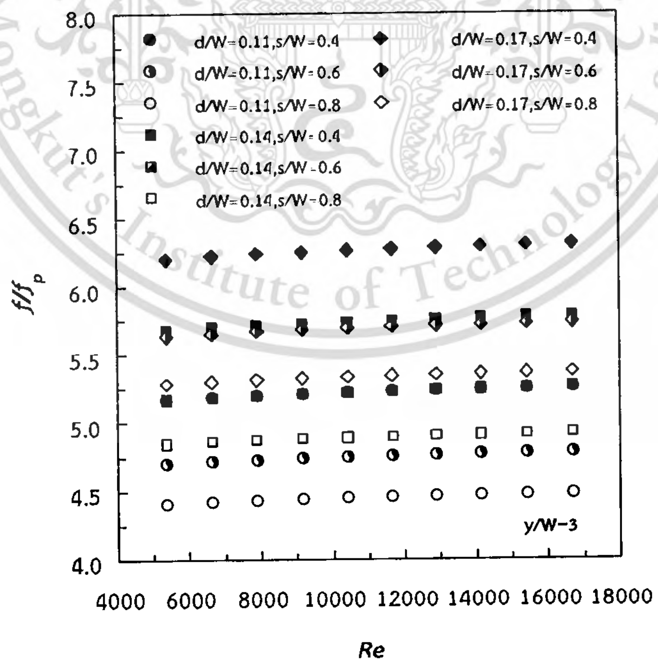


Figure 4.14: Effect of perforation hole diameter ratio (*d/W*) on friction factor ratio.

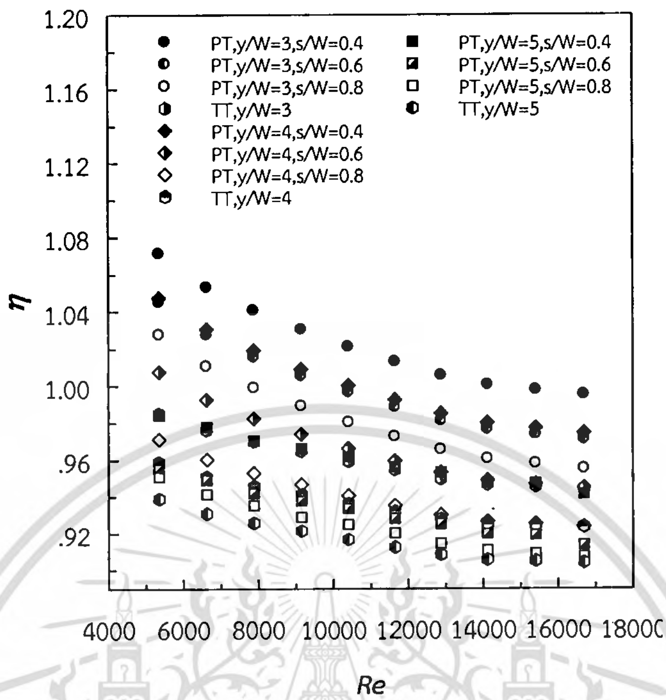


Figure 4.15: Effect of spaced pitch ratio on thermal performance factor.

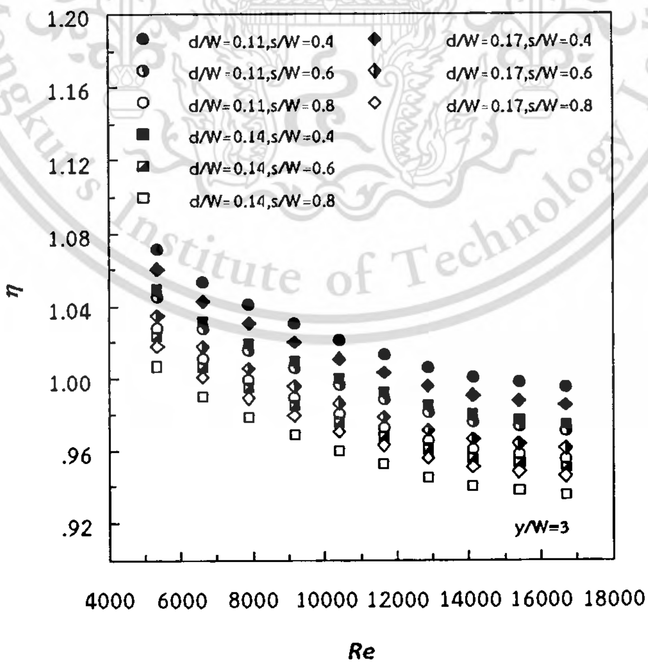


Figure 4.16: Effect of perforation hole diameter ratio (*d/W*) on thermal performance factor.

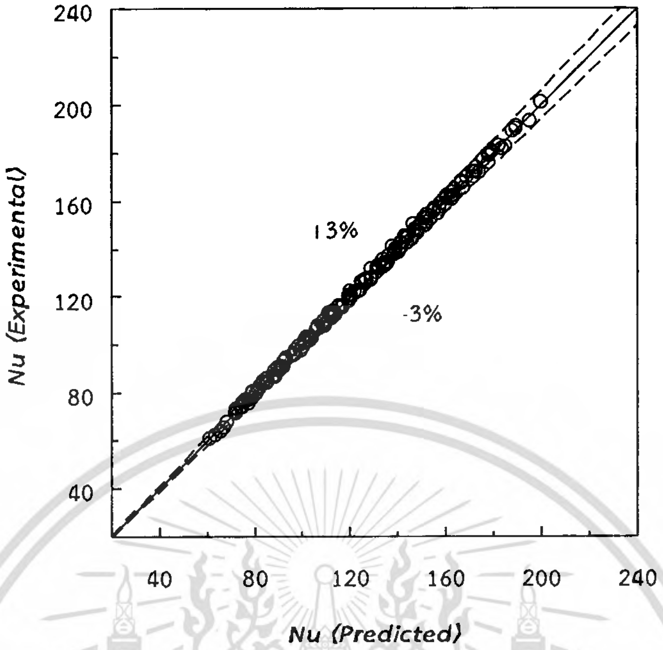


Figure 4.17: Validation of empirical correlations for Nusselt number.

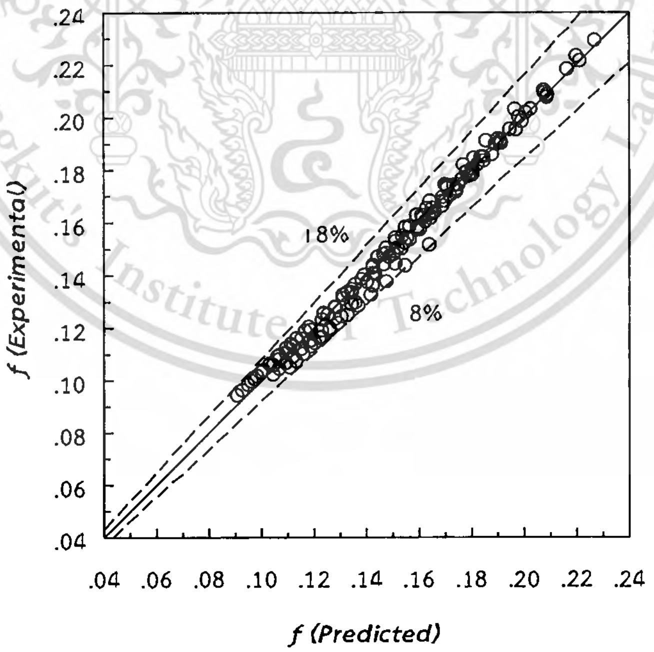


Figure 4.18: Validation of empirical correlations for friction factor.

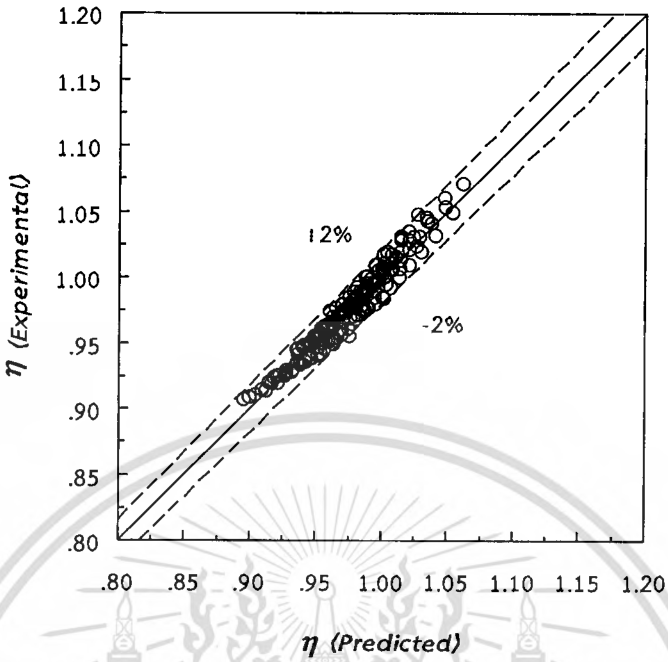


Figure 4.19: Validation of empirical correlations for thermal performance factor.

4.4 Delta-winglet twisted tape

In this section, heat transfer rate, friction factor and thermal performance factor behaviors in the tubes fitted with the delta-winglet twisted tapes (oblique or straight delta-winglet twisted tape) for different twist ratios (*y/W*) and depth of wing cut ratios (*DR*) are reported. The results obtained by using plain tube and the tubes equipped with typical twisted tape are also presented for comparison.

Variation of Nusselt number with Reynolds number in the tube fitted with delta-winglet twisted tape (*DWT*), the tube fitted with typical twisted tape (*TT*) and also the plain tube are presented in Figure 4.20(a-b). For all experiments, Nusselt number increases with increasing Reynolds number. This is attributed to the increase of heat convection and also swirl flow intensity. Apparently, Nusselt numbers in the tube with both oblique and straight delta-winglet twisted tapes are higher than those in the plain tube and the tube with typical twisted tape insert over the considered Reynolds number range. Refers to the past investigations, typical delta-winglet tapes were applied as vortex generators to amplify turbulence intensities and produce secondary flows near the tube wall. Therefore, the high heat transfer rate in the tube with both oblique and straight delta-winglet twisted tapes (*O-DWT and S-DWT*) for the present work may be caused by the synergy effect of (1) vortex circulation together with secondary flow generated by the delta-winglet part and (2) main swirl flow produced by twisted tape. This effect results in superior heat transfer enhancement over that caused by the typical tape (*TT*) which induces only swirling flow. In the range of the present experiments, the Nusselt numbers for tube equipped with *DWT*, are respectively 1.1 to 2.55 and 1.02 to 1.64 times of those for plain tube and tube with typical twisted tape insert.

Friction factors are plotted against Reynolds numbers in Figure 4.21(a-b). Not surprisingly, the friction factors obtained from the tube with delta-winglet twisted tape (*DWT*) are substantially higher than those from the plain tube and also the tube with typical twisted tape insert (*TT*). This is a result of additional blockage and disturbance to the flowing stream generated by the delta-winglet part of *DWT* over those induced by typical twisted tape. Over the range considered, the friction factors for tube equipped with *DWT* are respectively 2.5 to 7.02 and 1.08 to 1.95 times of those for plain tube and tube with typical twisted tape insert.

Figure 4.20(a-b) shows variation of Nusselt number with Reynolds number in tube with oblique delta-winglet twisted-tape (*O-DWT*) and straight delta-winglet twisted tape (*S-DWT*) for various twist ratios (*y/W* = 3, 4 and 5). At the given Reynolds number, the Nusselt number consistently increases with the decrease of twist ratio (*y/W*). This is due to the fact that, the tape with smaller twist ratio (*y/W*) induces stronger swirl/turbulent intensity, and also gives longer flowing path, leading to longer residence time and thus more efficient heat transfer compared to that with larger twist ratio (*y/W*). Mean Nusselt numbers for the *S-DWT* with twist ratio, *y/W* of 3, 4 and 5 are respectively, 1.6, 1.4 and 1.23 times of that for the plain tube and 1.2, 1.16 and 1.1 times of that in the tube equipped with typical twisted tape (*TT*). With a similar trend, mean Nusselt numbers for the *O-DWT* with twist ratios, *y/W* of 3, 4 and 5 are respectively, 1.66, 1.45 and 1.31 times of that for the plain tube and 1.25, 1.2 and 1.17 times of that for the tube with typical twisted tape insert. Depending on the Reynolds number, Nusselt numbers from using the *O-DWT* and *S-DWT* with twist ratio (*y/W*) of 3 are around 7% to 29 % and 15.8% to 45.6% greater than those from the tapes with twist ratios (*y/W*) of 4 and 5, respectively.

Figure 4.21(a-b) shows variation of friction factor with Reynolds number for oblique delta-winglet twisted-tape (*O-DWT*) and straight delta-winglet twisted tape (*S-DWT*) at various twist ratios (*y/W* = 3, 4 and 5). As found, the friction factor increases twisted tape with decreasing twist ratio (*y/W*). This is a consequence of the reasons mentioned for Nusselt number, in which twisted tape with shorter twist length provides longer flowing path, resulting in larger tangential contact between the flowing stream and tube surface. In the range of present experiments, mean friction factors for the *S-DWT* with twist ratios, *y/W* of 3, 4 and 5 are respectively, 4.35, 3.86 and 3.4 times of that for the plain tube and 1.25, 1.34 and 1.4 times of that for the tube with typical twisted tape insert. For a tube with *O-DWT* , friction factors with twist ratios, *y/W* of 3, 4 and 5 are respectively, 4.68, 4.18 and 3.67 times of that for the plain tube and 1.33, 1.45 and 1.52 times of that for the tube with typical twisted tape insert. In addition, friction factors for *O-DWT* and *S-DWT* with twist ratio (*y/W*) of 3 are approximately 3.5% to 22.8 % and 17.9% to 37.8% over those for the tapes equipped with twist ratios (*y/W*) of 4 and 5, respectively.

Effects of the of depth of wing cut ratios (*DR* = *d/w* = 0.11, 0.21 and 0.32) on the Nusselt number and friction factor are presented in Figure 4.20(a-b) and Figure 4.21(a-b). Refer to Figure 4.20(a-b), for both tubes with *S-DWT* and *O-DWT*, Nusselt number increases with increasing wing cut ratios (*DR*). As the depth of wing cut increases, longer wing part protruded into flowing stream. Consequently, the flow is more seriously disturbed and degree of turbulence intensity is raised. This leads to more effective heat transfer enhancement. The results show that mean Nusselt numbers for the *S-DWT* and *O-DWT* with the highest depth of wing cut ratios, *DR* = 0.32 are respectively, 1.25 to 2.16 and 1.35 to 2.55 times of that for the plain tube and 1.16 to 1.39 and 1.29 to 1.64 times of that for the tube with typical twisted tape. In addition, the average Nusselt number for employing the tapes with of depth of wing cut ratio *(DR)* of 0.32 is respectively, 17.3 % and 9.7% higher than those for the tapes depth of wing cut ratios of 0.11 and 0.22.

Figure 4.21(a-b) indicates that the tape insert with larger depth of wing cut ratio (*DR*) induces higher friction factor. This is caused by increasing extent of flow obstruction by the wing parts of the twisted tape. Mean friction factors for the *S-DWT* and *O-DWT* with the highest depth of wing cut ratio, *DR* = 0.32 are respectively, 3.22 to 5.85 and 3.62 to 7 times of that in the plain tube and 1.31 to 1.63 and 1.51 to 1.95 times of that in the tube with typical twisted tape. In addition, the average friction factor for employing the tape with depth of wing cut ratio, *DR* = 0.11 is found to be 10.9% and 25.2% lower than that for the tapes with *DR* = 0.21 and 0.32.

Comparisons shown in Figures 4.22 and 4.23 reveal that the oblique delta-winglet twisted tapes (*O-DWT*) give higher Nusselt number and also friction factor than the straight delta-winglet twisted tapes (*S-DWT*), when other parameters are controlled. These results are directly related to geometries of both tape arrangements. As depicted in Figure 4.21(a-b), with the same depth of wing cut, the *O-DWT* can be more effectively twisted, thus the tape possesses larger protruding part than the *S-DWT*. Consequently, greater heat transfer rate and also friction factor are obtained for the *O-DWT*. The calculated data show that the mean Nusselt number and friction factor for the *O-DWT* are respectively, 4.2% and 7.8% higher than those for the *S-DWT*.

In general, the thermal performance factor above unity indicates that the effect of heat transfer enhancement due to the turbulator (or enhancing device) is more dominant than the effect of rising friction and vice versa. The thermal performance factors for the oblique delta-winglet twisted tapes (*O-DWT*) and straight delta-winglet twisted tapes (*S-DWT*) which calculated from equations (4.17) and (4.20) based on the same pumping power, are plotted versus Reynolds number in Figures 4.24(a-b) and 4.25. The performance factors in the tube equipped with typical twisted tape (*TT*) are also plotted for comparison. At the same Reynolds number, the performance factors for both *O-DWT* and *S-DWT* are found to be greater than those for the typical twisted tape (*TT*). The performance factors for all twisted tapes tend to decrease with increasing Reynolds number. This suggests that both *DWTs* are feasible in terms of energy saving at higher Reynolds numbers compared to typical twisted tape (*TT*) which is suitable only for low Reynolds number as suggested by previous works.

Similar to the effects found for Nusselt number and friction factor, the performance factor increases with decreasing twist ratio (*y/W*) and increasing depth of wing cut for all given Reynolds numbers. In addition, the performance factor for the

O-DWT is greater than that in the *S-DWT* tubes, for the similar operation test conditions. Over the range considered, the values of performance factors obtained from using *O-DWT* and *S-DWT* are about 0.92 to 1.24 and 0.88 to 1.21, respectively. The maximum thermal performance factor of 1.24 is achieved by using of *O-DWT* with *DR* = 0.32 and *y/W* = 3 at Reynolds number of 3000.

Empirical correlations for Nusselt number, friction factor, thermal performance factor are developed for the tube with delta-winglet twisted tape inserts in the range of Reynolds number between 3000 and 27,000, *Pr* = 4.91 to 5.57, twist ratio (*y/W* = 3, 4 and 5), and depth of wing cut ratios *(DR = d/w =* 0.11, 0.21 and 0.32*)* as follows.

For oblique delta-winglet twisted tapes (*O-DWT*):

$$Nu = 0.18Re^{0.67}Pr^{0.4}(y/W)^{-0.423}(1+(d/W))^{0.982} \tag{4.8}$$

$$f = 24.8Re^{-0.51}(y/W)^{-0.566}(1+(d/W))^{1.87} \tag{4.9}$$

$$\eta = 2.04Re^{0.042}(y/W)^{-0.261}(1+(d/W))^{0.45} \tag{4.10}$$

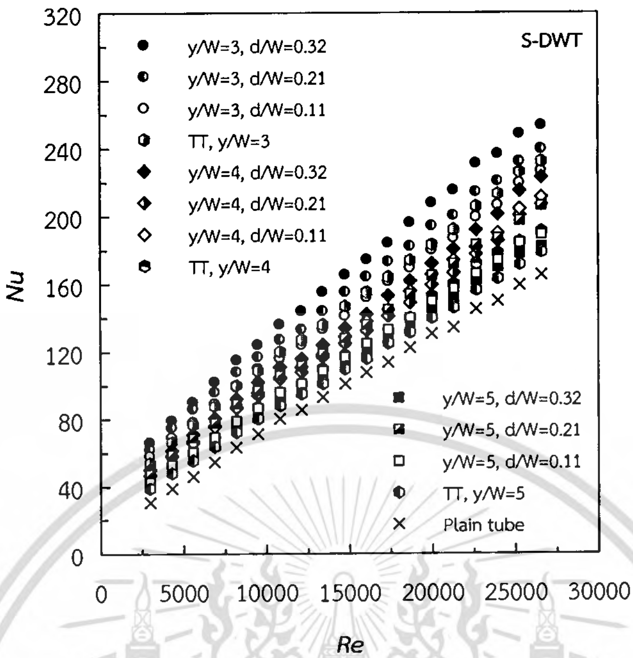
For straight delta-winglet twisted tapes (*S-DWT*):

$$Nu = 0.184Re^{0.675}Pr^{0.4}(y/W)^{-0.465}(1+(d/W))^{0.76} \tag{4.11}$$

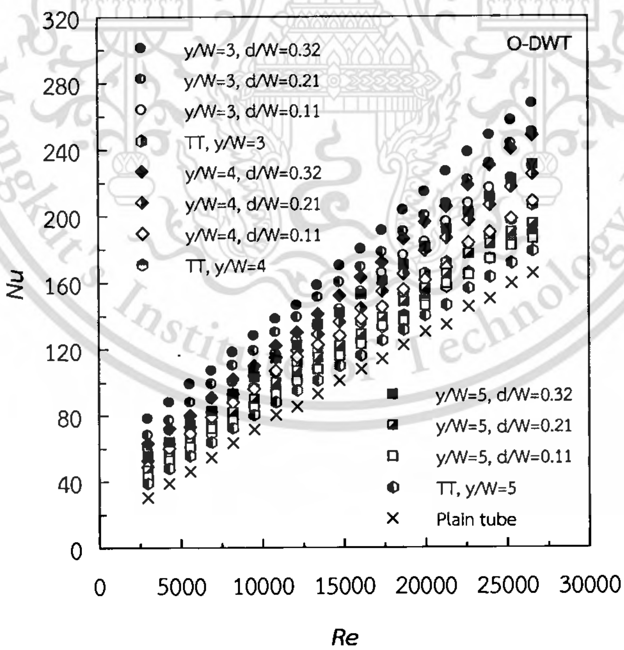
$$f = 21.7Re^{-0.45}(y/W)^{-0.564}(1+(d/W))^{1.41} \tag{4.12}$$

$$\eta = 2.164Re^{-0.0435}(y/W)^{-0.304}(1+(d/W))^{0.356} \tag{4.13}$$

The experimental data and predicted data of the correlation by equations (4.8) to (4.13) are compared in Figures 4.26 and 4.27. It appears that the mean absolute percentage deviation of the present experimental Nusselt number data is within 10% for equations (4.15) and 10% for equations (4.18) from the values predicted by the present correlations while that of the friction factor is within ±10% and ±10% for equations (4.16) and (4.19), respectively.

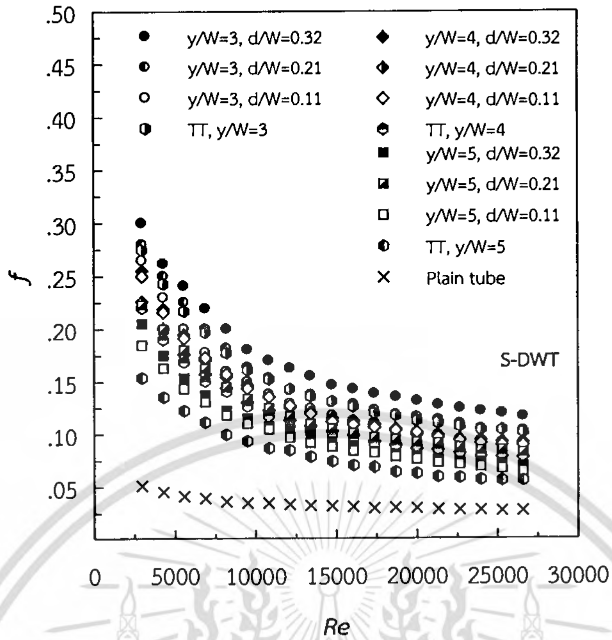


(a) *S-DWT*

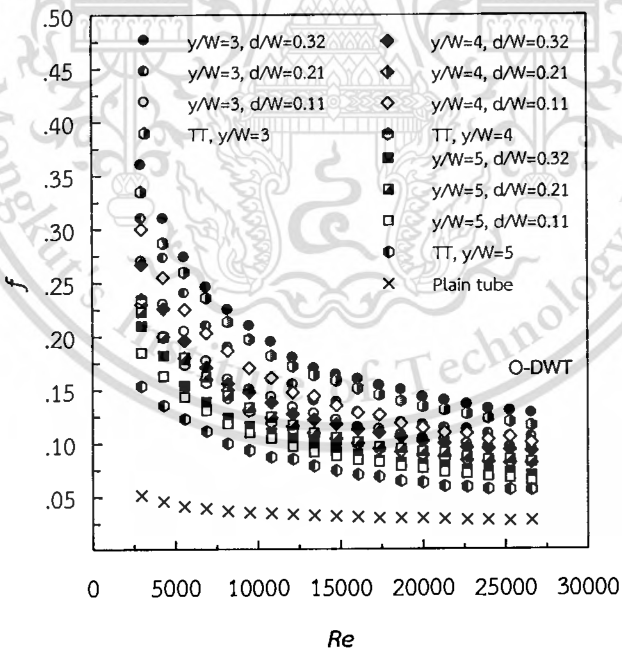


(b) *O-DWT*

Figure 4.20: Effect of delta-winglet twisted tape on Nusselt number.



(a) *S-DWT*



(b) *O-DWT*

Figure 4.21: Effect of delta-winglet twisted tape on friction factor.

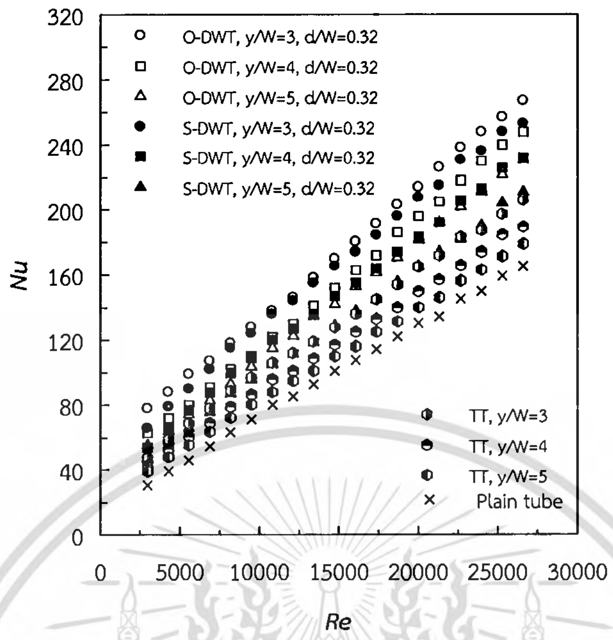


Figure 4.22: Effect of straight and oblique wings on Nusselt number.

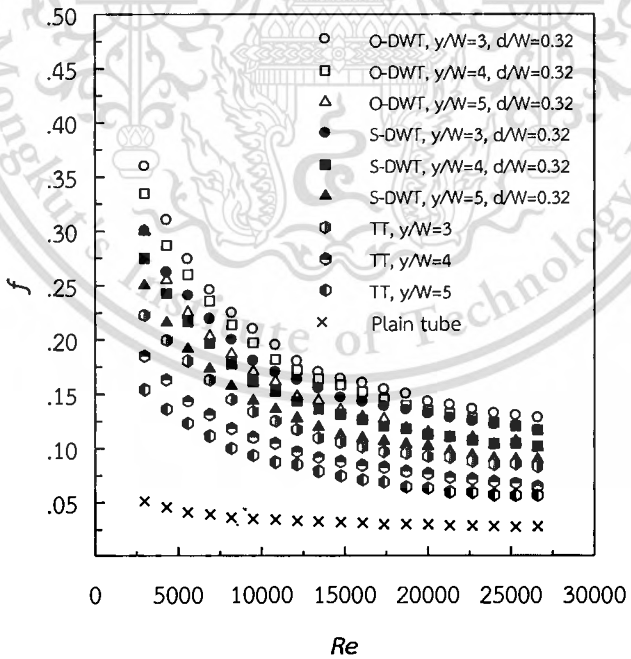
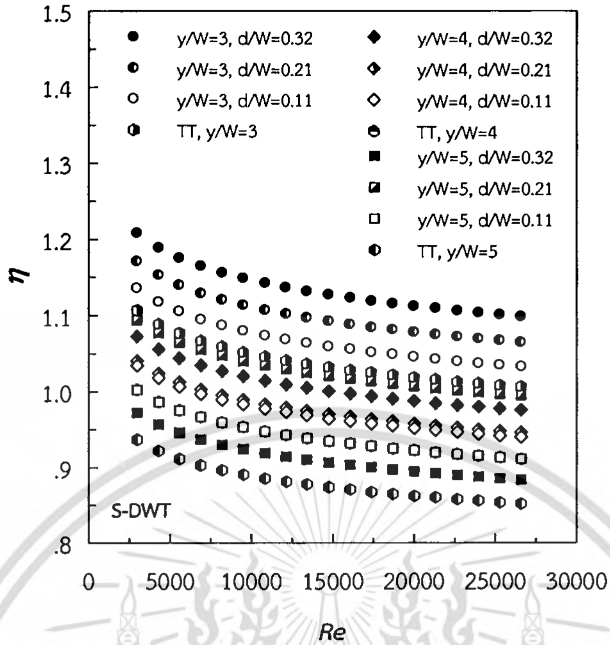
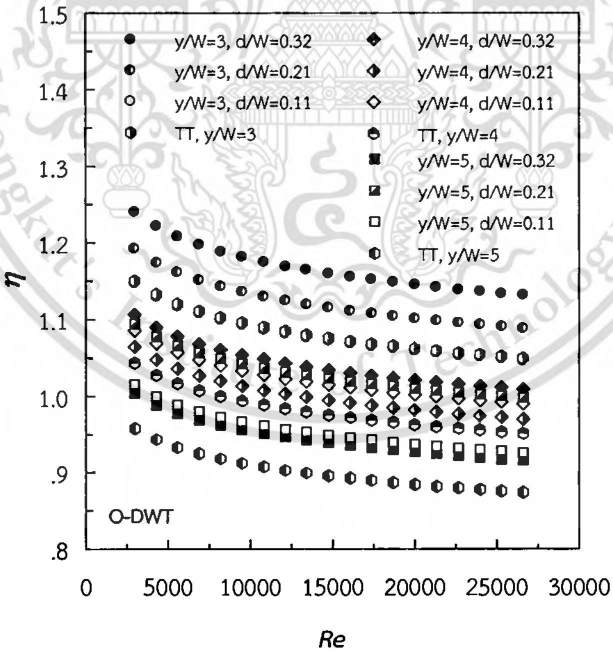


Figure 4.23: Effect of straight and oblique wings on friction factor.



(a) *S- DWT*



(b) *O-DWT*

Figure 4.24: Effect of delta-winglet twisted tape on thermal performance factor.

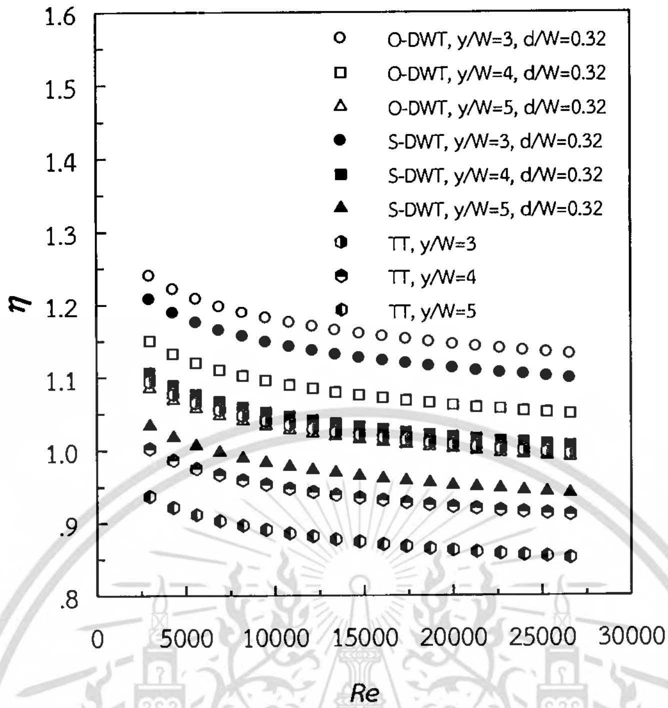


Figure 4.25: Effect of straight and oblique wings on thermal performance factor.

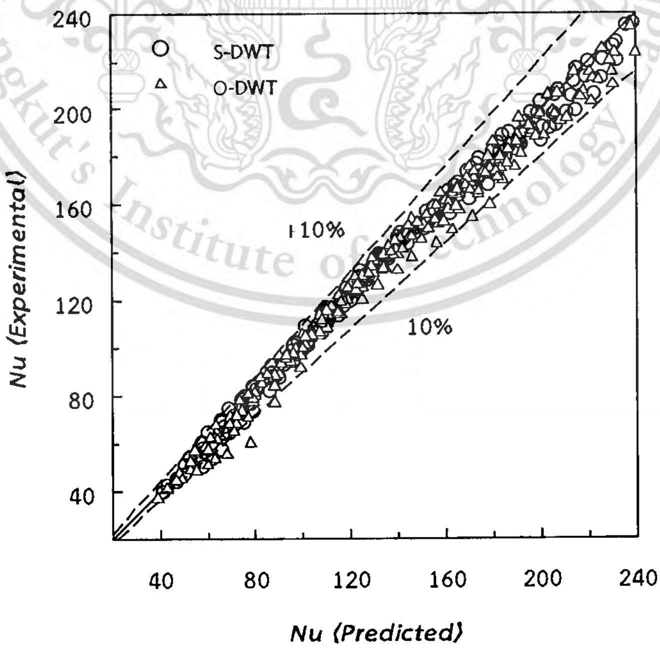


Figure 4.26: Validation of empirical correlations for Nusselt number.

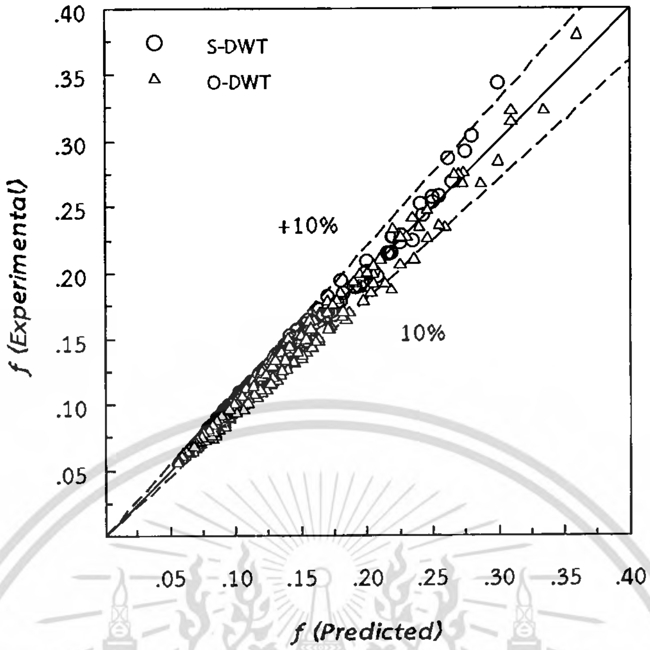


Figure 4.27: Validation of empirical correlations for friction factor.

4.5 Twisted tapes with rectangular wing

The results of heat transfer, friction factor and thermal performance behaviors in a heat exchanger tube fitted with the twisted-tapes with rectangular-wing (*T-Rec*) are described and discussed along with those of in the tubes with *TT* and *TA* as well as the plain tube.

Figure 4.28 displays the variation of Nusselt number with Reynolds number of the tubes with opposite rectangular wing twisted-tape *(OW-T)*, parallel rectangular wing twisted-tape *(PW-T)* and typical twisted tape (*TT*) and also with the plain tube. The figure demonstrates that both the modified twisted tapes (*OW-T* and *PW-T*) possess considerably higher Nusselt number than the *TT*, owing to the additional flow disturbance acted by the wings and alternate axes on the tapes. At a similar operating condition, the Nusselt number in the tube with *OW-T* is slightly higher than that in the one with *PW-T*. This can be attributed to a better turbulence distribution on both sides of twisted tape induced by the opposite wings. For the range examined, the increases of mean Nusselt numbers in the tubes equipped with *OW-T*, *PW-T* and *TT*, respectively, are found to be 110%, 100% and 27.5% over the plain tube with no insert.

Figure 4.29 portrays the variation of friction factor with Reynolds number values for all tubes. Similar to Nusselt number, the friction factor of the *OW-T* is slightly higher than that of the *PW-T*, and friction factors of the *OW-T* and *PW-T* are considerably higher than that of the *TT*. This implies that both the *OW-T* and *PW-T* generate higher force against the tube wall than the *TT*, due to the extra forces exerted by wings and alternate axes. The friction factor obtained by the *OW-T* and the *PW-T* is higher than that by the *TT*, by the factors of 1.87 and 1.76 times, respectively.

The experimental heat transfer result in term of Nusselt number obtained in the present investigation is illustrated in Figure 4.30. In general, Nusselt number increases with increasing Reynolds number due to the rise of fluid turbulence intensity. Evidently, all tubes fitted with *T-Recs* give higher Nusselt numbers than the ones with *TT, TA* and the plain tube. The possible actions for heat transfer enhancement by the *T-Rec* can be summarized as follows: (1) a swirl flow due to their helical geometry (2) an extra fluctuation of flow at each alternate-point (3) an extra force caused by wings located near the tube wall. All actions cause the disruption of thermal/hydrodynamic boundary layer along the tube wall and thus heat transfer auxiliary. Therefore, the combined effects associated with *T-Rec* lead to superior heat transfer over the effect of swirl flow alone given by *TT* or swirl flow coupled with extra flow fluctuation provided by *TA*. It also can be observed that the heat transfer enhancing effect by alternate-axis is more significant than that by wing. In addition, Nusselt number increases as wing-depth ratio (*d/W*) increases. Mean Nusselt numbers of the tubes equipped with *T-Rec* at wing-depth ratios of 0.1, 0.2 and 0.3 are enhanced respectively by around 39%, 49% and 57% over those in the tube with *TT* and 78%, 91% and 100% over those in the plain tube.

Friction factors are shown in Figure 4.31 as a function of Reynolds number. For the tubes fitted with twisted tape inserts, friction factor noticeably decreases with increasing Reynolds number while that in the plain tube insignificantly changes over the studied Reynolds number range. Friction factors of the tubes with *T-Recs* are consistently higher than those in the tubes with *TT, TA* and also the plain tube, under similar conditions. The higher friction factors in the tubes with *T-Recs* are the results combined effects associated by both wing and alternate-axis actions leading to the high dissipation of dynamic pressure of the fluid over those in other tubes. The effect of wing-depth ratio on friction factor is similar to that on Nusselt number as friction factor increases with increasing wing-depth ratio. Mean friction factors of the tubes equipped with *T-Recs* at wing-depth ratios of 0.1, 0.2 and 0.3 are increased respectively by around 46%, 66% and 76% over those in the tube with *TT* and 330%, 389% and 418% over those in the plain tube. It is noteworthy that the percentage increase in friction factor of a tube with *T-Rec* is significantly larger than that in Nusselt number at the identical volumetric flow rate.

Figure 4.32 shows the variation of thermal performance factor with Reynolds number. All performance factors are evaluated at equivalent pumping power constraint. For the tubes equipped with *T-Recs* and *TA*, thermal performance factor considerably decreases as the Reynolds number increases while that of the tube with *TT* slightly decreases for the present Reynolds number range. This signifies that the use of tape insert becomes less feasible in terms of energy saving at high Reynolds numbers. The thermal performance factors associated with *T-Recs* are consistently higher than those given by *TA* and *TT*, at the given Reynolds number. Therefore, it can be concluded that the use of *T-Rec* is more promising than the use of *TA* or *TT*, for energy saving purpose. Over the range examined, the *T-Recs* provide highest thermal performance factor than the *TT* up to 42%. In addition, thermal performance factor increases as wing-depth ratio increases. For the present range, the maximum thermal performance factor is achieved with the use of *T-Rec* at wing-depth ratio of 0.3 and Reynolds number of 5500.

The correlations developed for the present experimental results of the tube equipped with twisted tapes in a range of Reynolds number between 5500 and 20,200, for Nusselt number, friction factor and thermal performance factor are shown in equations (4.14) to (4.16), respectively. The validity of these correlations is verified by comparing the predictions with the experimental data as shown in Figure 4.33(a-c). Evidently, the predicted values are in good agreement with experimental data within ±6% for Nusselt number, ±5% for friction factor and ±6% for thermal performance factor.

Twisted tapes with rectangular wing (T-Rec):

$$Nu = 0.48Re^{0.562}Pr^{0.4}(d/W)^{0.104} \tag{4.14}$$

$$f = 40.59Re^{-0.56}(d/W)^{0.17} \tag{4.15}$$

$$\eta = 5.89Re^{-0.17}(d/W)^{0.048} \tag{4.16}$$

A comparison of flow behaviors in the plain tube, the tube with *TT* and *T-Rec* is presented in Figure 4.34(a-c). It should be mentioned here that the flow visualization by dye injection technique is limited for low Reynolds number range. In the present work, all flows are visualized at the identical Reynolds number of 2000, for the qualitative comparison. As expected, in the plain tube, fluid shows a common longitudinal flow. On the other hand, in the tube with *TT*, a centrifugal force is superimposed over the longitudinal flow due to the helically rotating, this increases the flow length and thus heat transfer area. The flows in the tube equipped with T-Rec possess a similar behavior. The dyes which are not directly injected to the wings, show common swirl flow and do not disperse for the whole length tested. On the other hand, the blue and green dyes which injected toward the wings is separated into two streams and dispersed behind the wing position. This can be considered as the effects of fluid disturbance by the wings, indicating the superior fluid mixing over that of associated by the *TT*.

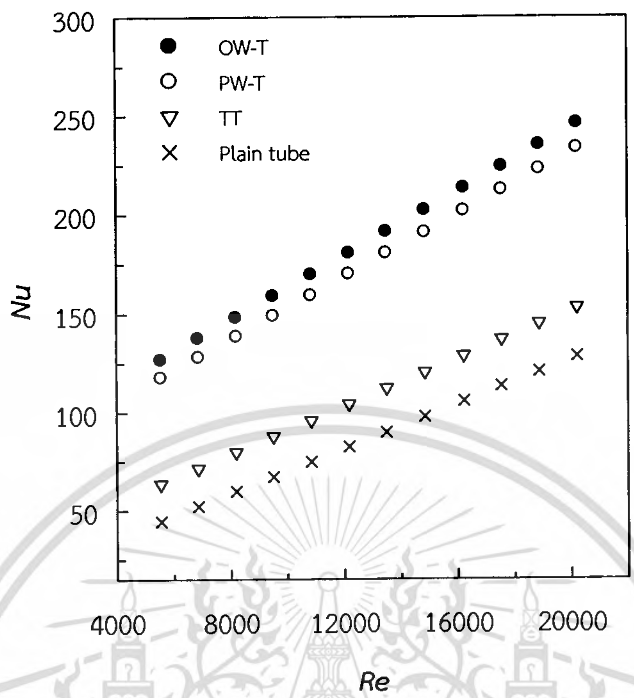


Figure 4.28: Effect of twisted tapes with *OW-T/PW-T* on Nusselt number.

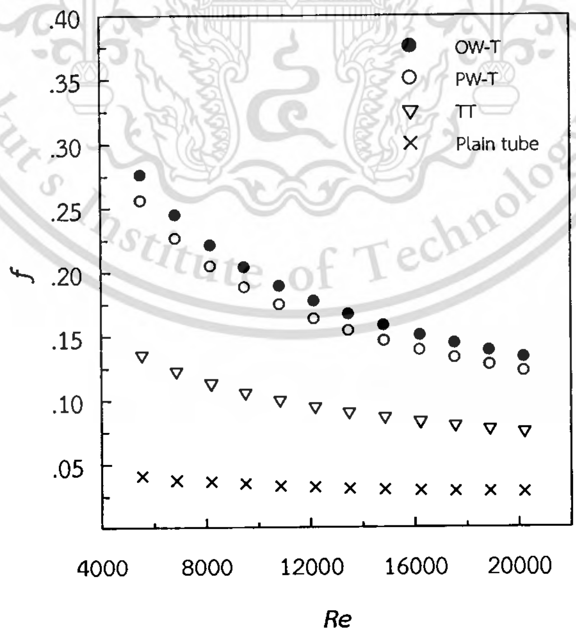


Figure 4.29: Effect of twisted tapes with *OW-T/PW-T* on friction factor.

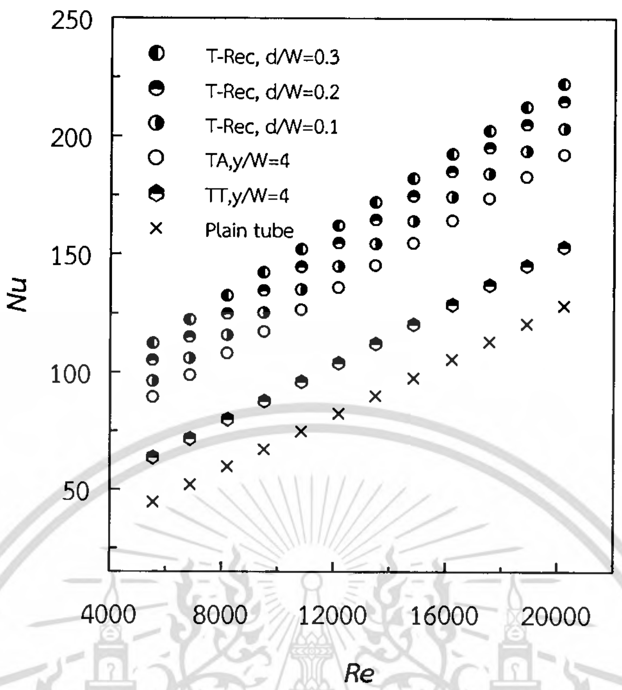


Figure 4.30: Effect of twisted tapes with rectangular wings on Nusselt number.

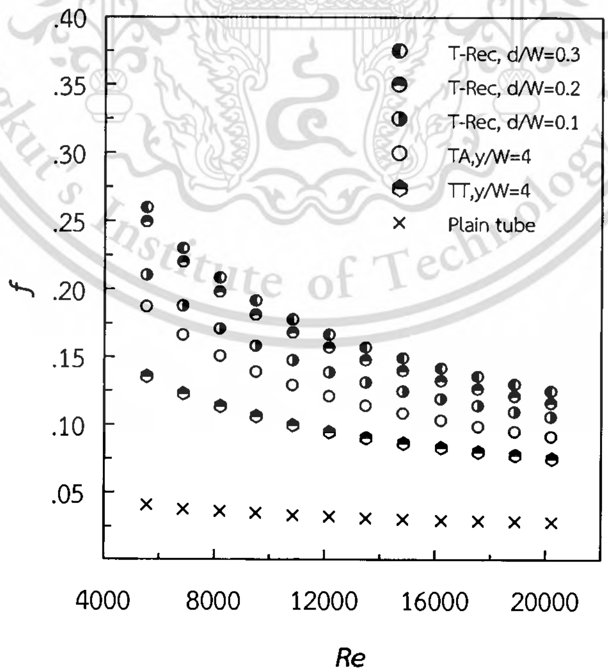


Figure 4.31: Effect of twisted tapes with rectangular wings on friction factor.

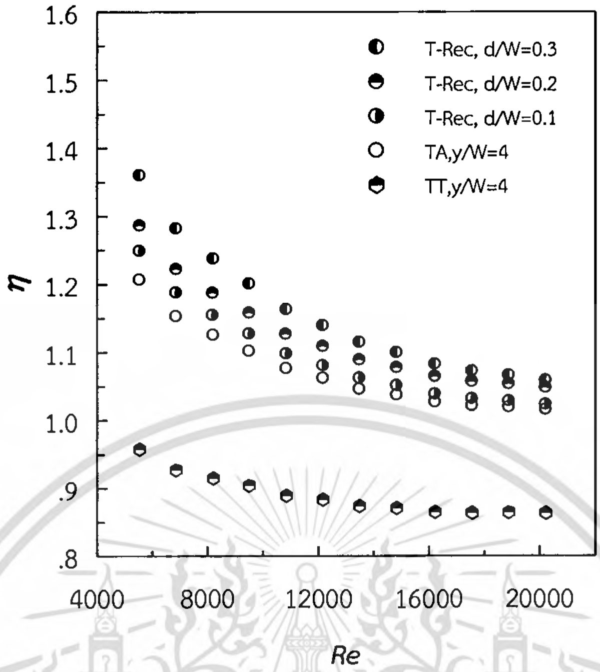
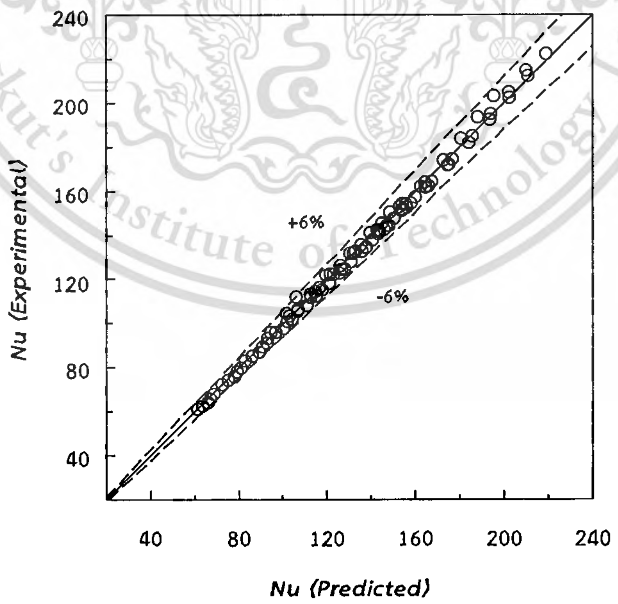
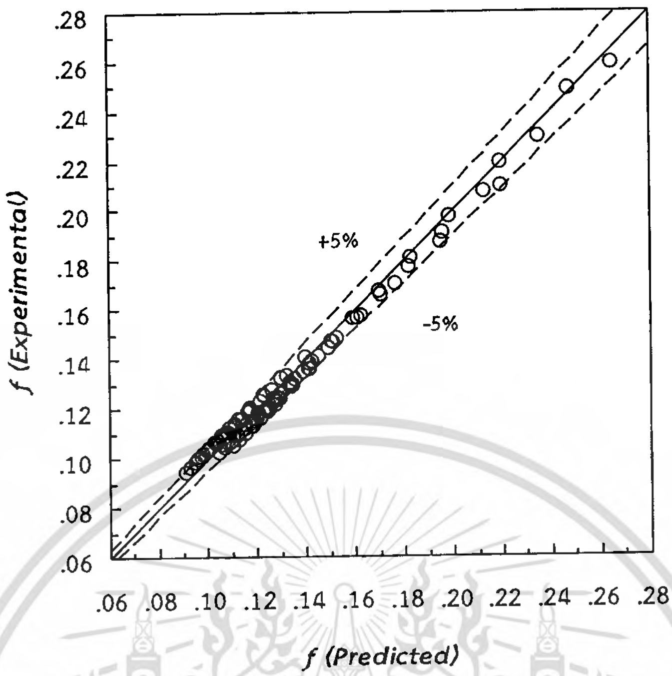


Figure 4.32: Effect of twisted tapes with rectangular wings on thermal performance factor.

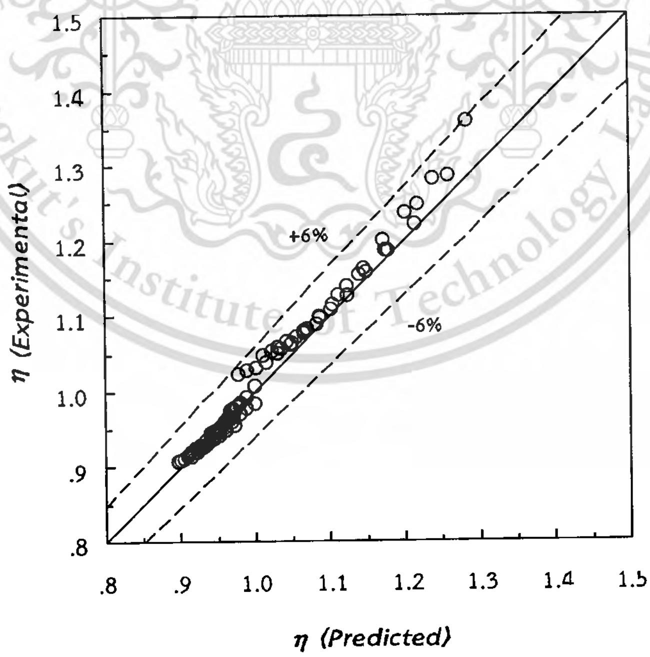


(a) Nusselt number

Figure 4.33: Validation of empirical correlations for (a) Nusselt number, (b) friction factor and (c) thermal performance factor.

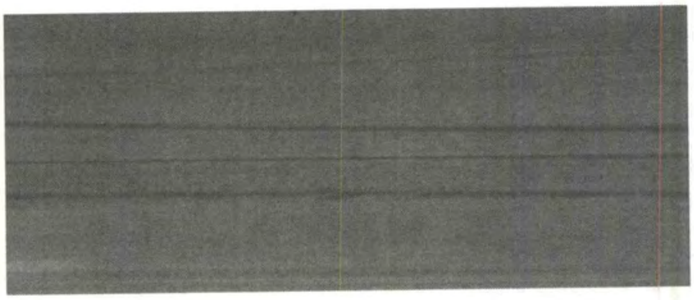


(b) friction factor



(c) thermal performance factor

Figure 4.34: Validation of empirical correlations for (a) Nusselt number, (b) friction factor and (c) thermal performance factor. (continued)



(a) plain tube

(b) plain tube with *TT*

(c) plain tube with *T-Rec*

Figure 4.35: Photograph of flow behaviors by dye technique.

4.6 Perforated twisted tape with parallel wing

The experimental investigations of heat transfer, friction factor and thermal performance behaviors in a heat exchanger tube fitted with the perforated twisted tape with parallel wing (*PTT*) with parallel wings of varying wing-cut ratio (*w/W*) and hole diameter ratio (*d/W*) are described.

Since it is difficult to distinct the main flow from the secondary flows induces by *TT* and *PTT* via the dye visualization technique in turbulent regime, thus the visualization was performed in laminar regime instead, only for qualitative comparison. Figure 4.38 shows the flow visualization of the flows through tubes with and without twisted tape. Only axial flow was observed in the plain tube (Figure 4.38a) while common swirl flow was found with the presence of *TT* (Figure 4.38b). In case of *PTT* (Figure 4.38c), apart from a common swirl flow, there was attack of dye streams on the wings of *PTT* leading to an extra turbulence as well as collision among dye streams and thus superior fluid mixing to the case of *TT*. It should be mentioned that the dye stream positioned around the mid of the tape was directed through a hole to another side of the tape, and the stream direction was between those of an axial flow and a swirl flow.

The experimental results of the tubes equipped with *PTTs* and *TT* as well as the plain tube obtained under turbulent flow conditions presented in Figure 4.36(a-b). Influence of the presence of wing on heat transfer was significant, notified by the considerably higher Nusselt number given by *PTT* compared to those provided by *TT*. The superior heat transfer is responsible by the induction of extra turbulent flows near the tube wall as detected by the dye visualization (Figure 4.38b). This efficiently disturbs a thermal boundary layer by generating periodic disruption of the viscous boundary layer. At similar operating conditions, it was found that Nusselt number increased with increasing wing depth ratio *(w/W)*. It can be explain by the fact that the larger depth ratio causes higher turbulence intensity and thus better mixing fluid near the tube wall. For the tape with the largest depth ratio (*w/W* = 0.33), the increase in heat transfer rate was up to 49% and 23% over those of the ones with *w/W* = 0.11 and *w/W* = 0.22, respectively. Figure 4.37(a-b) shows the variation of the friction factor with Reynolds number. Obviously, the use of the tape with larger wing depth ratio generated higher friction factor. This is directly related to the higher turbulence intensity as mentioned above. As found the tape with the largest depth ratio (*w/W* = 0.33) yielded 40.8% and 18.3% higher mean friction factor than the ones with *w/W* = 0.11 and *w/W* = 0.22, respectively. The result of thermal performance factor in which both heat transfer and friction are taken into account based in the same pumping power, is illustrated in Figure 4.39. It is found that *PTTs* consistently gave higher thermal performance factor than *TT*, at similar conditions. Among *PTTs*, thermal performance factor increased with increasing wing depth ratio. Thermal performances varied between 0.71 and 1.01, 0.77 and 1.16, and 0.91 and 1.32 for the *PTTs* with w/W = 0.11, 0.22 and 0.33, respectively. It noteworthy that thermal performance was higher at lower Reynolds number, this implies that twisted tapes are more suitable for practical application at lower Reynolds number.

The effect of perforated diameter ratio (d/W = 0.11, 0.33 and 0.55) on the heat transfer rate is shown in Figure 4.36(a-b). Apparently, Nusselt number increased with the decrease of diameter ratio. This supports the finding by the dye visualization (Figure 4.38c) that the stream directed through a hole behaved between an axial flow and a swirl flow, with the larger hole diameter, the flow is assumed to be more similar to the axial flow leading to the loss of swirl intensity imparted to the flow between tape and surface wall. Heat transfer rate decreased up to 2.8% and 6% by the uses of the tapes with d/W = 0.33 for d/W = 0.55 compared to that of the tape with d/W = 0.11. By the same reason, friction generated by PTTs decreased with the increase diameter ratio (d/W) as presented in Figure 4.37(a-b). The tapes with d/W = 0.33 for d/W = 0.55 yielded 6% and 11.6% lower mean friction factor than the one with d/W = 0.11. Effect of the perforated diameter ratio (d/W) on the thermal performance factor in a heat exchanger tube fitted with *PTT* is presented in Figure 4.40. The thermal performances from using *PTT* with smaller hole diameter ratio (d/W) were observed to be higher than that those achieved from the tape with larger d/W. This signifies the dominant effect of increased heat transfer over that of increased friction factor as hole diameter decreases. It was found that the tape with the smallest diameter ratio (d/W = 0.11) provided higher thermal performance factor than the ones with d/W = 0.33 and 0.55 by around 4.9% and 10.4%, respectively. Note that for the present range, thermal performances achieved by using all tape inserts are above unity (low *Re*), indicating the economic benefit due to the heat transfer enhancement. The experimental results of Nusselt number, friction factor and thermal performance were fitted, using least square regression analysis, in which a wing-cut ratio (w/W) and a hole diameter ratio (d/W) were taken into account. As shown from these figures the maximum deviations between the experimental data and correlations are ±8%, ±8%, and ±6%, respectively (Figure 4.41a-c).

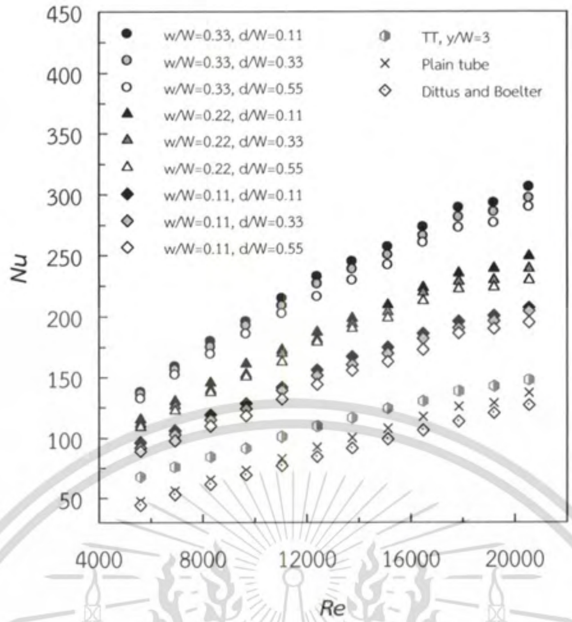
For perforated twisted tape with parallel wing (*PTT*):

$$Nu= 0.489Re^{0.615}Pr^{0.4}(w/W)^{0.355}(d/W)^{-0.036} \tag{4.17}$$

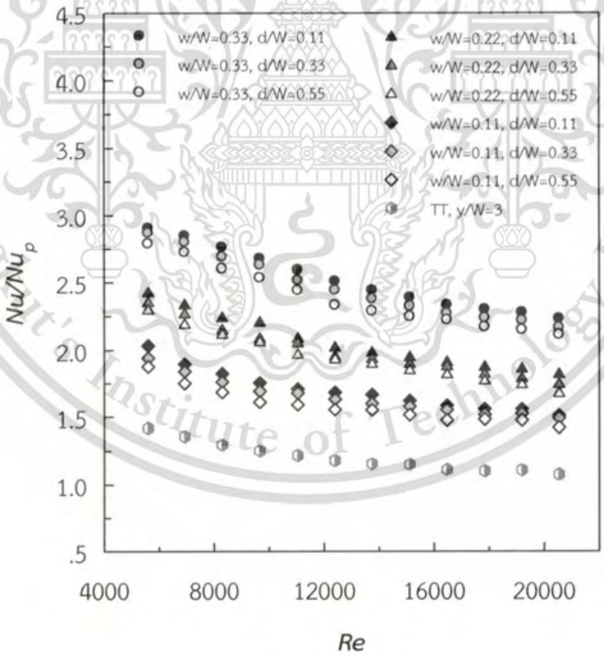
$$f = 11.376Re^{-0315}(w/W)^{0.304}(d/W)^{0.066} \tag{4.18}$$

$$\eta = 9.818Re^{-0.216}(w/W)^{0.253}(d/W)^{-0.058} \tag{4.19}$$

Empirical correlations for Nusselt number, friction factor, thermal performance factor are developed for the tube with perforated twisted tape with parallel wing inserts in the range of Reynolds number between 7000 and 22,000 as above.

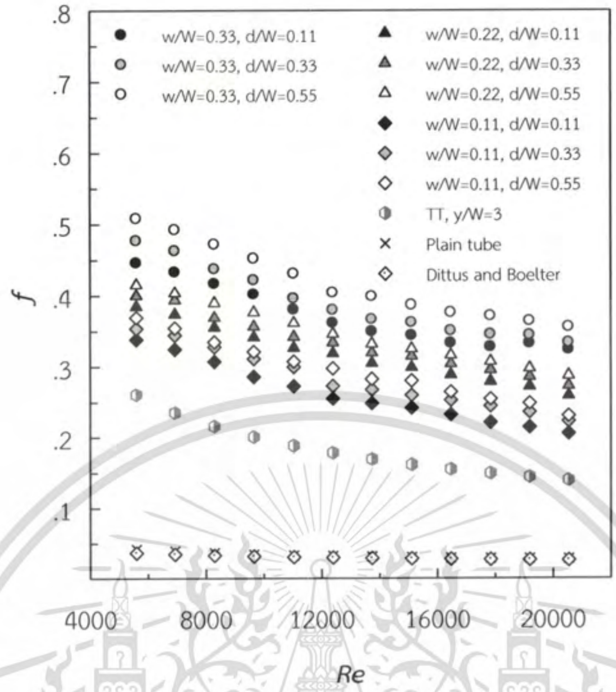


(a) Nusselt number

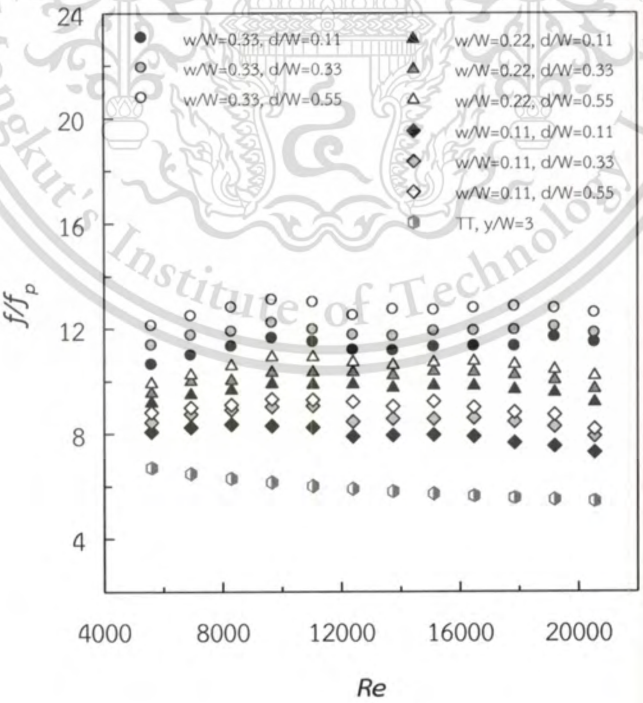


(b) Nu/Nu_p

Figure 4.36: Effect of perforated twisted tape with parallel wing on Nusselt number.

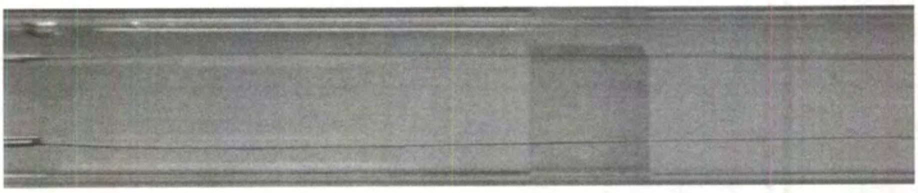


(a) friction factor

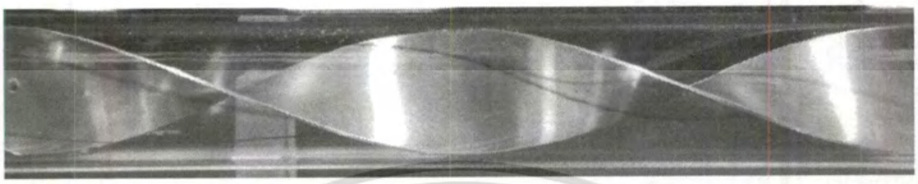


(b) f/f_p

Figure 4.37: Effect of perforated twisted tape with parallel wing on friction factor.



(a) plain tube



(b) *TT*

(c) *PTT*

Figure 4.38: Flow visualization of flows through tube fitted with *PTT*.

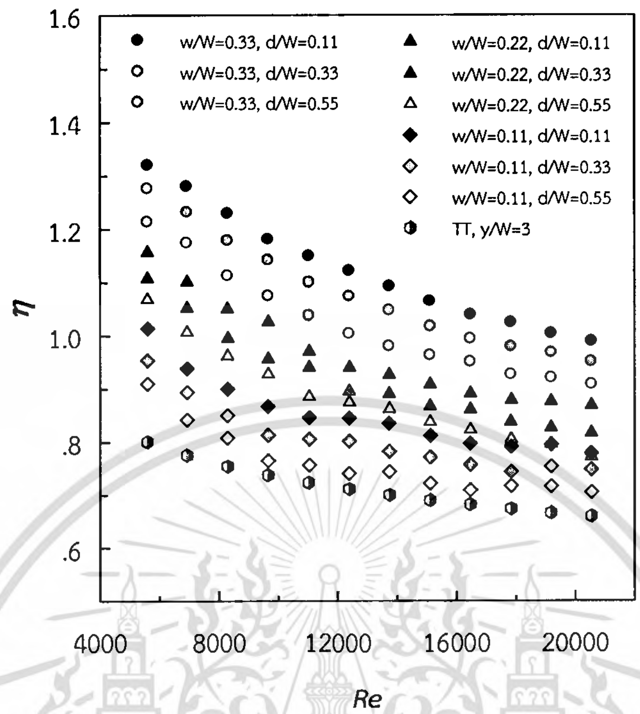
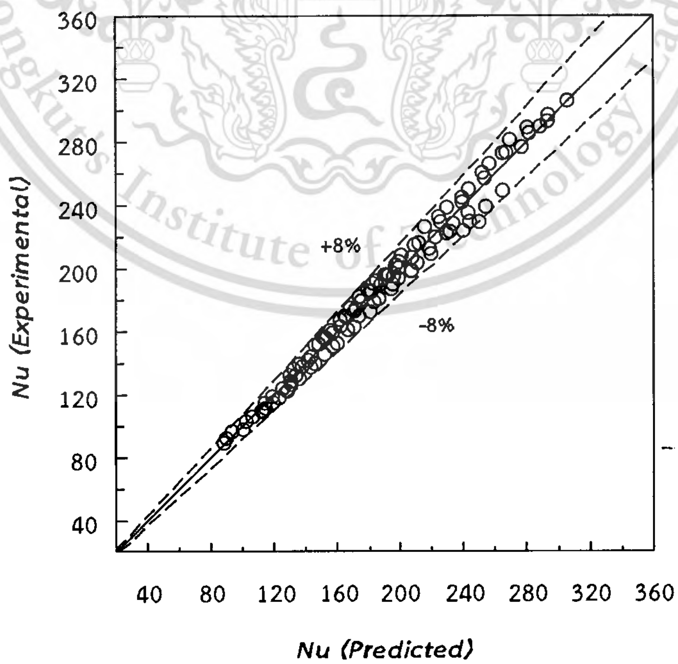
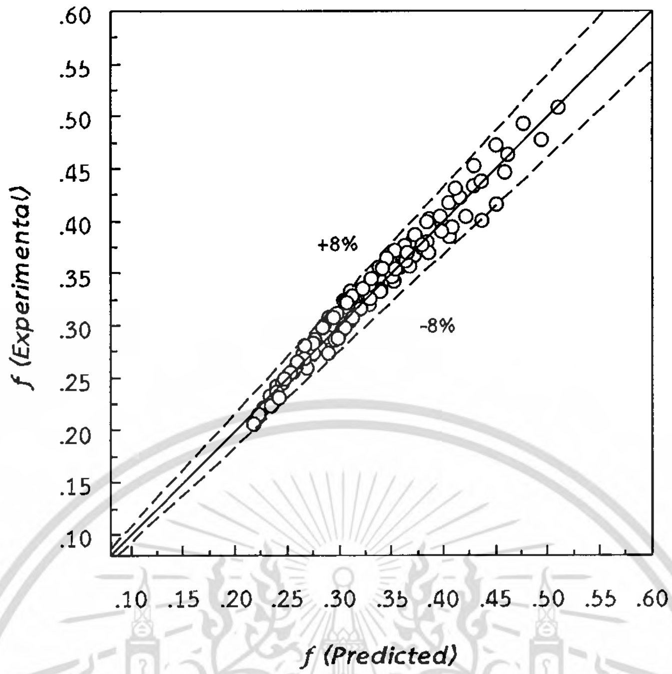


Figure 4.39: Effect of perforated twisted tape with parallel wing on thermal performance factor.

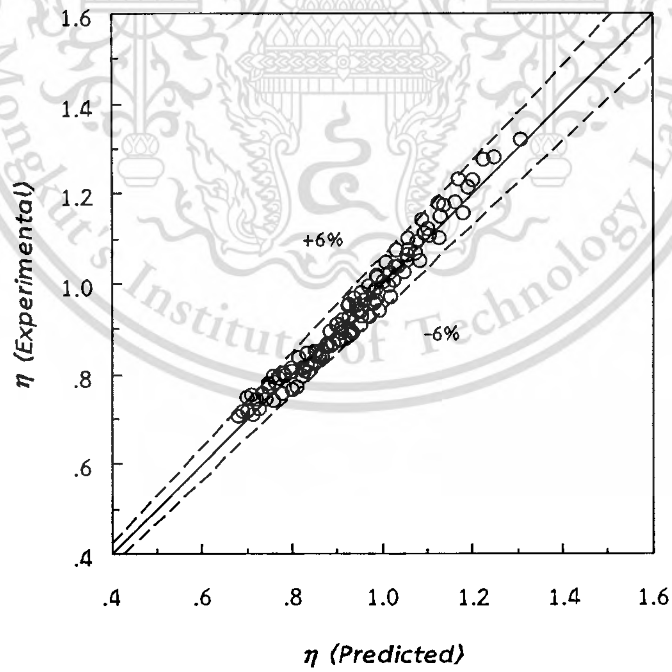


(a) Nusselt number

Figure 4.40: Validation of empirical correlations for (a) Nusselt number



(b) friction factor



(c) thermal performance factor

Figure 4.41: Validation of empirical correlations for (a) Nusselt number, (b) friction factor and (c) thermal performance factor.

4.7 Twin twisted tapes

In this section, the heat transfer and friction factor characteristics and also thermal performance factor in a tube fitted with twin counter/co twisted tapes (*counter/co-swirl tape*) are presented. The experiments are performed using twin twisted tapes with four different twist ratios, *y/W* = 2.5, 3.0, 3.5 and 4.0, in the range of Reynolds number between 3700 and 21,000. The results obtained for the tube fitted with the single twisted tapes (*ST*) or typical twisted tape (*TT*) with three different twist ratios, *y/W* = 3.0, 3.5 and 4.0, and the plain tube are used as the reference data for the performance evaluation of the modified twisted tapes.

Effects of the twin counter twisted tapes (*CTs: counter-swirl flow generators*) and the twin co twisted tapes (*CoTs: co-swirl flow generators*) on the heat transfer rate in a uniform heat flux tube are shown in Figure 4.42. The results obtained for the tube fitted with the single twisted tape (*ST*), and the plain tube are also plotted for comparison. General trend found in Figure 4.42 is that, the Nusselt number increases with the rise of Reynolds number. The influence for applying *CTs* on heat transfer rate is significant for all Reynolds numbers.

In the present study, the phenomena of the swirling flow in the tube fitted with *ST*, *CoTs* and *CTs* is demonstrated in Figure 4.43(a-c) using finite volume method. The results are based on the computational conditions, for the flow at the constant Reynolds number, *Re* = 10,000. Apparently, twin swirl flows are produced for both *CoTs* and *CTs* while only single swirl flow is generated by *ST*. The higher number of streams generated by both twin tapes is responsible for a better mixing and thus higher heat transfer rate as experimental results shown in Figure 4.42. For all twisted tapes, each stream is induced along each side of tape with axisymmetric profile in direction led by the tape geometry. With the close look for twin co twisted tapes (*CoTs: co-swirl flow generators*), recirculation zones appear on the top/bottom of vortex core and the generated swirl flows show just small contact zone in the clearance between two twisted tapes, indicating little interaction between them. Therefore, the mixing in the tube equipped with *CoTs* is dominantly enhanced by rather independent swirl flows. On the other hand, the swirls generated by twin counter twisted tapes (*CTs: counter-swirl flow generators*) are found to be converged in the clearance between two twisted tapes, this apparently provides higher intensity of vortex strength than that generated by the *CoTs*. In addition, the recirculation zone is not induced in the tube with *CTs* as the strong swirl flows impinge to the tube wall. Therefore, it can be addressed herein that the higher interaction between swirl flows (or high vortex strength) induced by the *CTs* leads to superior fluid mixing, resulting in uniform fluid temperature in tubes, and thus more efficient heat transfer in comparison with the *CoTs*. This confirmed by the obtained results in which, the heat transfer rates in the tube fitted with *CTs* (counter-swirl flow) are noticeably higher than those in the tube fitted with *CoTs* (co-swirl flow) for all range Reynolds numbers studied (Figure 4.42). From the figure, it is also found that the influence of *CTs* on the heat transfer rate is more intense at higher Reynolds number. Over the range considered, heat transfer rate in the tubes fitted with *CTs* are respectively 12.5 to 44.5% and 17.8 to 50% higher than those in tubes equipped with *CoTs* (*co-swirl flows*) and *ST* (*single swirl flow*).

Figure 4.44 presents the friction factor in the corresponding tubes as shown in Figure 4.42. Basically, the friction factor decreases with increasing Reynolds number. At the same Reynolds number, the friction factors in the tube fitted with *CTs* are higher than those in tube fitted with *CoTs*, tube fitted with single twisted tape (*ST*) and the plain tube (*non-swirl flow*). This is a consequence of the repeated acts of strong intensity of vortex in the tube with *CTs*. Over the range studied, the friction factors for tube fitted with *CTs* are higher than those in tube fitted with *CoTs* around 26.5 to 45.5%.

Effect of twist ratios (y/W = 2.5, 3.0, 3.5 and 4.0) on the heat transfer rate in the tube fitted with *CTs* and *CoTs* is demonstrated in Figure 4.42. From the experimental results, it can be observed that heat transfer enhancement increase as twist ratio decreases. In common, the smaller twist ratio generates stronger swirl intensity, leading to more efficient interruption of boundary layer along the flow path. Hence, heat can be transferred efficiently over thin boundary layer. Moreover, the residence time of the flow increases with the increasing swirl flow intensity. This extend the duration of heat transfer between the working fluid and heat source (tube wall). For the entire conditions studied, the *CTs* and *CoTs* with the smallest twist ratio (y/W = 2.5) provide the heat transfer rates around 7.8 to 74.6% and 59.4 to 187 % higher than those for the tube with larger twist ratios and the plain tube, respectively.

As seen in Figure 4.44, friction factor tends to increase with decreasing twist ratio. This is in the same trend found for Nusselts number. This can be explained by the fact that the use of a twisted tape with a smaller twist ratio leads to a higher viscous loss near the tube wall regions caused by a stronger swirl flow. Over the range studied, the friction factors for the tube fitted with *CTs* with y/W = 2.5, 3.0, 3.5, and 4.0, are respectively 6.37, 5.33, 4.57 and 3.95 times of those in the plain tube.

In the present investigation, the tests are made in a uniform heat flux tube with water as working fluid and correlations are applicable to turbulent region of Reynolds number (*Re*) between 3700 and 21,000. The empirical correlations from the experimental results of the plain tube fitted with *ST*, *CoTs* and *CTs* can be writing in term of twist ratio (*y/W*), Reynolds number (*Re*) and Prandtl number (*Pr*) as below:

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

$$Nu = 0.473Re^{0.66}Pr^{0.4}(y/W)^{-0.9} \quad (4.20)$$

$$f = 72.29Re^{-0.53}(y/W)^{-1.01} \quad (4.21)$$

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

$$Nu = 0.264Re^{0.66}Pr^{0.4}(y/W)^{-0.61} \quad (4.22)$$

$$f = 41.7^{-0.52}(y/W)^{-0.84} \quad (4.23)$$

Comparison between the Nusselt number and the friction factor obtained from the present data with those calculated by the present correlations are portrayed in Figures 4.45 and 4.46. Evidently, the majority of the heat transfer data falls within ±10% for the present correlations of equations (4.20) and (4.22). Equations (4.21) and (4.23) provide the correlative results of the friction factor with maximum discrepancies of ±6% with the experimental measurements.

For the tube fitted with *CTs:*

$$\eta = 2.8Re^{-0.016}(y/W)^{-0.624} \quad (4.24)$$

For the tube fitted with *CoTs:*

$$\eta = 1.82Re^{-0.0186}(y/W)^{-0.38} \quad (4.25)$$

Figure 4.47 shows the effect of various twin twisted tapes at different twist ratios (*y/W*) on the thermal performance factor in Reynolds number range of 3700 to 21,000. The comparative data reveal that the thermal performance factor increases with slightly decreasing Reynolds number during to low pressure drop. In additions, thermal enhancement indices are varied between 1.01 and 1.39 for *CTs*, 0.89 and 1.1 for *CoTs*, and 0.81 and 0.9 for *ST*, depending upon the Reynolds number and the twist ratio (*y/W*). For *CTs*, the mean performance factor for the smallest twist ratio (*y/W* = 2.5) is respectively 12%, 23.4% and 34% higher than those for *y/W* = 3.0, 3.5, and 4.0. The above data suggest that the highest performance factor can be obtained at the low twist ratio (*y/W*), and relatively low Reynolds number. It is noteworthy that the enhancement indices in the tube fitted with *CTs* (*counter-swirl tapes*) are around 12 to 25.7% higher than those in the tube fitted with *CoTs* (*co-swirl tapes*).

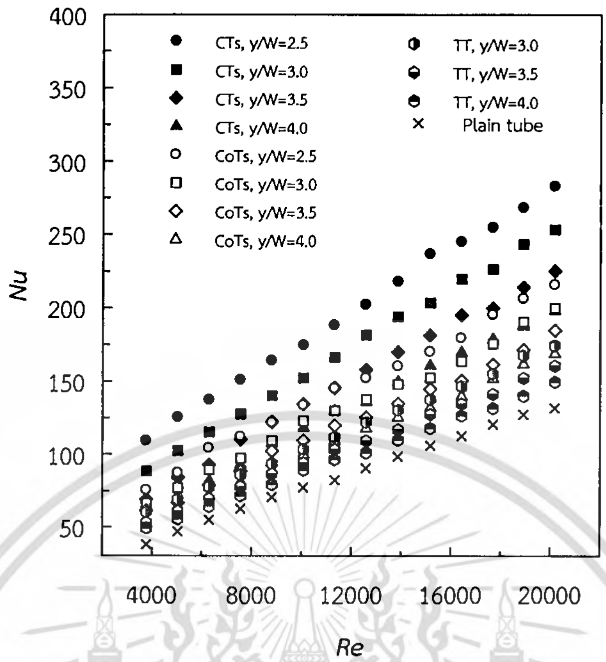
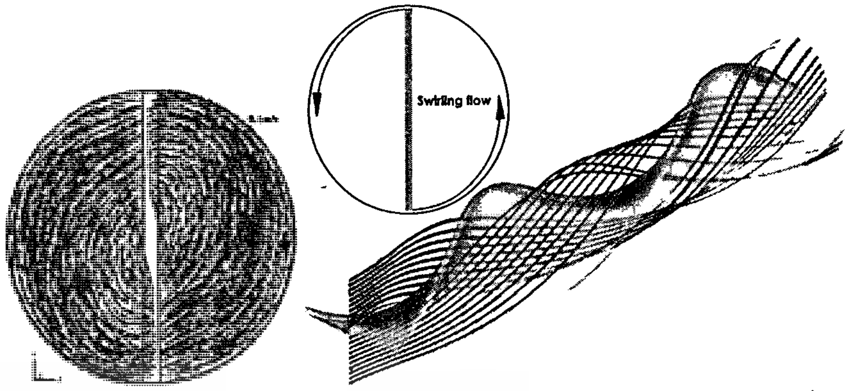
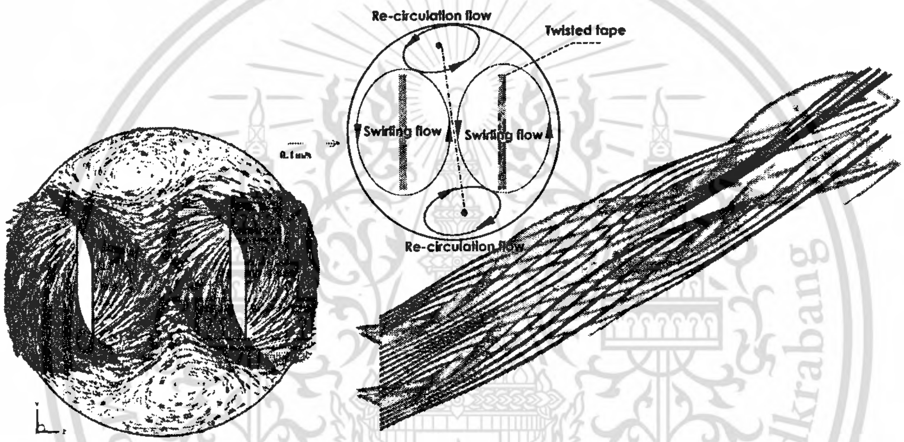


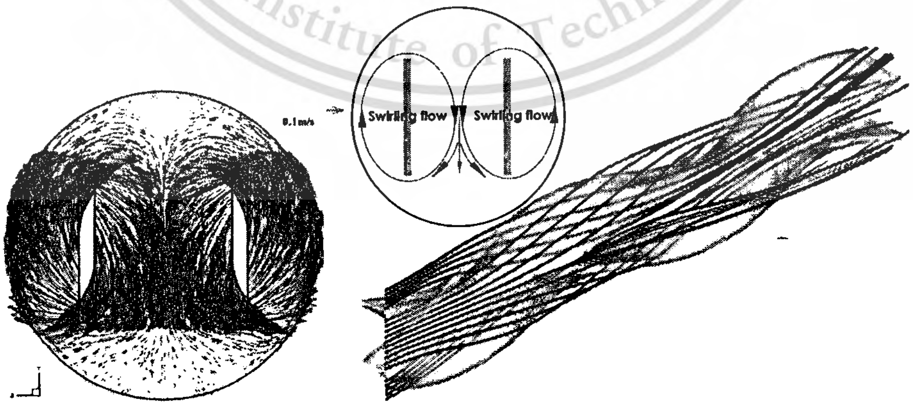
Figure 4.42: Effect of twin twisted tapes on Nusselt number.



(a) single twisted tape (*ST*)



(b) twin co twisted tapes (*CoTs*)



(c) twin counter twisted tapes (*CTs*)

Figure 4.43: Path line of flow through a tube fitted with twisted tape.

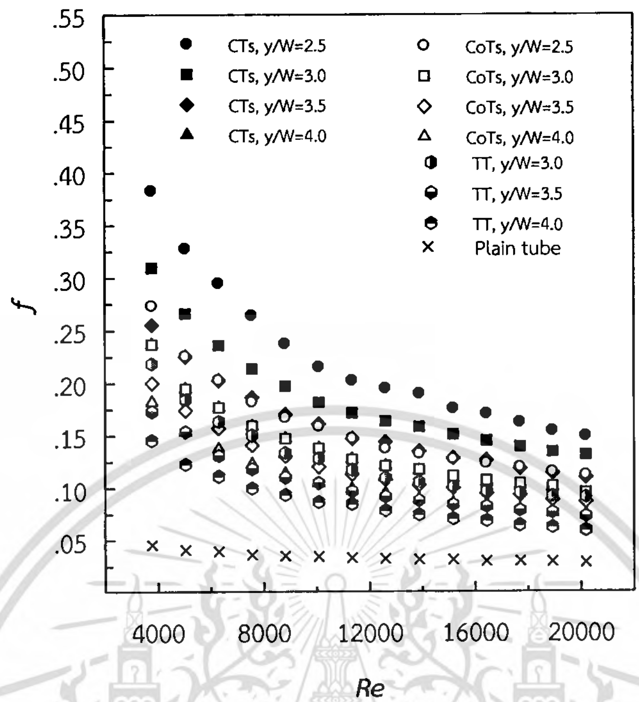


Figure 4.44: Effect of twin twisted tapes on friction factor.

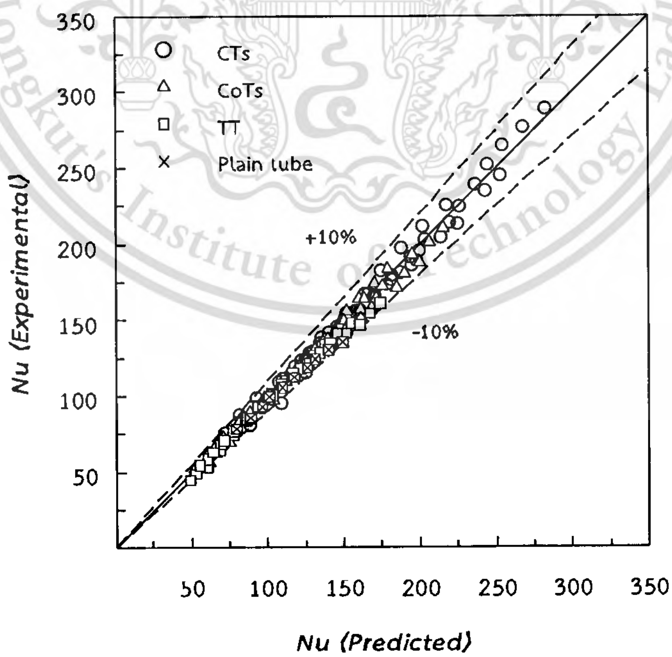


Figure 4.45: Validation of empirical correlations for Nusselt number.

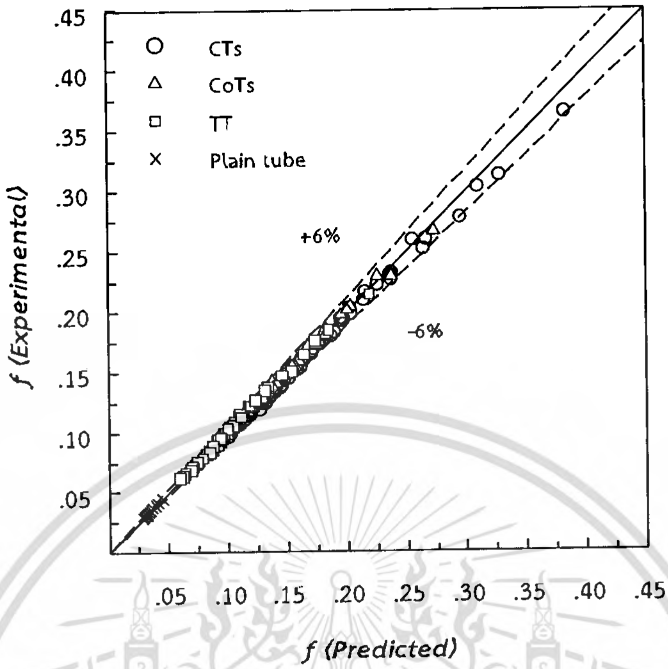


Figure 4.46: Validation of empirical correlations for friction factor.

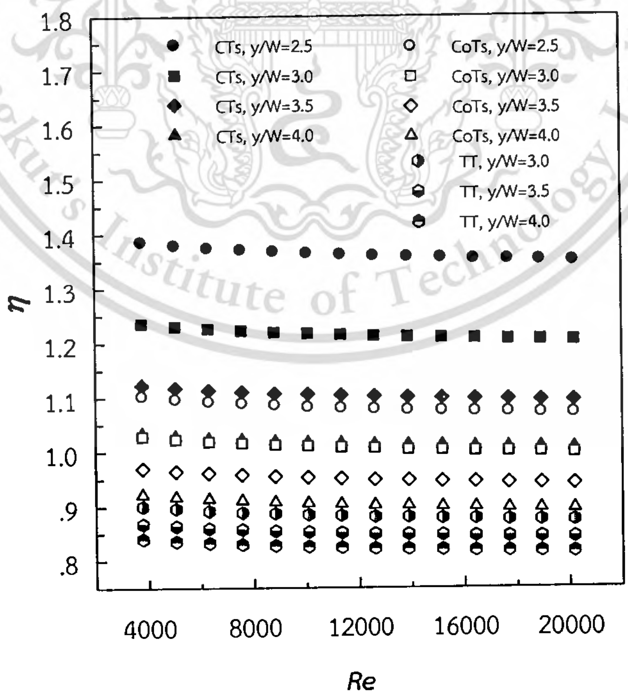


Figure 4.47: Effect of twin twisted tapes on thermal performance factor.

4.8 Wing twisted tape

The experimental results including Nusselt number, friction factor and the thermal performance factor behaviors in the tube with wing twisted tapes (*WT*) are presented and discussed in the following subsections.

The comparison of Nusselt numbers of the tube with typical twisted tape (*TT*) and twisted tapes with centre wings (*WT*) are demonstrated along with those of the plain tube in Figure 4.48. At the similar operating conditions, Nusselt numbers in the tube with *WT* were found to be consistently higher than those in the tube with *TT* and plain tube. Depending on Reynolds number and angle of attack the Nusselt numbers in the tube with *WT,* were 16% to 52% and 58% to 138% higher than those in the tube with *TT* and the plain tube, respectively. The better performance of the WT compared to that of the *TT,* can be attributed the combined effect of common swirl flow induce by the twisted tape and the vortex generated by the centre wings in the *WT*. This can be supported by the fact that the vortex provides the additional disturbance into the typical swirl stream generated by the twisted tape. This action directly improves heat transfer rate with respect to that offered by the swirl flow alone.

The experimental results for friction factor in the tube fitted with the *TT* and the WT as well as a plain tube under an isothermal flow condition are shown in Figure 4.49. Apparently, the friction factors in the tube fitted with the twisted tape consisting of centre wings (*WT*) were found to be higher than those in the tube with the typical twisted tape around 21.5% to 101% and higher than those in plain tube from 4.4 to 6.9 times. This is due to an extra blockage by the wings appears on the WT and the additional forces exerted by a vortex flow, giving rise to a more pronounced dissipation of the dynamical pressure of a working fluid.

The effect of angle of attack on heat transfer characteristics in the tube fitted with the *WT* is also reported in Figure 4.48. The result revealed that Nusselt number (Nu) increased with the increase of angle of attack. Over the range studied, the *WT* with the largest angle of attack of 74° provided the highest heat transfer rate. The maximum increase in Nusselt number obtained in the tube with this tape was 11%, 4.7% and 138% compared to those in the tube with the *WT* at angles of attack, angle of attack of 43° and 53° and the plain tube, respectively. This can be explained that the wing with the larger attack angle gives the more efficient mixing which provides more disruption to a thermal boundary, leading to a better heat transfer.

Figure 4.49 presents the effect of angle of attack on friction factor which was taken under an isothermal condition. As expected, under the similar conditions, friction factor increased with the increasing angle of attack. The mean friction factors in the tube with the *WT* at angle of attack of 43°, 53° and 74° were respectively 4.8, 5.6 and 6.4 times of those in the plain tube and 1.4, 1.6 and 1.8 times of those in the tube with the *TT*. This is simply attributed to a greater flow blockage by the centre wings at the larger angle of attack.

Determination from Figures 4.48, 4.49 and 4.50, it is found that under the similar conditions, the heat transfer coefficient, friction factor and thermal performance factor in the tube fitted with the *WT-A* were consistently higher than those in the tube equipped the *WT, TA* and plain tube. The heat transfer enhancement data is also presented in term of Nusselt number ratio which is plotted against the Reynolds number as shown in Figure 4.48(b). Obviously, that the Nusselt number ratio is high at low Reynolds number and then decreased with increasing Reynolds number and became nearly constant at high Reynolds number. Figure 4.49(b) shows the variation of the friction factor ratio with the Reynolds number value for various twisted tapes. The trend with Reynolds number is found to be similar to that for the Nusselt number ratio. Over the range investigated, *WT-A* with angle of attack of 74° gave maximum thermal performance factor of 1.4 at Reynolds number of 5200. Mean values of Nusselt number, friction factor, thermal performance factor provided by *WT-A* with angle of attack of 74° were respectively, 17.7%, 30.6% and 7.8% higher than those in the tube with *WT-A* with the same angle of attack, 20.8%, 53% and 4.9% higher than those in the tube with *TA*, and 62%, 123% and 24% higher than those in the tube with *TT*.

The visualization of flow by smoke wire technique through a tube with various twisted tape inserts, relating to their performance mentioned above, is presented in Figure 4.51(a-c). Evidently, the *TT* induced the swirl flow as depicted in Figure 4.52(a). The swirl flow is a basic flow commonly generated by any twisted tape. The photograph of fluid flowed through the tube with *WT* in Figure 4.51(b) demonstrated that fluid stream was split into two streams and then recombined behind the wing, producing double longitudinal vortices. The flow behavior caused by the alternate axis in the *WT-A* is presented in Figure 4.51(c). According to the tape geometry, two streams of fluid (in front and back sides of tape) were split into four streams around the alternate axis. Then they recombined into two streams with the stream collision behind the alternate point. The flow caused by the centre wing induced by the *WT-A* is similar to that induced by the *WT* as shown in Figure 4.51(b).

The sketch of flow phenomena induced by each twisted tape is demonstrated in Figure 4.52, which are: (1) the *TT* generates only single direction-swirl flow, (2) the *WT* provides a single-direction swirl flow caused by the twisted tape together with vortices induced by the wings (3) the *TA* generates double-direction swirl flows and

(4) a *WT-A* offers combined actions from both *WT* and *TA* twisted tape. As mentioned in above section, the wings on WT appear on one side/twist ratio (left or right) of the tape while those on *WT-A* appear on two sides (left and right) as a consequence of the generating the alternate axis. Therefore, the additional flow disturbance by the *WT-A* over that by the *WT* is not only caused by the periodic change of swirl direction but also a better distribution of the wings in the tube. As described above, the *WT-A* thus gives a better fluid mixing and superior heat transfer rates as well as thermal performance factor over the other twisted tapes do. The thermal performance factor above unity offered by the *WT-A* indicates the potential of the *WT-A* in view point of pumping energy saving.

From the experimental results above, a least-squares regression was utilized to yield the equations. These equations are expressed in terms of Reynolds number (*Re*), Prandtl number (*Pr)*, and angle of attack. Figure 4.53(a-c) shows a comparison of experimental values of Nusselt number, friction factor and thermal performance factor with the predicted values, obtained from those equations. Apparently, the predicted data agree well with the experimental data, within ±10% (*WT*) and ±6% (*WT-A*) for Nusselt number, ±10% (*WT*) and ±9% (*WT-A*) for friction factor and ±7% (*WT*) and ±4% (*WT-A*) for thermal performance, respectively.

The tube fitted with *WT*):

$$Nu = 0.232Re^{0.595}Pr^{0.4}(1+\tan\beta)^{0.202} \tag{4.26}$$

$$f = 14.039Re^{-0.505}(1+\tan\beta)^{0.406} \tag{4.27}$$

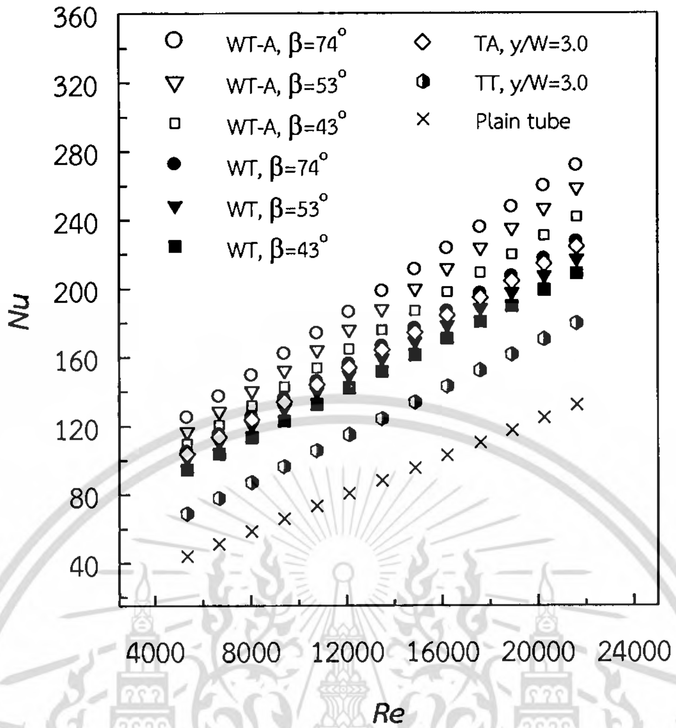
$$\eta = 4.629Re^{-0.166}(1+\tan\beta)^{0.067} \tag{4.28}$$

The tube fitted with *WT-A*:

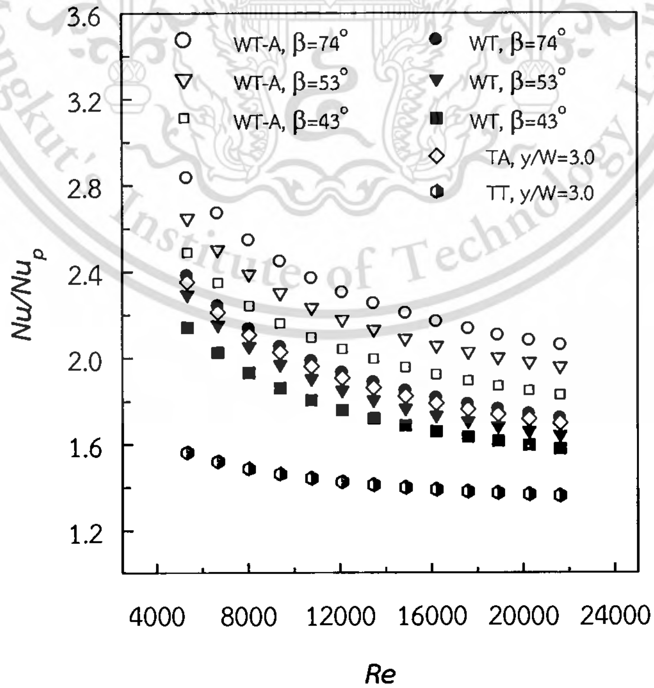
$$Nu = 0.358Re^{0.568}Pr^{0.4}(1+\tan\beta)^{0.129} \tag{4.29}$$

$$f = 20.445Re^{-0.504}(1+\tan\beta)^{0.283} \tag{4.30}$$

$$\eta = 6.772Re^{-0.194}(1+\tan\beta)^{0.035} \tag{4.31}$$

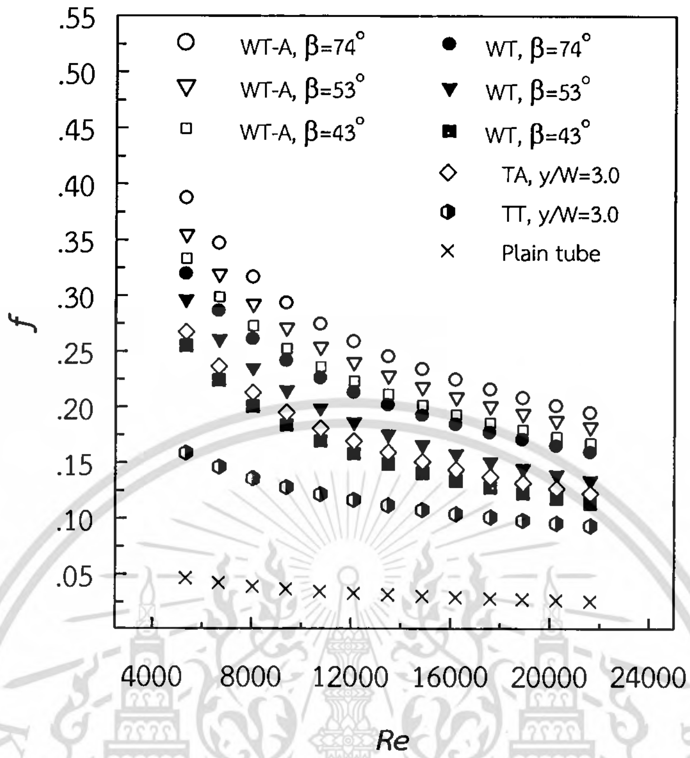


(a) Nusselt number

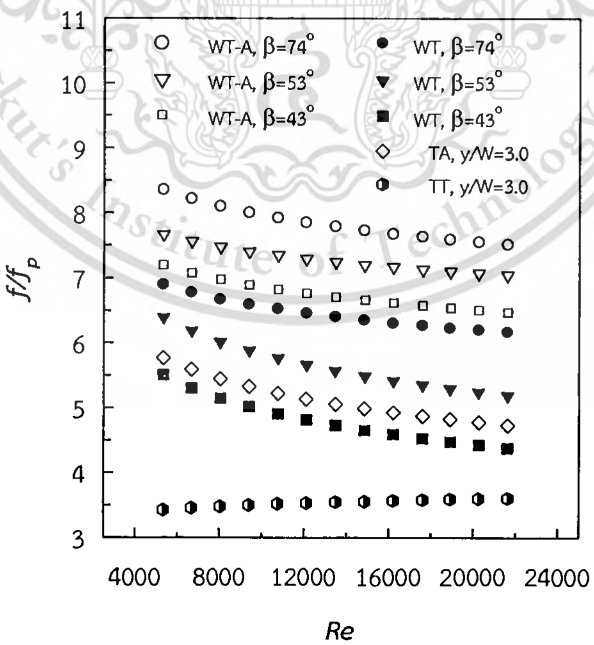


(b) Nu_t/Nu_p

Figure 4.48: Effect of wing twisted tape on Nusselt number.



(a) friction factor



(b) f_t/f_p

Figure 4.49: Effect of wing twisted tape on friction factor.

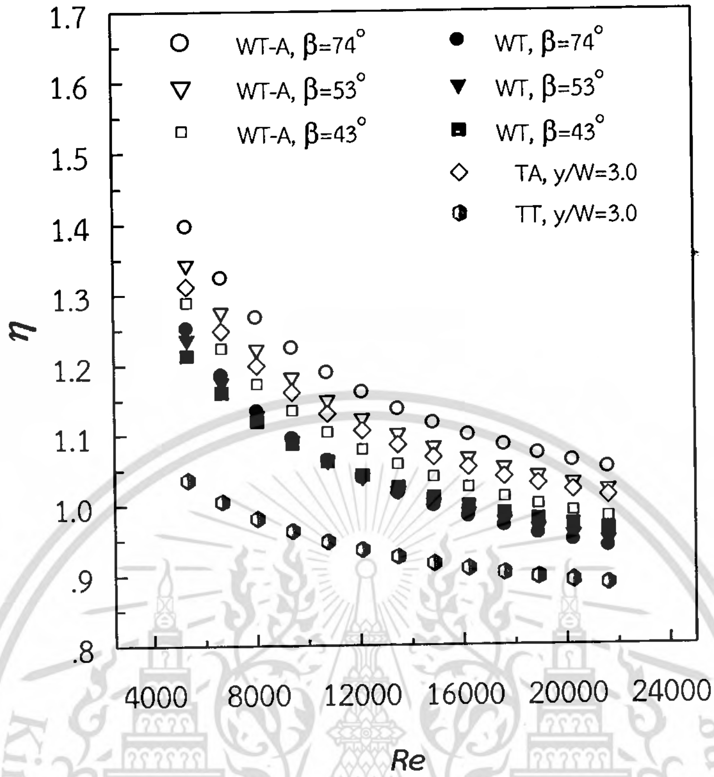
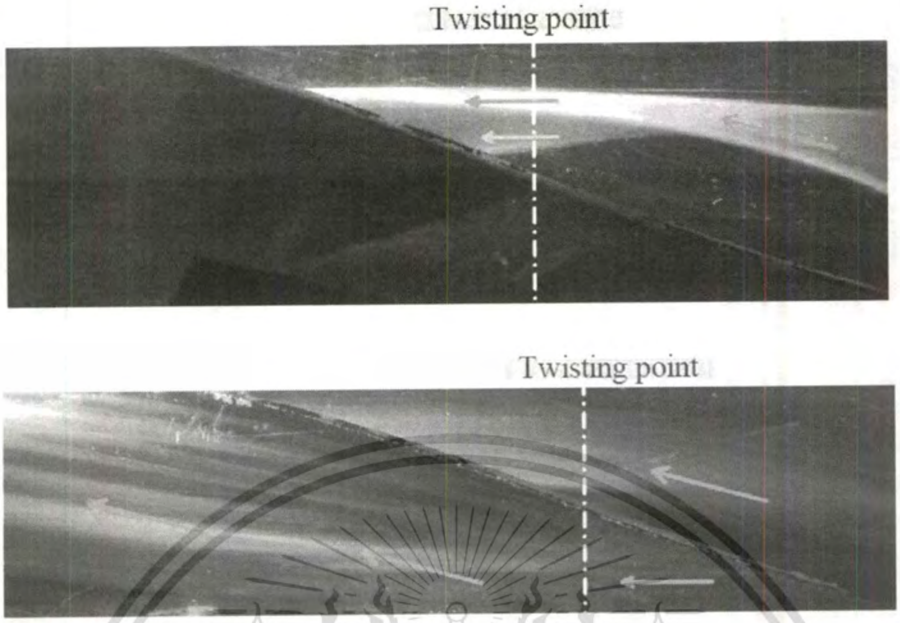
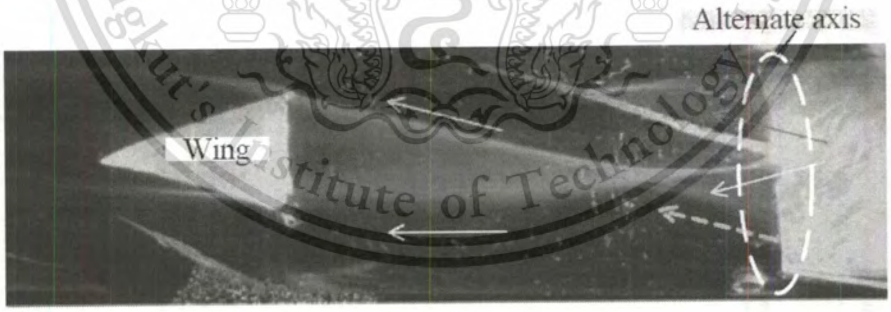


Figure 4.50: Effect of wing twisted tape on thermal performance factor.



(a) *TT*

(b) *WT*



(c) *WT-A*

Figure 4.51: Visualization of flow through tube with twisted tape inserts by smoke wire technique.

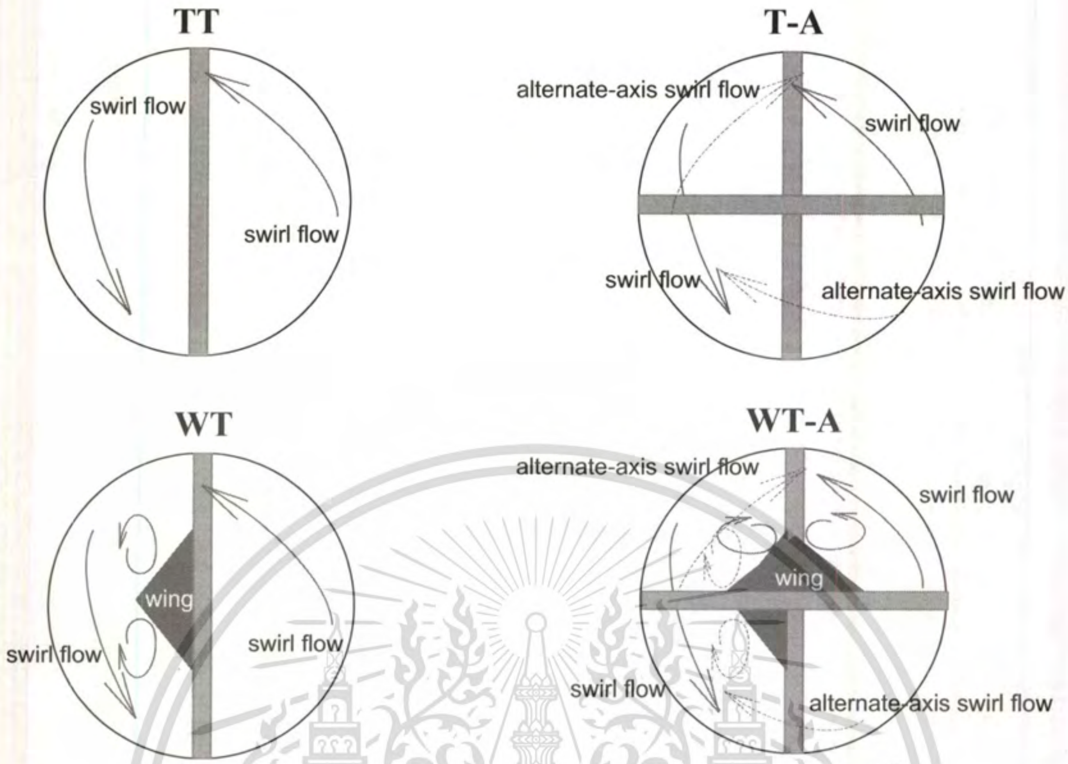
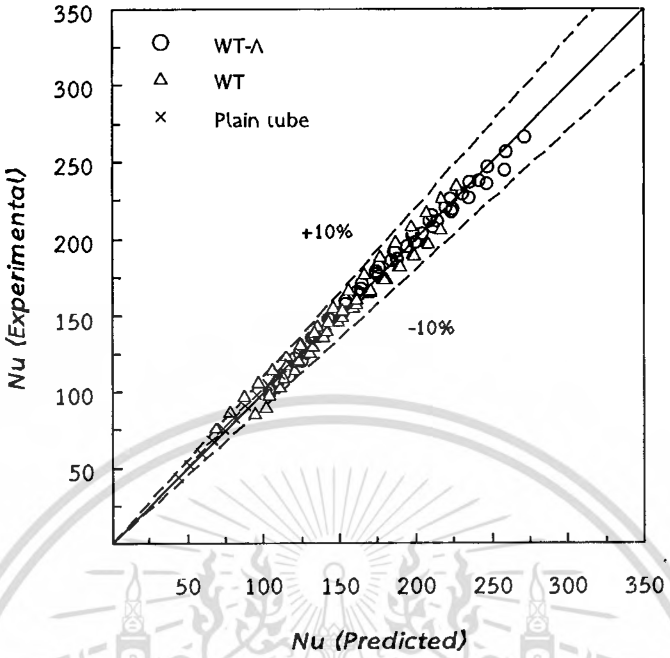
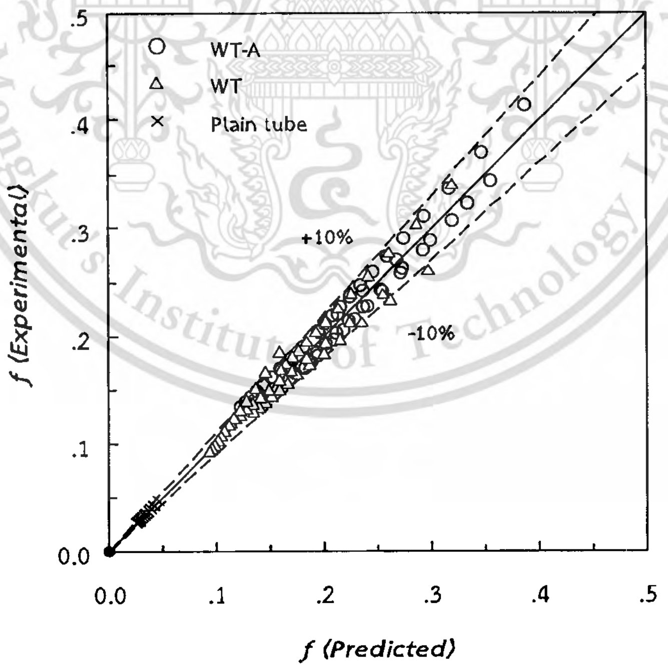


Figure 4.52: Sketch of flow phenomena in the front view of the tube with various twisted tapes.

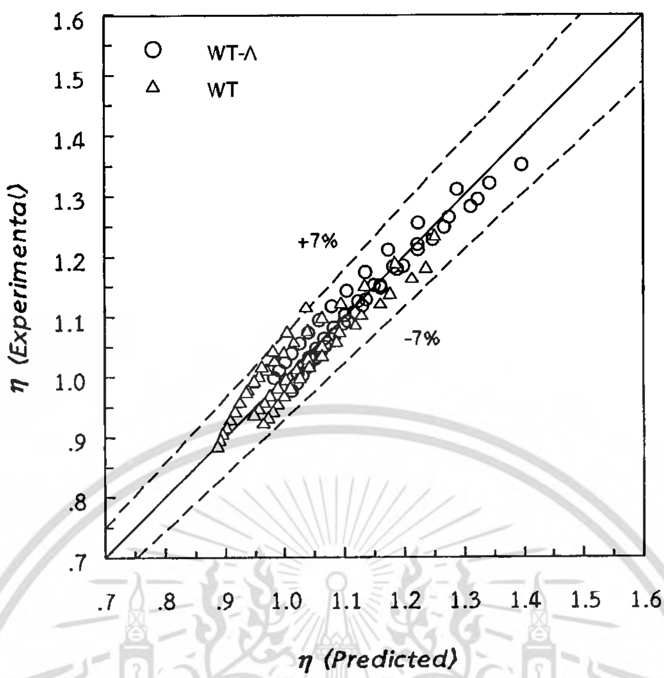


(a) Nusselt number



(b) friction factor

Figure 4.53: Validation of empirical correlations for (a) Nusselt number (b) friction factor and (c) thermal performance factor.



(c) thermal performance factor

Figure 4.54: Validation of empirical correlations for (a) Nusselt number, (b) friction factor and (c) thermal performance factor.

4.9 Comprehensive comparison

The comparison of heat transfer rate in tubes equipped with the various present modified twisted tapes (perforated twisted tape, delta-winglet twisted tape, twisted tapes with rectangular wing, perforated twisted tape with parallel wing, twin twisted tapes, wing twisted tape with alternate axis) is shown in Figure 4.55. The typical twisted tape (*TT*) was chosen for comparison. The general trend found for all tapes is that the heat transfer rate increases with increasing Reynolds number. Comparatively, all present modified twisted tapes give higher heat transfer rate than *TT*, due to the higher flow disturbance near the core or edge twisted tape. It is found that, the perforated twisted tape with parallel wing offers higher the heat transfer rate than those other modified tapes.

The present results are assessed by comparison with the results obtained from similar turbulent flows for friction factor and plotted in Figure 4.56. The result from the perforated twisted tape with parallel wing yields higher friction factor than those from the other modified tapes. The friction factor of the perforated twisted tape provides the highest friction factor than those other tape inserts.

Several works form the modified twisted tapes have been focus on the effect of modified tape geometry for enhancing the heat transfer rate and improving the thermal performance factor of the tube fitted with this kind swirl generator (Figure 4.57). All modified twisted tapes showed that the thermal performance factor can improve by modifying it at the core twisted tape (perforated twisted tape, perforated twisted tape with parallel wing, twin twisted tapes, wing twisted tape with alternate axis) or edge twisted tape (delta-winglet twisted tape, twisted tapes with rectangular wing, perforated twisted tape with parallel wing, wing twisted tape with alternate axis) and found that the thermal performance factor can improve higher than those the typical twisted tape (*TT*) under the same conditions. The thermal performance factor of the twin twisted tapes provide the highest friction factor than those other tape inserts.

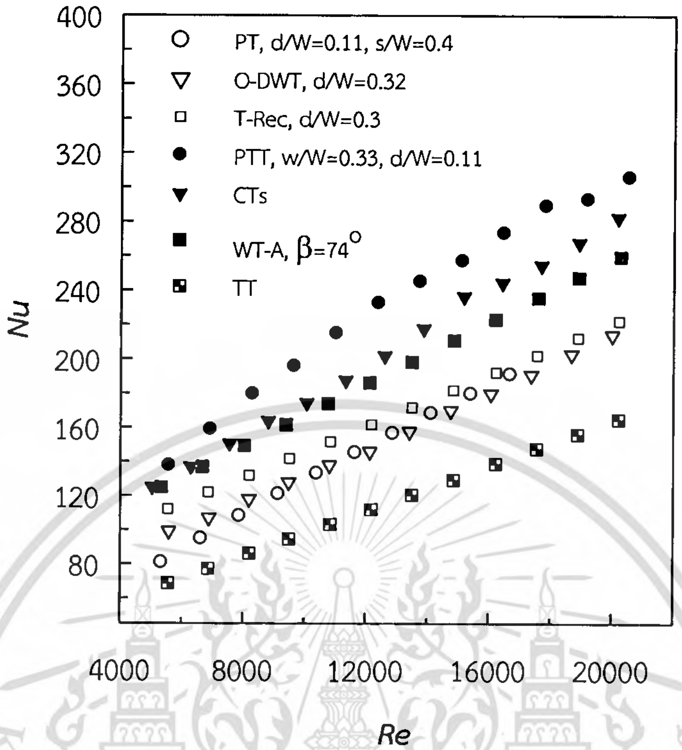


Figure 4.55: Comparison of Nusselt number with various modified twisted tapes.

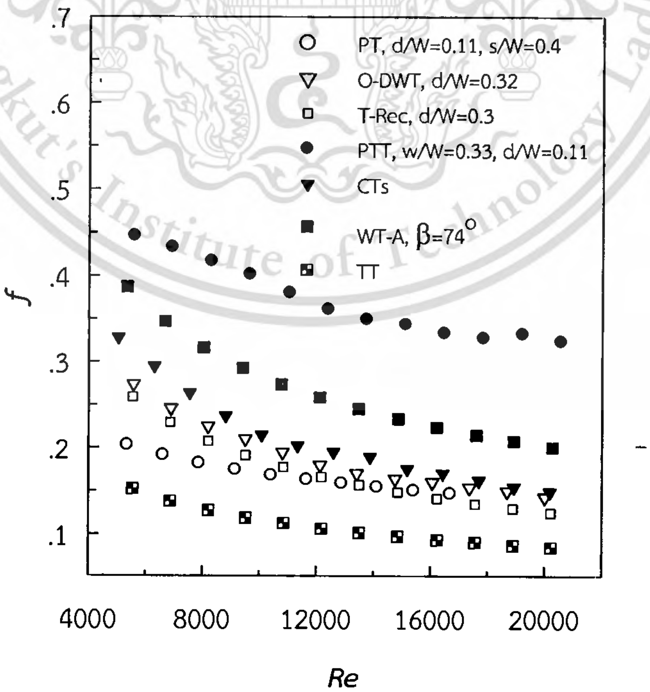


Figure 4.56: Comparison of friction factor with various modified twisted tapes.

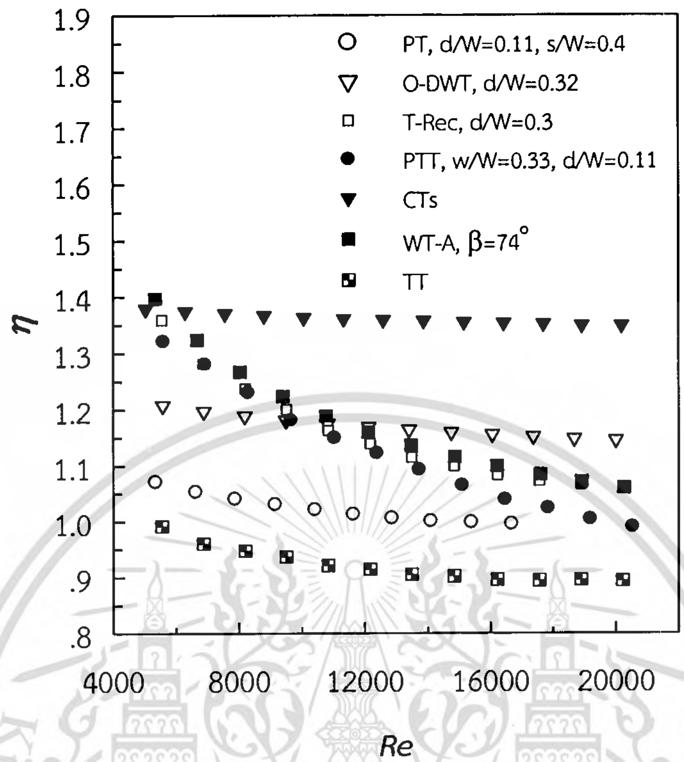


Figure 4.57: Comparison of thermal performance factor with previous work.

Table 4.1: Typical twisted tape (*TT*)

y/W	Nu/Nu_p	f/f_p	η
3.0	1.53	3.71	0.99
4.0	1.42	3.32	0.96
5.0	1.33	3.05	0.93

*Re = 5548

Table 4.2: Perforated twisted tape (*PT*)

	Nu/Nu_p	f/f_p	η
y/W=3.0			
s/W=0.4	1.85	5.17	1.07
s/W=0.6	1.75	4.70	1.05
s/W=0.8	1.69	4.41	1.03
y/W=4.0			
s/W=0.4	1.64	3.87	1.05
s/W=0.6	1.56	3.68	1.01
s/W=0.8	1.47	3.47	0.97
y/W=5.0			
s/W=0.4	1.51	3.59	0.99
s/W=0.6	1.43	3.38	0.96
s/W=0.8	1.40	3.20	0.95
d/W=0.11			
s/W=0.4	1.85	5.17	1.07
s/W=0.6	1.75	4.70	1.05
s/W=0.8	1.69	4.41	1.03
d/W=0.14			
s/W=0.4	1.87	5.69	1.05
s/W=0.6	1.77	5.17	1.02
s/W=0.8	1.70	4.89	1.01
d/W=0.17			
s/W=0.4	1.95	6.21	1.06
s/W=0.6	1.84	5.64	1.03
s/W=0.8	1.77	5.29	1.02

* Re = 5354

Table 4.3: Delta-winglet twisted tape (*O-DWT/S-DWT*)

O-DWT	Nu/Nu_p	f/f_p	η
y/W=3.0			
d/W=0.11	1.94	5.4	1.15
d/W=0.21	2.19	6.2	1.19
d/W=0.32	2.52	7.2	1.24
y/W=4.0			
d/W=0.11	1.52	4.6	1.09
d/W=0.21	1.71	5.4	1.06
d/W=0.32	2.03	6.8	1.11
y/W=5.0			
d/W=0.11	1.39	4.2	1.02
d/W=0.21	1.58	4.8	1.09
d/W=0.32	1.77	6.0	1.00

S-DWT	Nu/Nu_p	f/f_p	η
y/W=3.0			
d/W=0.11	1.87	5.4	1.14
d/W=0.21	2.00	5.6	1.17
d/W=0.32	2.13	6.0	1.21
y/W=4.0			
d/W=0.11	1.52	4.6	1.03
d/W=0.21	1.61	5.2	1.04
d/W=0.32	1.74	5.6	1.07
y/W=5.0			
d/W=0.11	1.32	4.2	1.00
d/W=0.21	1.39	4.4	1.09
d/W=0.32	1.52	5.0	0.97

* Re = 2975

Table 4.4: Twisted tapes with rectangular wing (*T-Rec*)

	Nu/Nu_p	f/f_p	
OW-T	2.82	6.73	
PW-T	2.62	6.24	
	Nu/Nu_p	f/f_p	η
T-Rec			
d/w =0.1	2.13	5.12	1.25
d/w =0.2	2.33	6.07	1.29
d/w =0.3	2.49	6.34	1.36

* Re = 5548

Table 4.5: Perforated twisted tape with parallel wing (*PTT*)

	Nu/Nu_p	f/f_p	η
w/W=0.11			
d/W=0.11	2.03	8.08	1.01
d/W=0.33	1.94	8.44	0.95
d/W=0.55	1.88	8.81	0.91
w/W=0.22			
d/W=0.11	2.42	9.18	1.16
d/W=0.33	3.39	9.55	1.11
d/W=0.55	2.29	9.92	1.07
w/W=0.33			
d/W=0.11	2.90	10.65	1.32
d/W=0.33	2.87	11.38	1.28
d/W=0.55	2.79	12.12	1.21

* Re = 5598

Table 4.6: Twin twisted tapes (*CTs/CoTs*)

	Nu/Nu_p	f/f_p	η
CTs			
y/W=2.5	2.87	8.33	1.39
y/W=3.0	2.32	6.74	1.24
y/W=3.5	1.82	5.54	1.12
y/W=4.0	1.66	5.15	1.03
CoTs			
y/W=2.5	1.97	5.93	1.10
y/W=3.0	1.74	5.15	1.03
y/W=3.5	1.61	4.35	0.97
y/W=4.0	1.37	3.96	0.92
TT			
y/W=3.0	1.58	4.76	0.90
y/W=3.5	1.39	3.76	0.87
y/W=4.0	1.30	3.17	0.84

* Re = 3771

Table 4.7: Wing twisted tape (*WT-A/WT*)

	Nu/Nu_p	f/f_p	η
WT-A			
$\beta=43^\circ$	2.49	7.20	1.29
$\beta=53^\circ$	2.65	7.66	1.34
$\beta=74^\circ$	2.83	8.35	1.40
WT			
$\beta=43^\circ$	2.14	5.50	1.21
$\beta=53^\circ$	2.30	6.40	1.24
$\beta=74^\circ$	2.38	6.89	1.25

* Re = 5337, *y/W*=3.0

Table 4.8: Comprehensive comparison of different tape inserts

Type of modified twisted tapes	Re	Nu/Nu_p	f/f_p	η
*PT, d/W=0.11, s/W=0.4, y/W=*3.0	5354	1.85	5.17	1.07
*O-DWT, d/W=0.32, y/W=*3.0	2975	2.52	7.20	1.21
*T-Rec, d/w=0.3, y/W=*3.0	5548	2.49	6.34	1.36
*PTT,w/W=0.33,d/W=0.11, y/W=*3.0	5598	2.90	10.65	1.32
*CTs, y/W=*3.0	3771	2.87	8.33	1.38
WT-A, β=74°, *y/W=*3.0	5337	2.83	8.35	1.40
*TT, y/W=*3.0	5548	1.93	3.71	0.99

CHAPTER 5
Conclusions and Future Work

5.1 Open remarks

Mostly, heat transfer rates associated by the modified twisted tapes are higher than those the typical twisted tape (*TT*). However, an improve heat transfer is unavoidably accompanied by an increase of pressure drop. To obtain optimum conditions, the proper geometries of the modified twisted tape are necessary. Effects of the modified twisted tapes (perforated twisted tape, delta-winglet twisted tape, twisted tapes with rectangular wing, perforated twisted tape with parallel wing, twin twisted tapes, wing twisted tape with/without alternate axis) on the heat transfer rate (*Nu*), friction factor (*f*) and thermal performance factor characteristics are described and reported in this thesis. Major findings can be summarized as follows:

5.1.1 Typical twisted tape

5.1.1.1 The typical twisted tapes (*TTs*) have been utilized in circular tube to enhance heat transfer for the Reynolds number between 5500 and 20,200 under uniform heat flux condition. With the presence of twist tape, heat transfer and friction factor are promoted over those found in the plain tube, case. Heat transfer rate, friction factor and thermal enhancement factor increase tape twist ratio decreases.

5.1.1.2 In the range determined, the tubes with *TTs* at twist ratios, y/W = 3.0, 4.0, 5.0 give Nusselt number over that of the plain tube by around 37.3%, 27.5% and 19.9%, respectively which are accompanied with friction loss penalties over that of the plain tube by around 230%, 194% and 170%, respectively. The highest thermal enhancement factor of 0.99 is obtained by utilizing the tape with the smallest twist ratio (y/W = 3.0) at Reynolds number of 5500.

5.1.2 Perforated twisted tape

5.1.2.1 The present study explores the effect of perforated twisted tapes (*PTs*) on the heat transfer, friction factor and thermal performance at different tape-twist ratios (y/W) and pitch ratios (s/W). It is found that *PTs* cause a considerable increase in heat transfer rate and friction factor in comparison with the typical twisted tape (*TT*). The increase in heat transfer rate by using *PT* is found to be in range of 36 to 85%, over those of the corresponding plain tube. Heat transfer rate increases with decreasing tape-twist ratio and pitch ratio. The average heat transfer and friction factors generated by *PTs* are approximately 25% and 24% higher than those the given by *TTs*.

5.1.2.2 Thermal performance evaluation analysis shows that the maximum thermal performance factor of 1.07 is offered by the *PT* with the smallest tape-twist ratio (*y/W* = 3), perforation hole diameter ratio (*d/W* = 0.11) and pitch ratio (*s/W* = 0.4). It can be observed that the thermal performance factors obtained by the uses of the modified tapes proposed in the present work (*PT*) are mostly below unity, especially at high Reynolds number. The results are caused by the dominant effect of the substantial increase in pressure drop over that of the considerable improvement in heat transfer.

5.1.3 Delta-winglet twisted tape

5.1.3.1 The enhancement of the heat transfer in a tube fitted with delta-winglet twisted tapes which act as swirl generator is experimentally investigated. The values of Nusselts number and friction factor in the test tube equipped with delta-winglet twisted tape are noticeably higher than that in the plain tube and also tube equipped with typical twisted tape (*TT*). Nusselt number and friction factor increase with decreasing of twist ratio (*y/W*) and increasing depth of wing cut ratio for all Reynolds number studied. In addition, the *O-DWT* gives higher Nusselt number and friction factor than that of the *S-DWT*.

5.1.3.2 The thermal performance factor in the tube with *O-DWT* is greater than that with *S-DWT* and the factor increases with decreasing Reynolds number and increasing twist ratio. Over the range considered, the performance factor in the tubes equipped with the *O-DWT* and *S-DWT* are found to be around 0.92 to 1.24 and 0.88 to 1.21, respectively.

5.1.3.3 In this study, several *DWT* tapes give the thermal performance factor higher than unity. It is obvious that the *DWT* performs better heat transfer enhancement than that *TT*. It indicates that the heat exchanger fitted with *DWT* is more compacted than one with the typical twisted tape. Again, the *DWT* can be replaced any of the *TT* efficiently to reduce the size of the heat exchanger.

5.1.4 Twisted tapes with rectangular wing

5.1.4.1 At a similar operating condition, the *OW-T* gives the highest heat transfer rate, friction factor and thermal performance factor and the *PW-T* performs better than the *TT*. For the range examined, the mean Nusselt numbers increases of the tubes equipped with the *TT*, *OW-T* and *PW-T*, are found to be 27.5% 110% and 100% over the plain tube, respectively. In addition, the average thermal performance factor of the *OW-T* is higher than those of the *TT* and the *PW-T* at around 24.9% and 2.74%, respectively. The maximum thermal performance factor of 1.4, for the *OW-T* is found at low Reynolds number.

5.1.4.2 The heat transfer rate in the heat exchanger tube can be enhanced by insertion of *T-Recs* while it brings about an energy loss of the fluid flow. The mean heat transfer rates of the tubes with *T-Recs* at *d/W* = 0.1, 0.2, and 0.3 are respectively 78%, 91% and 100%

over that of the plain tube. However, the percentage increase in friction factor is much higher than that in Nusselt number at the same Reynolds number.

The thermal performance factors of the tubes equipped with *T-Recs* considerably increase with decreasing Reynolds number and increasing wing-depth ratio. At the given Reynolds number, *T-Recs* consistently provide higher performance factor than *TA* and *TT*. For the present range, the maximum thermal performance factor of 1.36 is achieved with the use of *T-Recs* at wing-depth ratio of 0.3 and Reynolds number of 5500.

5.1.5 Perforated twisted tape with parallel wing

5.1.5.1 Augmentation of heat transfer rate in heat exchanger tubes by means of perforated twisted tapes (*PTT*) inserts is investigated experimentally. The results showed those heat transfer and friction factors were significantly influenced by the presences of wings and holes on *PTTs*. Both heat transfer and friction increased with the increase of wing depth ratio (*w/W*) and the decrease of perforation hole diameter ratio (*d/W*).

5.1.5.2 Due to the dominant effect of increased heat transfer over that of increased friction factor, the thermal performance factor was found to be increased as wing depth ratio (*w/W*) increased and hole diameter ratio (*d/W*) decreased.

5.1.6 Twin twisted tapes

5.1.6.1 The present paper shows the feasibility of convection heat transfer enhancement by inserting *CTs (counter-swirl flow generators)* and *CoTs (co--swirl flow generators*) for turbulent flow with a uniform heat flux. For general observation, it is found that heat transfer, friction factor, and thermal performance factor increase as twist ratio (*y/W*) decreases. In addition, the Nusselt number increases with increasing Reynolds number while the opposite trends are found for the friction factor and the thermal enhancement index.

5.1.6.2 The *CTs* (*counter-swirl tapes*) can enhance heat transfer more efficiently than the *CoTs* (*co-swirl tapes*). The quantitative results show that, heat transfer rates for *CTs* are around 12.5 to 44.5% higher than those for *CoTs* and around 17.8 to 50% higher than those for single twisted tape (*ST*) *or* typical twisted tape (*TT*). The maximum thermal performance factors obtained at the constant pumping power by using *CTs* with *y/W* = 2.5, 3.0, 3.5 and 4.0, are 1.39, 1.24, 1.12 and 1.03, respectively while those obtained by using *CoTs* with the same range of *y/W* are 1.1, 1.03, 0.97 and 0.92. The achieved results indicate that the modified swirl flow (*counter-swirl flow*) generated by the *CTs* is a promising approach for heat transfer enhancement.

5.1.7 Wing twisted tape with/without alternate axis

5.1.7.1 The present study explored the effects of the wing and alternate axis of the modified twisted tapes on the heat transfer and fluid friction characteristics in a heat exchanger tube. The combined actions of the wing and alternate axis in the *WT-A* resulted in a better fluid mixing and thus heat transfer enhancement compared to those induced by wing alone (*WT*) or alternate axis alone (*TA*). The results also revealed that Nusselt number (Nu) increased with the increasing angle of attack. The *WT-A* generated higher pressure drop within the tube than *WT, TA* and also *TT*, due to the flow disturbance caused by both the wing and the alternate-axis and pressure drop became larger as the angle of attack increased.

5.1.7.2 The similar trend also found for the effects of the twisted tape form and the angle of attack on thermal performance factor which is the consequential results from both heat transfer rate and pressure drop. In the present range, the *WT-A* with the largest angler of attack of 74° gave the highest Nusselt number, friction factor as well as thermal performance facto. Mean values of these three parameters provided by the *WT-A* attack of 74° were respectively, 17.7%, 30.6% and 7.8% higher than those in the tube with *WT*, 20.8%, 53% and 4.9% higher than those in the tube with *TA*, and 62%, 123% and 24% higher than those in the tube with *TT*.

5.2 General conclusion

From the present results, most of the modified twisted tapes are not practically feasible in terms of energy saving for general cases. However, they can be applied at very low Reynolds number or in places where pumping power is not important. It should also be stated that the geometry of twisted tape insert is progressively developed; the twisted tape with appropriate geometry may lead to an overall energy gain because of a more efficient heat transfer and a lower pressure drop.

5.3 Future work

Investigation study the simulation physical behavior of the thermal and fluid flow in the tube fitted with modified twisted tape under uniform heat flux conditions in the turbulent flow regime.

Study investigates the heat transfer properties over developing and developed flow regimes, the pressure drop coefficients and the thermal performance factors of tubular flow with twist tapes.

Numerical study on heat transfer and friction factor characteristics on laminar and tubular flow in a tube with twist tapes.

Investigated and configuration optimization performed based on the heat transfer enhancement efficiency using computational fluid dynamics (CFD) modeling.

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Heat transfer and thermal performance characteristics of heat exchanger tube fitted with perforated twisted-tapes

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Heat and Mass Transfer
Wärme- und Stoffübertragung

ISSN 0947-7411

Heat Mass Transfer
DOI 10.1007/s00231-011-0943-0

Springer

Heat Mass Transfer (2012) 48:881–892
DOI 10.1007/s00231-011-0943-0

ORIGINAL

Heat transfer and thermal performance characteristics of heat exchanger tube fitted with perforated twisted-tapes

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Received: 16 March 2011 / Accepted: 13 November 2011 / Published online: 23 November 2011

Abstract Twisted tape insert was applied as a swirling flow generator for the passive heat transfer enhancement in the present work. The influences of the perforated twisted tapes (PTs) on the heat transfer, pressure loss and thermal performance characteristics were investigated experimentally. The experiments were performed under uniform wall heat flux condition by using PTs with $y/W = 3$, 4 and 5, $d/W = 0.11$, 0.14 and 0.17 and $s/W = 0.4$, 0.6 and 0.8 where y is a twist length, d is a perforation hole diameter, s is a spacing between holes (pitch) and W is a tape width. The experimental results reveal that Nusselt number increased with decreasing s/W and y/W and increasing d/W. For the present range, the maximum heat transfer was obtained by utilizing the tape with $s/W = 0.4$, $d/W = 0.17$ and $y/W = 3$, which is higher than those obtained from the plain tube with and without typical twisted tape by around 27.4 and 86.7%, respectively. In addition, the empirical correlations for Nusselt number, friction factor and thermal performance are also proposed in the present paper.

List of symbols

A	Heat transfer surface area
C_p	Specific heat of fluid
D	Inside diameter of the test tube
d	Diameter of perforated
f	Friction factor $= \Delta P/((L/D)(\rho U^2/2))$
h	Heat transfer coefficient
I	Current
k	Thermal conductivity of fluid
L	Length of the test section
M	Mass flow rate
Nu	Nusselt number
P	Pressure of flow in stationary tube
ΔP	Pressure drop
Pr	Prandtl number
Q	Heat transfer rate
Re	Reynolds number $= \rho UD/\mu$
s	Spaced-pitch length of perforated, mm
t	Thickness of the test tube
T	Temperature
$\tilde{T}$	Mean temperature
U	Mean axial flow velocity
V	Voltage
W	Twisted tape width
y	Twisted tape pitch

Greek letters

ρ	Fluid density
δ	Twisted tape thickness
μ	Fluid dynamic viscosity
η	Thermal performance factor

Subscripts

b	Bulk
c	Convection
i	Inlet
o	Outlet
p	Plain
s	Surface

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t Twisted tape
w Water

Abbreviations
PT Perforated twisted tape
TT Typical twisted tape

1 Introduction

In recent years, heat transfer enhancement devices (HTED) have become more prevalent in many engineering applications (e.g., heat exchanger, nuclear/chemical reactor, drying process, refrigeration, gas turbine and more). Heat transfer enhancement techniques (passive and active methods) have been developed for saving materials and reducing sizes of the equipments [1–3]. A twisted tape as a swirl generator is one of widely used devices for heat transfer enhancement. Functionally, a twisted tape induces swirl flow, enhances fluid turbulence and thus fluid mixing which directs toward reducing the thermal boundary thickness. An application of a twisted tape concerns with a trade-off between the heat transfer and pressure drop. Early research works have shown that twisted tapes with proper designs offer the augmentation of heat transfer rate with reasonable pressure loss penalty, and thus overall energy savings.

Some modified twisted tapes were designed to minimize pressure loss [1–5]. Patil [1] examined heat transfer and pressure drop characteristics of in tubes with varying width twisted tape inserts. Similarly, Eiamsa-ard et al. [2] performed simulation on the heat transfer in a tube fitted with loose-fit twisted tape. Effects of the clearance ratio defined as ratio of clearance between the edge of tape and tube wall to tube diameter on the heat transfer enhancement, friction factor and thermal performance factor were reported. Saha et al. [3] numerically studied the heat transfer and flow friction characteristics of laminar swirl flow through a circular tube fitted with regularly spaced twisted-tape elements. Regularly-spaced twisted tapes were also utilized in form of dual twisted tapes [4]. Eiamsa-ard et al. [5] investigated the heat transfer, friction factor and thermal performance characteristics in a tube with short-length twisted tape at several tape length ratios. Jaisankar et al. [6] introduced spacing or rod at the trailing edge of twisted tape, as an alternative form of twisted tape insert. Although all modified twisted tapes mentioned above generated lower pressure drop than typical twisted tape (TT), they yielded considerably lower heat transfer rate, resulting in diminishment of overall heat transfer performance. The results are responsible by a lower swirl intensity/smaller swirling number produced by the modified twisted tapes as compared to those generated by typical ones.

There have been attempts to improve overall heat transfer enhancement performance by different design of twisted tape, with aspect to promote fluid mixing and thus heat transfer rate. Rahimi et al. [7] presented the effect of the new modified twisted tape (jagged/perforated/notch tape) on the heat transfer, friction factor and thermal performance factor. They results showed that the jagged twisted tape offered superior heat transfer rate and thermal performance factor to other tapes including other tape the typical one. The enhanced heat transfer and thermal performance by the jagged twisted tapes were 31 and 22% compared to those of the typical one. Effect of peripherally-cut twisted tape with alternate axis on the fluid flow and heat transfer enhancement characteristic was investigated by Seemawute and Eiamsa-ard [8]. Because of an extra flow mixing along the tube wall of the test tube, the peripherally-cut twisted tape provided higher heat transfer, friction factor and thermal performance than the typical tape. Saha [9] studied the heat transfer enhancement in rectangular and square ducts having corrugated internal surfaces in accompany by twisted-tape inserts with and without oblique teeth. The combined effect by axial corrugation together with twisted-tapes resulted in superior overall heat transfer performance to those of the individual effects. In addition the tape with oblique teeth gave better thermal performance factor than the one without oblique teeth. Murugesan et al. [10, 11] reported that their modified twisted tapes in forms of trapezoidal and square-cut twisted tapes gave higher heat transfer rate, friction factor and thermal enhancement factor than a TT, as a result of an additional disturbance and secondary flow in the vicinity of the tube wall induced by the modified part. Wongcharee and Eiamsa-ard [12] studied the influence of the twisted tapes with alternate-axes and wings (triangular, rectangular and trapezoidal wings) with three different wing-chord ratios of 0.1, 0.2 and 0.3 on heat transfer and thermal performance characteristics. The alternate-axis was made by arranging each plane of twisted tape to 60° difference relative to the adjacent plane. They found that the heat transfer rate, friction factor and thermal performance given by all modified twisted tape were consistently higher than those typical tapes due to higher turbulence neat the tube wall, leading to better heat transfer rate. They also reported that that Nusselt number, friction factor and thermal performance increased with increasing wing-chord ratio and the tape with trapezoidal wings provided the higher thermal performance than the ones with other shaped wings. Murugesan et al. [13] presented the effect of combined device between twisted tape and wire-nail on the heat transfer enhancement friction factor and thermal performance factor. As found, the combined device gave considerably higher heat transfer, friction factor and thermal performance factor than the TT, due to the improvement in fluid mixing associated with the presence of wire nails.

Several researchers also reported that overall performance factor was significantly changed depending on geometrical conditions of tapes. Eiamsa-ard et al. [14] employed twin counter and co twisted tapes to produce counter and co swirl flows. Apparently, the tapes with counter arrangement gave better heat transfer rate and also overall performance factor than the ones with co-arrangement. Heat transfer and flow friction characteristics in a tube equipped alternate clockwise and counter-clockwise twisted tapes were examined by Eiamsa-ard and Promvonge [15] for turbulent region and Wongcharee and Eiamsa-ard [16] for laminar region. The results showed that the heat transfer rate and thermal performance factor by the uses the tapes increased with the decrease of twist ratio and the increase of twist angle value. Eiamsa-ard et al. [17] used twisted tapes consisting of centre wings with three different angles of attack in a heat exchanger. It was found that the heat transfer rate increased with increasing angle of attack. In addition, the twisted tapes consisting was found to be effective for heat transfer enhancement, giving thermal performance factor higher than that TT due to a vortex generated by the wing and a strong collision of the recombined streams behind each alternate point. Eiamsa-ard and Promvonge [18] described the effect of twisted tape with serrated edge insert with different serration width ratios and serration depth ratios on heat transfer and pressure loss behaviors in a heat exchanger tube. They observed that the heat transfer rate increased with the rise in the depth ratio but decreased with raising the width ratio. The thermal performance factors of the twisted tape with serrated edge were found to be above unity indicating that is advantageous over the TT. Eiamsa-ard et al. [19] utilized oblique delta-winglet twisted-tapes and straight delta-winglet twisted tapes with three depth of wing cut ratios, for heat transfer enhancement. They results reveled that heat transfer, friction factor and thermal performance in a tube with the delta-winglet twisted tape increased with increasing depth of wing cut ratio. The oblique delta-winglet tape was more effective turbulator as it gave higher heat transfer coefficient than the straight delta-winglet one. Eiamsa-ard et al. [20] examined the effects of peripherally-cut twisted tape insert on heat transfer enhancement characteristics. Nine different peripherally-cut twisted tapes with constant twist ratio and different three tape depth ratios each with three different tape width ratios were applied in laminar and turbulence regions. They observed that the heat transfer rate and friction factor of tube equipped with the peripherally-cut twisted tapes were significantly higher than those in the tube fitted with the TT, especially in the laminar flow regime. It was also demonstrated that heat transfer enhancement was improved as a depth ratio increased and a width ratio decreased.

Due to the trade-off between the heat transfer and pressure drop mentioned above, only properly modified twisted tapes provide higher thermal enhancement factors over those of typical twisted tapes (TTs) at the same pumping power. Regarding to the previous works mentioned above, it can be observed that the modification of twisted tape by introducing parts which induce extra fluid disturbance, potentially increases heat transfer rate with reasonable increase of pressure drop, resulting in superior thermal enhancement factor over TT. In the present work, the newly designed perforated twisted tapes (PTs) as shown in Fig. 1 are proposed. To the best of authors' knowledge, the heat transfer and friction factor characteristics in a circular tube with this type of the perforated twisted tape (PT) insert have not been reported. Holes were generated with prospect to promote fluid turbulence near the tube wall and the tape edge. The geometrical characteristics of modified twisted tapes subjected to the present study are (1) $y/W = 3, 4$ and 5 (2) $d = 2$, 2.5 and 3 mm (corresponding to $d/W = 0.11, 0.14$ and 0.17, respectively) and (3) $s/W = 0.4, 0.6$ and 0.8 where y is a twist length, d is a perforation hole diameter, s is a spacing between holes (pitch) and W is a tape width. The results are reported in terms of Nusselt number (heat transfer), friction factor and thermal performance. The results obtained from the tube fitted with the *PT* are compared with those from TT and plain tube.

2 Calculation of heat transfer and friction factor

In the present, water is used as the testing fluid. The independent parameters are (1) spacing between holes (pitch) (s), (2) perforation hole diameter (d), (3) tape-twist length per 180 degree of twist rotation (y), and (4) Reynolds number (Re).

Reynolds number can be expressed as

$$Re = UD/\nu \tag{1}$$

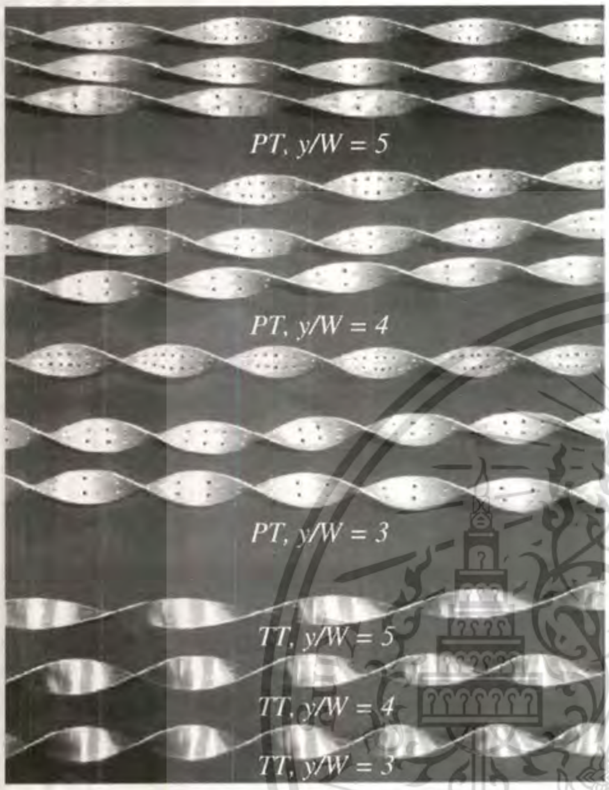
The mean heat transfer coefficients are evaluated from the measured temperatures and heat inputs (Q_{VI}). With heat added uniformly to fluid (Q_{water}) and the temperature difference of surface and bulk fluid ($T_s - T_b$), average heat transfer coefficient can be calculated from experimental data via the following equations:

$$Q_{water} = Q_{conv} = MC_p(T_o - T_i) \tag{2}$$

$$h = \frac{Q_{conv}}{A(\tilde{T}_s - T_b)} \tag{3}$$

in which, A is the heat transfer area of the circular tube wall.

(a)



(b)

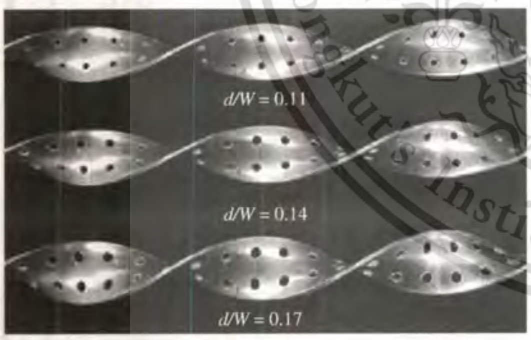


Fig. 1 Photograph of twisted tapes. **a** PTs with various twist and pitch ratios in comparison with TTs. **b** PTs with various perforation hole diameters

In the present experiment, the local wall temperatures (T_s) were measured at fifteen stations for calculation the mean wall temperatures $\tilde{T}_s = \sum T_s/15$ where the bulk temperature is calculated from the mean fluid temperature from the inlet and outlet of the water that using as the working fluid ($T_b = (T_o + T_i)/2$).

An average Nusselt number is obtained from average convective heat transfer coefficient as

$$Nu = hD/k \tag{4}$$

The friction factor is calculated from

$$f = 2\Delta PD/(\rho U^2 L) \tag{5}$$

3 Perforated twisted tape

Twisted tapes were made from aluminum strip with a thickness of 0.8 mm (δ) and a width of 18 mm (W). The tapes were formulated in three different twist lengths of $y = 54$ mm ($y/W = 3$), 72 mm ($y/W = 4$), and 90 mm ($y/W = 5$). PTs were then modified from the TT by drilling the small holes with three different hole diameters (d) of 2 mm ($d/W = 0.11$), 2.5 mm ($d/W = 0.14$) and 3 mm ($d/W = 0.17$) along the tape edge at three different space/pitch lengths of $s = 7.5$ mm ($s/W = 0.4$), 11 mm ($s/W = 0.6$), and 15 mm ($s/W = 0.8$) as shown in Fig. 1a, b.

4 Experimental apparatus and procedure

The experiments were carried out using an experimental facility as shown in Fig. 2. The test tube is made of copper and has an inner diameter of 19 mm (D), an outside diameter of 22 mm (D_o), a wall thickness of 1.5 mm (t), and a length of 1,000 mm (L). In experiments, each tape was inserted into a test tube, a rather long tube provided sufficient contact surface between the tapes and tube wall for the firm attachment of the tapes to the tube without a requirement of any extra fitting. The tube was wound with electrical heating wire covered with ceramic beads. During the test, the tube was heated by continually winding flexible electrical wire, providing a uniform heat flux condition. The electrical output power was controlled by a variac transformer to obtain a constant heat flux along the entire length of the test section by keeping the current less than 10 Amp. The outer surface of the test tube was well insulated to minimize convective heat losses to surroundings. In the experiment, the heat transfer losses from the test tube were found to be around 3 to 7% of the total heat input ($Q = IV$).

The inner and outer temperatures of the bulk water were measured at certain points with a multi-channel temperature measurement unit in conjunction with the resistance temperature detectors (RTDs). In the experiment, two thermocouples were placed at 760 mm (40D) upstream of the test section for measuring inlet water temperature while at 50 mm downstream of the test section for measuring outlet water temperature. Fifteen thermocouples were silver soldered to the test tube surface (embedded in v-groove tube surface) for measuring local wall temperatures. The positions of the thermocouples were 15, 85, 155, 225, 295,

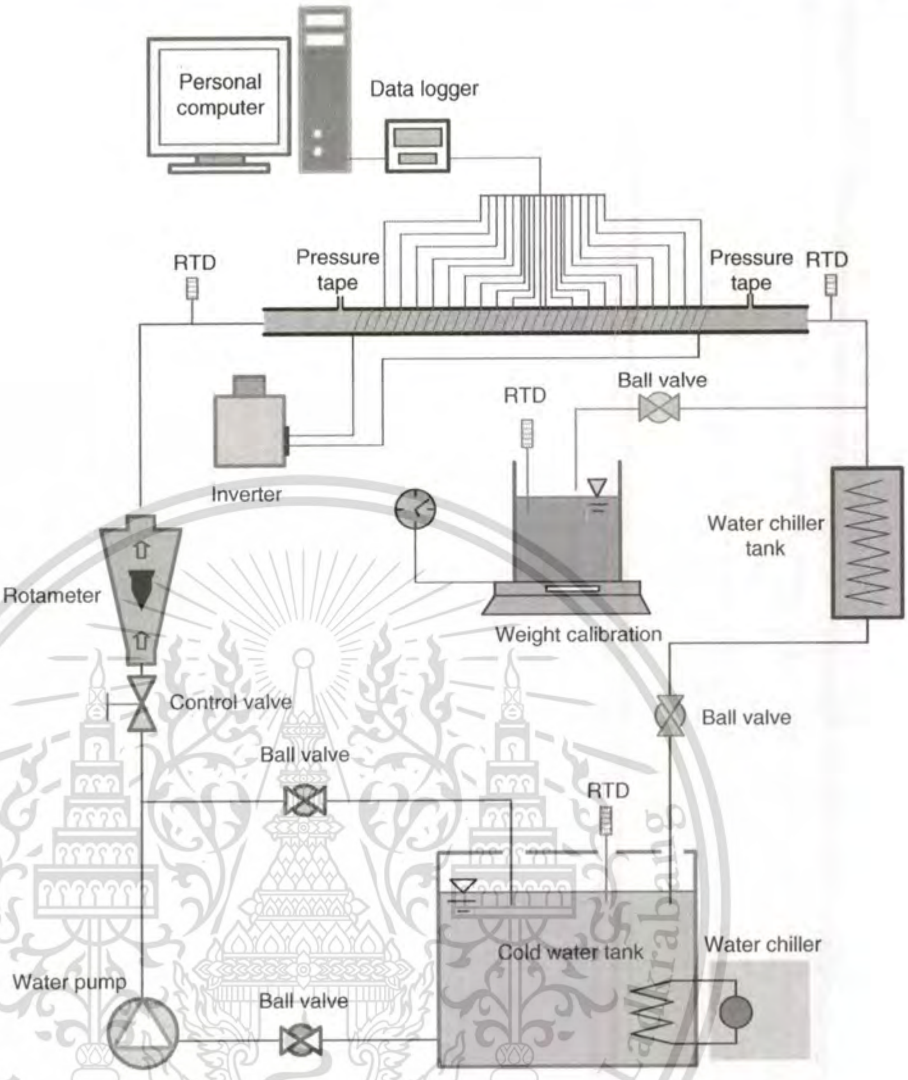


Fig. 2 Schematic diagram of the experimental heat transfer set-up

365, 435, 505, 575, 645, 715, 785, 855, 925 and 975 mm from the entrance of the test tube or heating section. The mean wall temperature was determined by means of calculation based on the reading of the Copper-Constantan thermocouples.

The test loop consists of a water pump, data logger, pressure transmitter, thermocouple/RTD, rotameter, and heat transfer test section as depicted in Fig. 2. In the apparatus setting above, the water was pumped to the overhead water tank then discharged through the rotameter to measure the volumetric flow rate and directed to the heat transfer test section. For each test run, it was necessary to record the data of temperature, volumetric flow rate and pressure drop of the bulk water at steady state conditions in which the inlet water temperature was maintained constant at 27°C. During each test, local wall temperatures, inlet and outlet water temperatures, pressure drop across the test section and flow velocity were measured. The isothermal friction factor was calculated from pressure drop measured by manometer at room temperature. In the experiment, the pressure taps were located around 50 mm upstream and 150 mm downstream of the test section. The length between both tapes was around 1,000 mm. Average Nusselt numbers were calculated using fluid properties at the overall bulk mean temperature. Apart from the PTs, the TT with three different tape-twist ratios (y/W) were also tested for comparison.

The Reynolds number of cold water was varied from 5,300 to 16,700. Uncertainties of measurements are estimated based on ANSI/ASME [21]. The uncertainties of axial velocity, pressure and temperature are found to be within ±7, ±5 and ±0.5%, respectively. In addition, the uncertainties of non-dimensional parameters are ±5% for Reynolds number, ±10% for Nusselt number and ±12% for friction.

5 Experimental results

The present experimental results of heat transfer rate and pressure drop in term of Nusselt number (*Nu*) and friction factor (*f*) of the plain tube are reported. The data are validated with the standard correlations [22, 23] for the fully developed turbulent flow, as depicted in Figs. 3a and 4a. In the figure, the cross symbols (×) represent the data predicted by developed correlations of the present work while the star symbols represent the ones predicted by the correlations of Dittus-Bolter [22]. As found the results of the present study are in good agreements with the standard correlations with error limits of ±2% for Nusselt number and ±2.9% for friction factor. The present plain tube work of Nusselt number and friction factor correlations can be written as follows:

$$Nu = 0.0225\,Re^{0.802}Pr^{0.4} \tag{6}$$

$$f = 0.512\,Re^{-0.3} \tag{7}$$

5.1 Heat transfer results

Results of heat transfer in term of Nusselt number in a heat exchanger equipped with PT inserts at various pitch ratios ($s/W = 0.4$, 0.6 and 0.8) and tape-twist ratios ($y/W = 3$, 4 and 5) for $d/W = 0.11$, are demonstrated in Fig. 3a, b. The results show that Nusselt number increases with decreasing the twist ratio and space ratio of tapes. This high heat transfer rate at small twist ratio and pitch ratio is attributed to a stronger swirl flow and higher turbulent intensity imparted to the flow. Thus, a more efficient disturbance of a thermal boundary layer is induced along the pipe wall, resulting in a thinner thermal boundary layer which is more favourable for heat transfer across the layer. In addition, the tape with twist ratio and space ratio prolongs resident time of fluid flow which is directly responsible for the increase of overall heat transfer. For the range studied, the twisted tape with pitch ratios (s/W) of 0.4, 0.6 and 0.8 respectively enhance heat transfer rate up to 77.5, 53 and 48% over that of the plain tube. The results on the effect of twist ratio (y/W) reveal that the Nusselt number provided by PT and TT with $y/W = 3$ are found to be around 10% and 17% higher than those given by the ones with $y/W = 4$ and 5.

The comparison between heat transfer augmentations offered by *PTs* and TTs is also presented in Fig. 3a, b. At similar conditions, PTs consistently provide higher Nusselt numbers than TTs. The superior heat transfer offered by PTs caused by an extra fluid turbulence around tube wall due to the presence of holes on the tapes, apart from a common swirl flow. Over the range investigated, PTs with $s/W = 0.4$, $d/W = 0.11$ and $y/W_c = 3$ give Nusselt number up to 25% higher than TTs at similar conditions (Reynolds number, inlet temperature and twist ratio, etc).

Effects of the PTs on the heat transfer enhancement with different perforation hole diameter ratios ($d/W = 0.11$, 0.14 and 0.17) at pitch ratios of $s/W = 0.4$, 0.6 and 0.8, are reported in Fig. 4a, b. The improvement of heat transfer

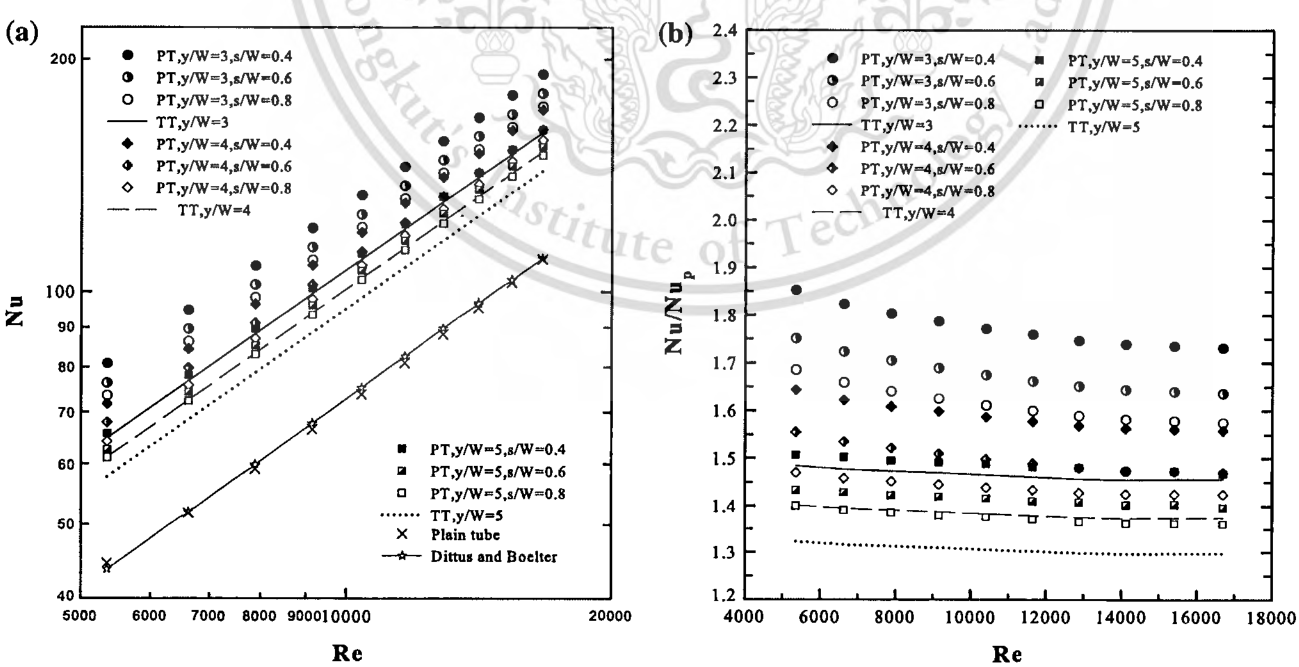


Fig. 3 **a** Effect of tape-twist ratio (*y/W*) and pitch ratio (*s/W*) on Nusselt number. **b** Effect of tape-twist ratio (*y/W*) and pitch ratio (*s/W*) on Nusselt number ratio

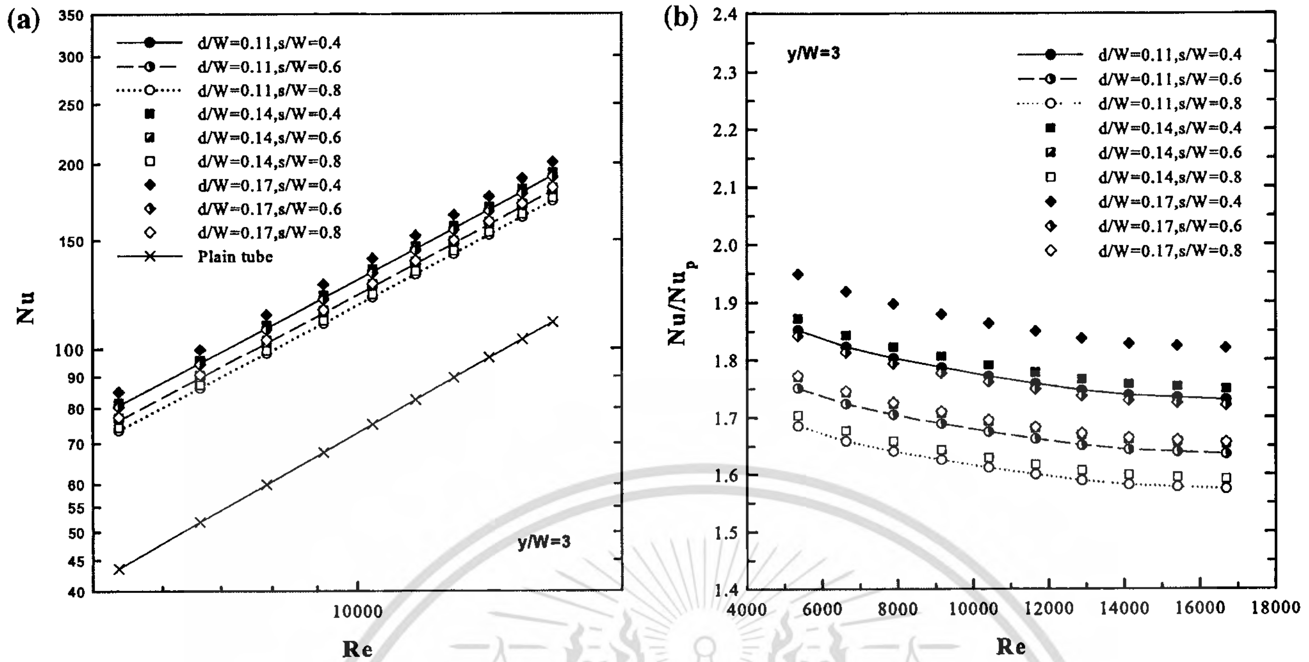


Fig. 4 **a** Effect of perforation hole diameter ratio (*d/W*) on Nusselt number. **b** Effect of perforation hole diameter ratio (*d/W*) on Nusselt number ratio

may also be caused eddy motion and the strong turbulence between the tube wall and perforated tape. It is also observed that the heat transfer enhancement increases with the increase of the perforation hole diameter. This can be explained by the fact that PTs with a larger perforation hole diameter induce higher turbulent intensity imparted to the flow between the wall tube and edge of the tapes. The Nusselt numbers obtained by the use of the tape with the largest perforation hole diameter ratio ($d/W = 0.17$), are 65.6–94.9% than those of the plain tube. It should be noted that heat transfer rate in term of Nusselt number for PTs with $d/W = 0.17$ are respectively, 4.8–6.1% and 2.1–5.1% higher than those for the PTs with $d/W = 0.11$ and 0.14, depending on pitch ratio (*s/W*).

5.2 Friction results

The variation of the pressure drop in terms of friction factor across the test with Reynolds number is shown in Fig. 5a, b. For all cases, friction factor decreases with increasing the Reynolds number. Similar to the results mentioned for Nusselt number, friction factor increases with decreasing the twist ratio and space ratio of tapes. At $d/W = 0$, friction factors generated by PT and TT with $y/W = 3$ are higher than those provided by the ones with twist ratios of $y/W = 4$ and 5 by 23 and 34%, respectively.

At similar operating conditions, PTs generate higher flow frictions than TTs. This is because of the extra dissipation of dynamic pressure of the fluid during act caused by the additional fluid disturbance due to the presence of holes on the tapes, resulting in an increase of interaction of the pressure force with inertial force around a velocity boundary layer. An average friction factor generated by PTs is found to be around 24% higher than that induced by TTs. In addition, the variation of the friction factor with Reynolds number for the PTs with different perforation hole diameter ratios (*d/W*) is demonstrated in Fig. 6a, b. At the given Reynolds number, the friction factors of PT with the largest perforation hole diameter ratio ($d/W = 0.17$) are found to be higher than that of the ones with $d/W = 0.11$ and 0.14 by around 15.5–24.3% and 7.8–10.4%, respectively, depending on pitch ratio.

5.3 Performance results

One method to evaluate potential of practical application of twisted tape is comparing convective heat transfer coefficient in a tube equipped with a twisted tape with that in a plain tube under the same pumping power.

For constant pumping power [7, 10, 16, 23]

$$(\dot{V}\Delta P)_p = (\dot{V}\Delta P)_t \tag{8}$$

and the relationship between friction and Reynolds number for both the plain tube and those with PT can be expressed as

$$(f\,Re^3)_p = (f\,Re^3)_t \tag{9}$$

The thermal performance factor (η) at constant pumping power is the ratio of the convective heat transfer coefficient

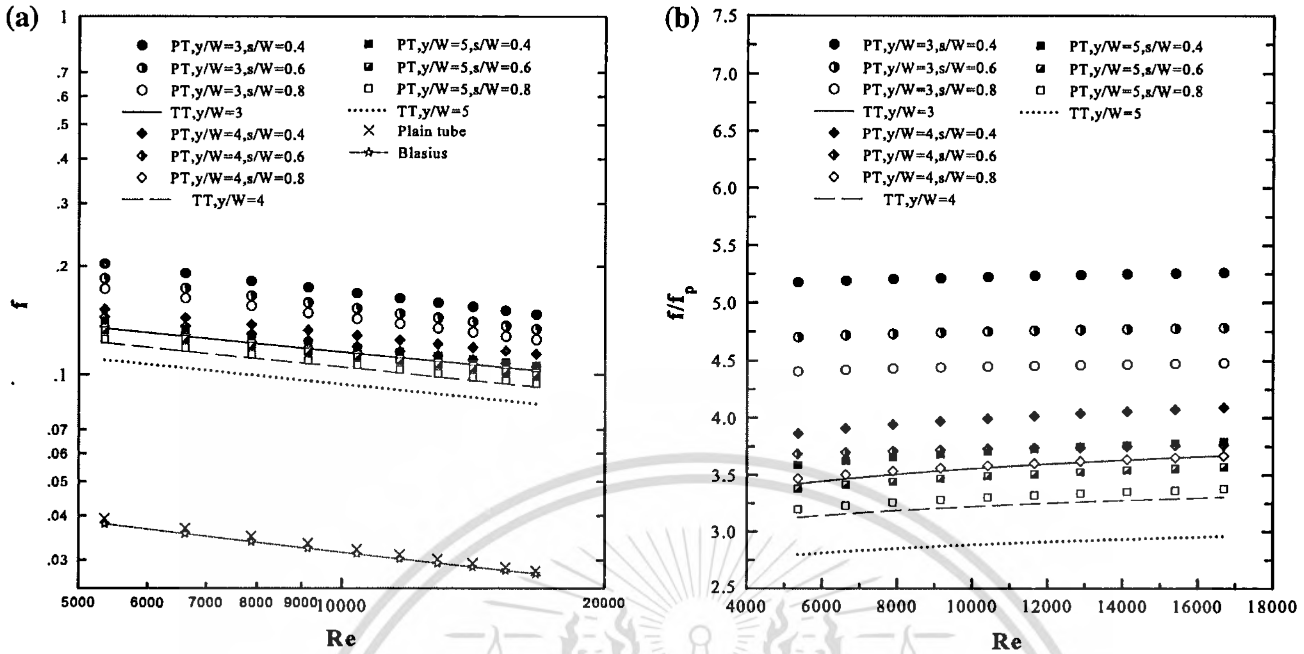


Fig. 5 **a** Effect of tape-twist ratio (*y/W*) and pitch ratio (*s/W*) on friction factor. **b** Effect of tape-twist ratio (*y/W*) and pitch ratio (*s/W*) on friction factor ratio

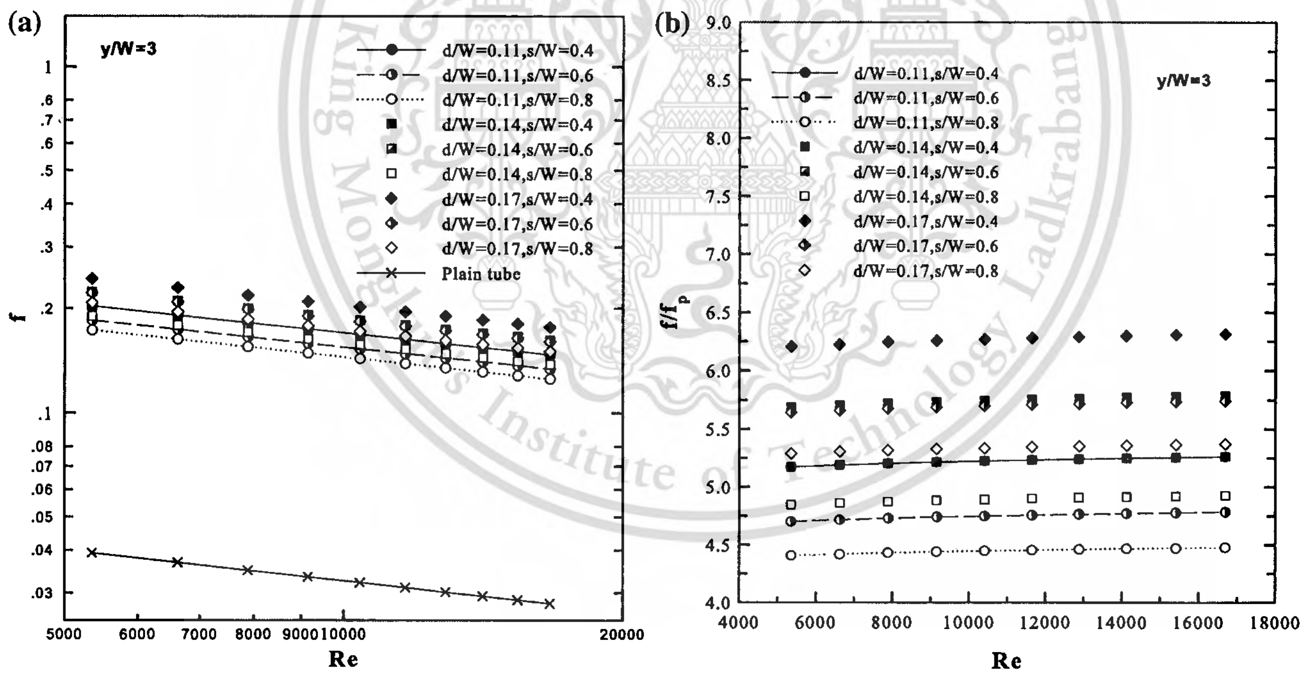


Fig. 6 **a** Effect of perforation hole diameter ratio (*d/W*) on friction factor. **b** Effect of perforation hole diameter ratio (*d/W*) on friction factor ratio

of the tube with PT to the plain tube which can be written as

$$\eta = \left.\frac{h_t}{h_p}\right|_{pp}, \tag{10}$$

The combination of Eqs. (8)–(10) yields

$$\eta = (Nu_t/Nu_p)(f_t/f_p)^{1/3} \tag{11}$$

The thermal performance factor above unity indicates an energy saving by using the tube with twisted tape relative to the plain tube. The variation between thermal performance factor and Reynolds number is presented in

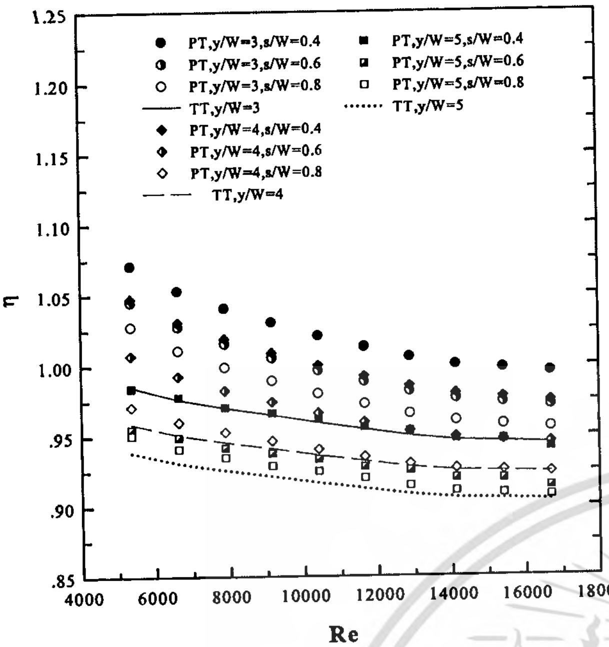


Fig. 7 Effect of tape-twist ratio (*y/W*) and perforated ratio (*s/W*) on thermal performance factor

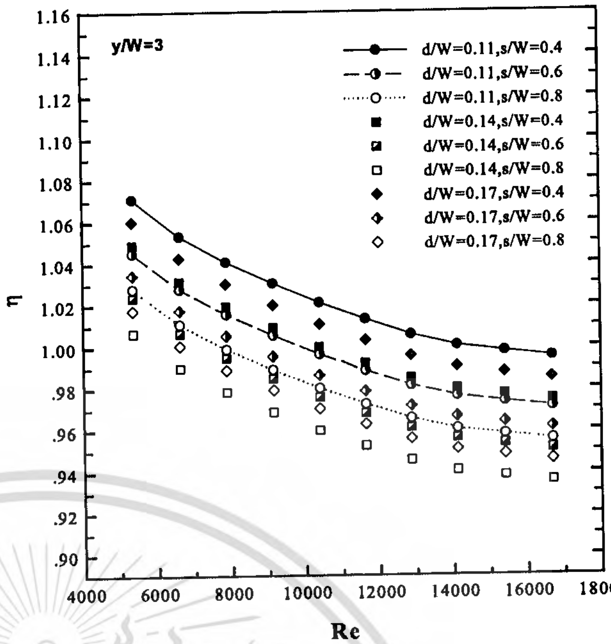


Fig. 8 Effect of perforation hole diameter ratio (*d/W*) on thermal performance factor

Fig. 7. In general, thermal performance factor decreases with increasing Reynolds numbers. The results also reveal that the tapes with smaller pitch ratio and twist ratio offer higher thermal performance factor. At similar conditions, the factor is varied between 0.94 and 1.07, 0.91 and 1.05, and 0.9 and 1.03 for *s/W* = 0.4, 0.6 and 0.8, respectively. Effect of the perforation hole diameter ratio (*d/W*) on the thermal performance factor is given in Fig. 8. Apparently, the thermal performance obtained by the use of the tape with a smaller perforation hole diameter ratio is considerably higher than that given by the one with a larger perforation hole diameter ratio for all Reynolds number value studied. This is principally influenced by a lower pressure drop penalty. The thermal performances obtained from the *PT* inserts with perforation hole diameter ratio of *d/W* = 0.11, are found to be around 1.7–2.1% and 0.8–1.3%, respectively, in comparison with those of the ones with *d/W* = 0.14 and 0.17. Although, the use of PT leads to a considerable increase in friction factor with respect to the use of TT, a reasonable increase in heat transfer rate results in slightly higher thermal performance. This indicates the beneficial gain by using PT over the use of TT in view point of energy saving.

The experimental results of the Nusselt number (*Nu*), friction factor (*f*) and thermal performance factor (η) for plain tube equipped with PT with three different tape-twist ratios (*y/W* = 3, 4 and 5), spaced-pitch ratios (*s/W* = 0.4, 0.6 and 0.8) and perforation hole diameter ratios (*d/W* = 0.11, 0.14 and 0.17) are fitted by the following empirical equations:

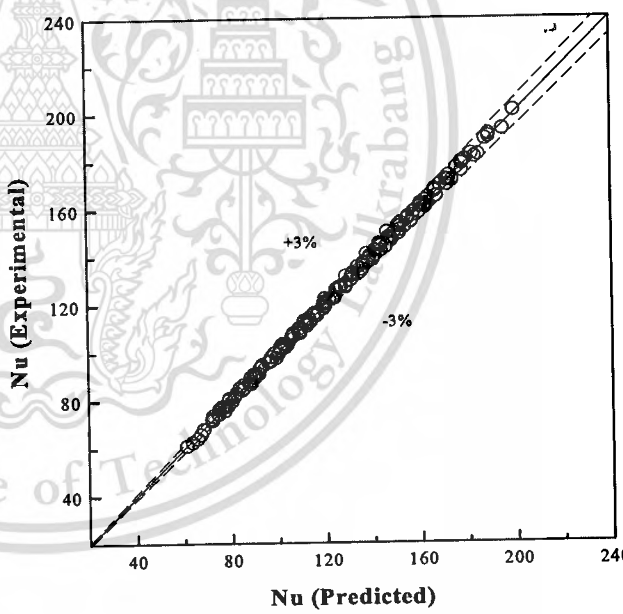


Fig. 9 Predicted Nusselt number versus experimental result

$$Nu = 0.09\,Re^{0.768}Pr^{0.4}(y/W)^{-0.325}(s/W)^{-0.133}(d/W)^{0.114} \quad (12)$$

$$f = 9.03\,Re^{-0.272}(y/W)^{-0.631}(s/W)^{-0.204}(d/W)^{0.428} \quad (13)$$

$$\eta = 1.764\,Re^{-0.059}(y/W)^{-0.114}(s/W)^{-0.065}(d/W)^{-0.028} \quad (14)$$

Comparisons between the Nusselt number and the friction factor obtained from the present data with those

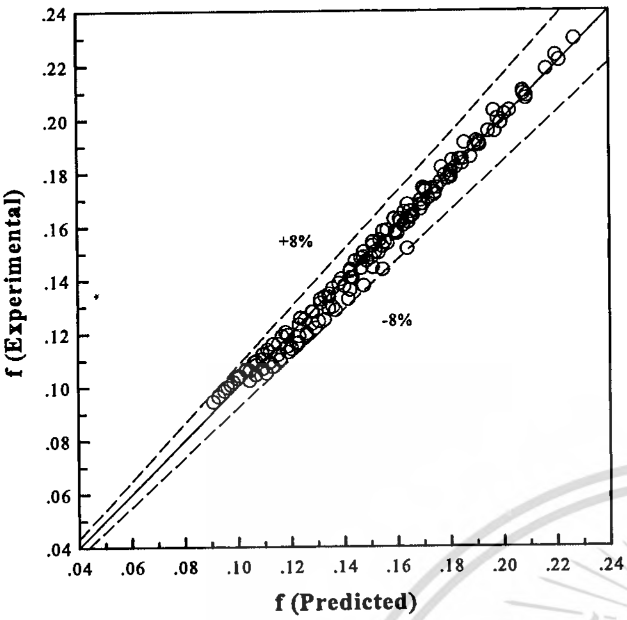


Fig. 10 Predicted friction factor versus experimental result

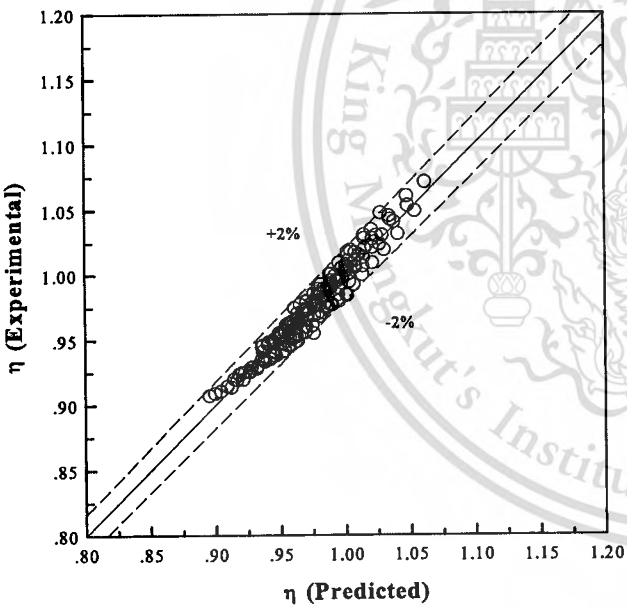


Fig. 11 Predicted thermal performance factor versus experimental result

calculated by the present correlations are portrayed in Figs. 9, 10 and 11. As found, the Eqs. (12)–(14) are fitting the experimental data within ±3, ±8 and ±2% for Nusselt number, friction factor and thermal performance, respectively.

5.4 Comparison with previous works

Some results obtained by the used of other modified twisted tapes in the earlier works including centrally perforated/jagged/notch twisted tape [7], short-length twisted tape [5] and serrated twisted tape [24], are selected for comparing with the results of the present work (the results of the tape with PT at $d/W = 0.11$ and $s/W = 0.4$). All tape geometries and test conditions are summarized in Table 1. For all cases, thermal performance factor tends decrease with the rise of Reynolds number. Evidently, the centrally PT [7] with a large hole and the short-length twisted tape [5] provided considerably lower thermal performance than other tapes, due to the suppressed swirl around the core tape [7] and swirl decaying [5]. On the other hand, the serrated tape [24] yielded high heat transfer rate due to an extra turbulence induced by the rib. However, the promoted heat transfer accompanied by significant pressure loss which brings thermal performance down under unity. The thermal performances of PT at $d/W = 0.11$ and $s/W = 0.4$ are found to be comparable with those of the jagged/notch twisted tape [7]. This may be responsible by the similar flow behaviour (especially that near tube wall) induced by these two modified tapes. A slightly lower thermal performance factor of the present modified tape is caused by a lower heat transfer rate as a result of weaker turbulence intensity (Fig. 12).

6 Conclusions

The present study explores the effect of PTs on the heat transfer, friction factor and thermal performance at different tape-twist ratios (y/W) and pitch ratios (s/W). It is found that PTs cause a considerable increase in heat transfer rate and friction factor in comparison with the TT. The increase in heat transfer rate by using PT is found to be in range of

Table 1 Details of previous studies

Author	Year	Reynolds number	Tape geometry
Eiamsa-ard et al. [5]	2009	5,000 to 15,000	Short length tape with $y/W = 3$
Rahimi et al. [7]	2009	2,950 to 11,800	Jagged tape with $y/W = 2.94$
Rahimi et al. [7]	2009	2,950 to 11,800	Notch tape with $y/W = 2.94$
Rahimi et al. [7]	2009	2,950 to 11,800	Perforated tape with $y/W = 2.94$
Chang et al. [24]	2007	5,000 to 25,000	Serrated tape with $y/W = 2.81$

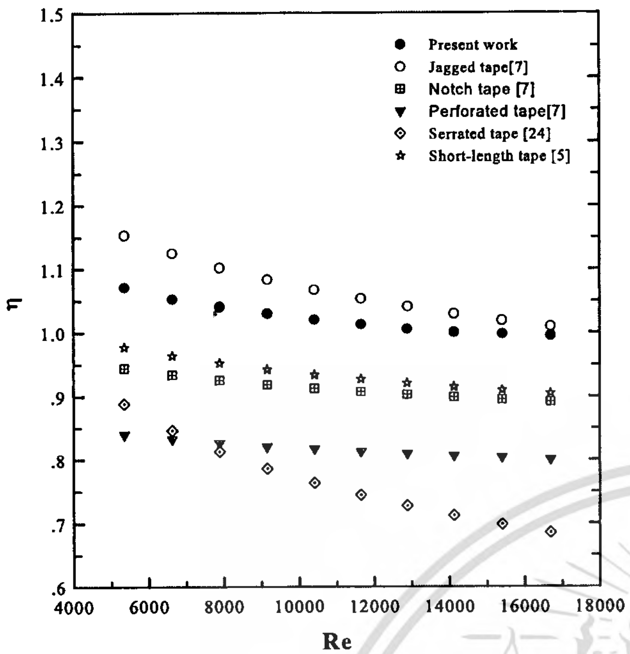


Fig. 12 Comparison of thermal performance factor between present work and previous works

36–85%, over those of the corresponding plain tube. Heat transfer rate increases with decreasing tape-twist ratio and pitch ratio. The average heat transfer and friction factors generated by PTs are approximately 25 and 24% higher than those the given by TTs. Thermal performance evaluation analysis shows that the maximum thermal performance factor of 1.07 is offered by the PT with the smallest tape-twist ratio ($y/W = 3$), perforation hole diameter ratio ($d/W = 0.11$) and pitch ratio ($s/W = 0.4$). It can be observed that the thermal performance factors obtained by the uses of the modified tapes proposed in the present work (PT) are mostly below unity, especially at high Reynolds number. The results are caused by the dominant effect of the substantial increase in pressure drop over that of the considerable improvement in heat transfer. Therefore, the modified tapes are not practically feasible in terms of energy saving for general cases. However, they can be applied at very low Reynolds number or in places where pumping power is not important. It should also be stated that the geometry of twisted tape insert is progressively developed, the twisted tape with appropriate geometry may lead to an overall energy gain because of a more efficient heat transfer and a lower pressure drop. The empirical correlations developed relating tape-twist ratio (y/W), spaced-pitch ratios (s/W), perforation hole diameter ratio (d/W) and Reynolds number, are fitting the experimental data within ±3, ±8 and ±2% for Nusselt number, friction factor and thermal performance factor, respectively.

Acknowledgments The authors would like to acknowledge with appreciation, the Energy Policy and Planning Office, Ministry of Energy, Thailand (EPPO) for financial support of this research.

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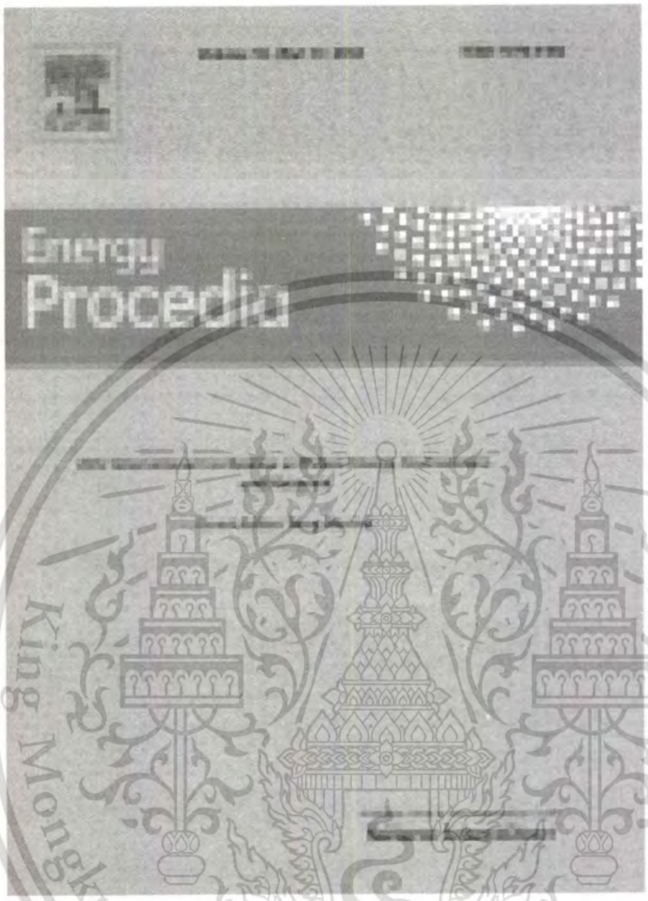
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Energy Procedia 14 (2012) 1117 – 1123

Energy Procedia

2nd International Conference on Advances in Energy Engineering (ICAEE 2011)

Effect of perforated twisted-tapes with parallel wings on heat transfer enhancement in a heat exchanger tube

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Abstract

This article reports an experimental investigation on heat transfer and pressure drop characteristics of turbulent flow in a heating tube equipped with perforated twisted tapes with parallel wings (PTT) for Reynolds number between 5500 and 20500. The design of PTT involves the following concepts: (1) wings induce an extra turbulence near tube wall and thus efficiently disrupt a thermal boundary layer (2) holes existing along a core tube, diminish pressure loss within the tube. The parameters investigated were the hole diameter ratio (d/W = 0.11, 0.33 and 0.55) and wing depth ratio (w/W = 0.11, 0.22 and 0.33). A typical twisted tape was also tested for an assessment. Compared to the plain tube, the tubes with PTT and TT yielded heat transfer enhancement up to 208% and 190%, respectively. The evaluation of overall performance under the same pumping power reveal that the PTT with d/W = 0.11 and w/W = 0.33, gave the maximum thermal performance factor of 1.32, at Reynolds number of 5500. Empirical correlations of the heat transfer, friction factor and thermal performance for tubes with PTTs were also developed. In addition, the swirling/axial flow patterns of tube with PTT were visualized using dye injection technique.

Keywords: Heat transfer enhancement; heat exchanger; twisted tape; thermal performance

1. Introduction

Heat transfer enhancement techniques are widely used in many engineering applications for example heat recovery process, shell-and-tube heat transfer exchanger, air conditioning and refrigeration systems, nuclear energy industry, chemical reactors, high power laser systems and chemical process plants, etc [1-2].

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876-6102 © 2011 Published by Elsevier Ltd. Selection and/or peer-review under responsibility of the organizing committee of 2nd International Conference on Advances in Energy Engineering (ICAEE).
doi:10.1016/j.egypro.2011.12.887

The most significant variables in reducing the size and cost of a heat exchanger are basically the heat transfer coefficient and pressure drop. An increase in the heat transfer coefficient generally leads to another advantage of reducing the temperature driving force, which increases the second law efficiency and decreases entropy generation. Several enhancement of heat transfer devices have been introduced and improved for increasing the heat transfer rate and thermal performance in heat exchangers by both of active and passive methods. Active method is the approach that needs the extra power source for example mechanical aids, surface-fluid vibration, injection and suction of the fluid, jet impingement, and use of electrostatic fields. On the other hand, passive method does not require an extra power source. The devices in this category are surface coating, rough surfaces, extended surfaces, turbulent/swirl flow devices, tube insert (wire, spiral spring, porous, static mixer, twisted tape, louvered strip, miniature hydraulic turbine, mesh, and internal fin inserts), convoluted (twisted) tube, additives for liquid and gases. Application of twisted tape in heat exchanger tube [3-5] is the one of key heat transfer enhancement techniques. Convective heat transfer assisted by modified twisted tapes with different geometries have been extensively investigated such as twin twisted tapes, dual twisted tape elements in tandem, alternate clockwise and counter-clockwise twisted-tape, twisted tape consisting of centre wings and alternate-axes, peripherally-cut twisted tape and serrated twisted-tape. As compared to a typical twisted tape, heat transfer enhancement by the modified twisted tapes with such geometries is achieved by extra turbulence and thus better fluid mixing. However, an improvement in heat transfer rate is generally achieved at an expense of substantially increased friction. The most desired twisted tape is the one yields excellent heat transfer with minimum increase in friction.

To deal with an antagonistic requirement as mentioned above, the newly design twisted tape called perforated twisted tape with parallel wings (PTT) is proposed. The design of PTT involves the following concepts: (1) wings induce an extra turbulence near tube wall and thus efficiently disrupt a thermal boundary layer (2) holes existing along a core tube, diminish pressure loss within the tube. To evaluate practical applications, the overall energy performance in term of thermal performance factors under the same pumping power is evaluated. The investigation was performed for fully developed flow in turbulence regime ($5500 \leq Re \leq 20500$). Empirical correlations for heat transfer, friction factor and thermal performance factor are also reported.

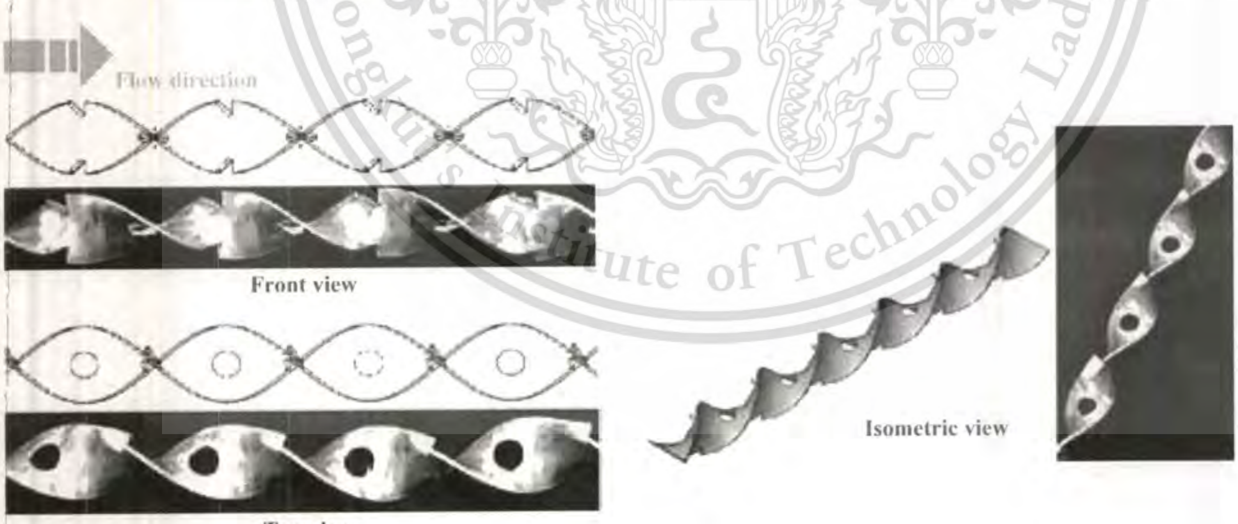

Fig. 1. Photograph and sketch of PTT with parallel wings

2. Perforated twisted tape with parallel wing

The tapes were made from the aluminum strip sheet having width (W) of 18 mm and thickness (δ) of 1.0 mm. All tapes were twisted at constant twist length of y = 56 mm which corresponds to twist ratio (y/W) = 3 to formulate typical twisted tapes (TTs). Then, TTs were modified by periodic generating holes along a core tube in straight line at every pitch length (180°/twist length). Furthermore, each tape was cut at the edge on both sides between adjacent holes, each cut was subsequently arranged in 45° to the axial flow in the same direction, so called parallel-wings the tap. The modified twisted tape (PTT) is depicted in Fig. 1. The parameters investigated were the hole diameter ratio (d/W = 0.11, 0.33 and 0.55) and wing depth ratio (w/W = 0.11, 0.22 and 0.33). The typical twisted tape (TT) was also subjected to the test, for comparison. All of the tapes were inserted at the core tube along the test section. More details of the twisted tape and experimental set-up can be found in the previous work given by Eiamsa-ard *et al.* [5].

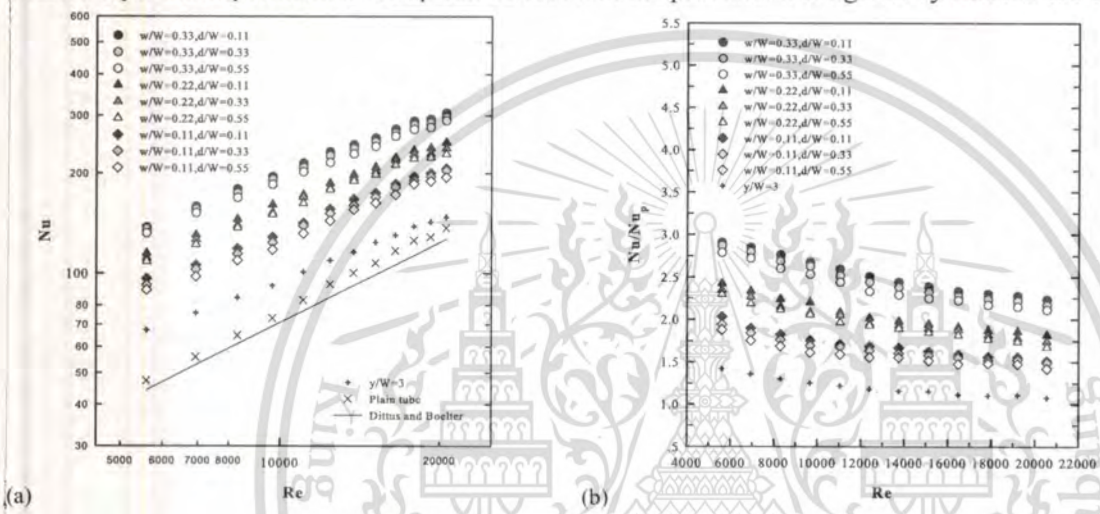

Fig. 2. Variations of (a) Nusselt number (b) Nu/Nu_p with Reynolds number

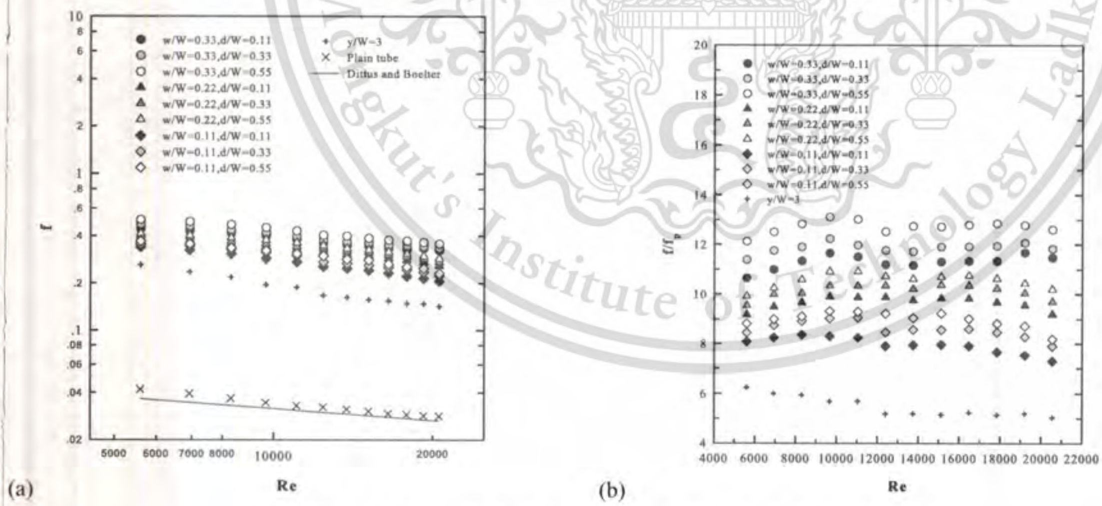

Fig. 3. Variations of (a) friction factor (b) f/f_p with Reynolds number

3. Results and discussion

The experimental investigations of heat transfer, friction factor and thermal performance behaviors in a heat exchanger tube fitted with the PTT with parallel wings of varying wing-cut ratio (*w/W*) and hole diameter ratio (*d/W*) are described. Prior to the main experiment, the plain tube was tested and validated. Figures 2 and 3 show the validations of the present experimental results and with those available in the earlier works. As shown, the present friction factors agree well with those obtained from Blasius equation within ±8.5% and the present Nusselt numbers are within ±8% of those calculated from Dittus-Boelter equation. The present results of the plain tube are therefore used as the bases for the evaluation of the effects of twisted tape on heat transfer enhancement.

3.1. Flow visualization

Since it is difficult to distinct the main flow from the secondary flows induces by TT and PTT via the dye visualization technique in turbulent regime, thus the visualization was performed in laminar regime instead, only for qualitative comparison. Figure 4 shows the flow visualization of the flows through tubes with and without twisted tape. Only axial flow was observed in the plain tube (Fig. 4a) while common swirl flow was found with the presence of TT (Fig. 4b). In case of PTT (Fig. 4c), apart from a common swirl flow, there was attack of dye streams on the wings of PTT leading to an extra turbulence as well as collision among dye streams and thus superior fluid mixing to the case of TT. It should be mentioned that the dye stream positioned around the mid of the tape was directed through a hole to another side of the tape, and the stream direction was between those of an axial flow and a swirl flow.

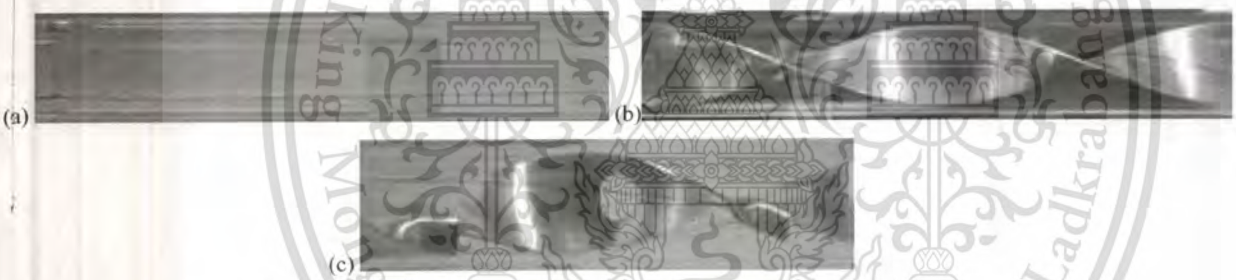

Fig. 4. Flow visualization of flows through (a) plain tube, (b) TT and (c) PTT

3.2. Effect of the presence of wing and wing depth ratio (w/W)

The experimental results of the tubes equipped with PTTs and TT as well as the plain tube obtained under turbulent flow conditions presented in Fig. 2(a-b). Influence of the presence of wing on heat transfer was significant, notified by the considerably higher Nusselt number given by PTT compared to those provided by TT. The superior heat transfer is responsible by the induction of extra turbulent flows near the tube wall as detected by the dye visualization (Fig. 4b). This efficiently disturbs a thermal boundary layer by generating periodic disruption of the viscous boundary layer. At similar operating conditions, it was found that Nusselt number increased with increasing wing depth ratio *(w/W)*. It can be explain by the fact that the larger depth ratio causes higher turbulence intensity and thus better mixing fluid near the tube wall. For the tape with the largest depth ratio (w/W = 0.33), the increase in heat transfer rate was up to 49% and 23% over those of the ones with w/W = 0.11 and w/W = 0.22, respectively. Figure 3(a-b) shows the variation of the friction factor with Reynolds number. Obviously, the use of the tape with larger wing depth ratio generated higher friction factor. This is directly related to the higher turbulence intensity as mentioned above. As found the tape with the largest depth ratio (w/W = 0.33) yielded 40.8% and 18.3% higher mean friction factor than the ones with w/W = 0.11 and w/W = 0.22, respectively. The result of thermal performance factor in which both heat transfer and friction are taken into account based in the same pumping power, is illustrated in Fig. 5. It is found that PTTs consistently gave higher thermal performance factor than TT, at similar conditions. Among PTTs, thermal performance factor increased with increasing wing depth ratio. Thermal performances varied between 0.71 and 1.01, 0.77 and 1.16, and 0.91 and 1.32 for the PTTs with w/W = 0.11, 0.22 and 0.33, respectively. It noteworthy that thermal performance was higher at lower Reynolds number, this implies that twisted tapes are more suitable for practical application at lower Reynolds number.

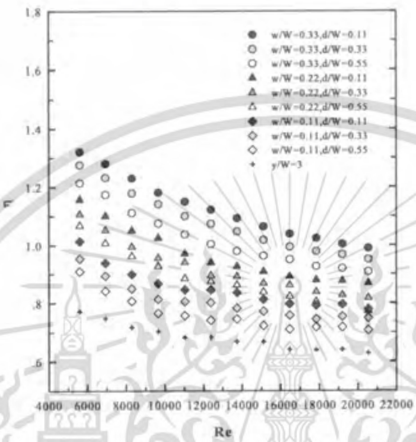

Fig. 5. Variation of thermal performance factor with Reynolds number

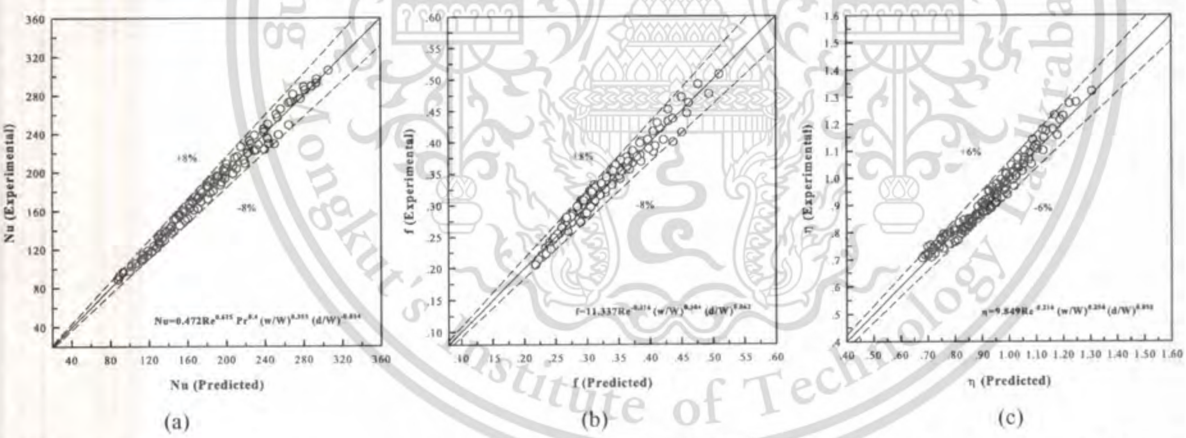

Fig. 6. Comparison between the predicted and experimental data of (a) Nu, (b) friction factor and (c) thermal performance factor

3.3. Effect of perforated diameter ratio (d/W)

The effect of perforated diameter ratio (d/W = 0.11, 0.33 and 0.55) on the heat transfer rate is shown in Fig. 2(a-b). Apparently, Nusselt number increased with the decrease of diameter ratio. This supports the finding by the dye visualization (Fig.4c) that the stream directed through a hole behaved between an axial flow and a swirl flow, with the larger hole diameter, the flow is assumed to be more similar to the axial flow leading to the loss of swirl intensity imparted to the flow between tape and surface wall. Heat transfer rate decreased up to 2.8% and 6% by the uses of the tapes with d/W = 0.33 for d/W = 0.55 compared to that of the tape with d/W = 0.11. By the same reason, friction generated by PTTs decreased with the increase diameter ratio *(d/W)* as presented in Fig. 3(a-b). The tapes with d/W = 0.33 for d/W = 0.55 yielded 6% and 11.6% lower mean friction factor than the one with d/W = 0.11. Effect of the perforated diameter ratio (*d/W*) on the thermal performance factor in a heat exchanger tube fitted with PTT is presented in Fig. 5. The thermal performances from using PTT with smaller hole diameter ratio (*d/W*) were observed to be higher than that those achieved from the tape with larger *d/W*. This signifies the dominant effect of increased heat transfer over that of increased friction factor as hole diameter decreases. It was found that the tape with the smallest diameter ratio (d/W = 0.11) provided higher thermal performance factor than the ones with d/W = 0.33 and 0.55 by around 4.9% and 10.4%, respectively. Note that for the present range, thermal performances achieved by using all tape inserts are above unity (low *Re*), indicating the economic benefit due to the heat transfer enhancement. The experimental results of Nusselt number, friction factor and thermal performance were fitted, using least square regression analysis, in which a wing-cut ratio (*w/W*) and a hole diameter ratio (*d/W*) were taken into account. The predicted data from the correlations of the Nu_{pred}, f_{pred} and η_{pred} are plotted against experimental data of the Nu_{exp}, f_{exp} and η_{exp} in Fig. 6(a-c). As shown from these figures the maximum deviations between the experimental data and correlations are ±8%, ±8%, and ±6%, respectively.

4. Conclusions

Augmentation of heat transfer rate in heat exchanger tubes by means of perforated twisted tapes (PTT) inserts is investigated experimentally. The results showed those heat transfer and friction factors were significantly influenced by the presences of wings and holes on PTTs. Both heat transfer and friction increased with the increase of wing depth ratio (*w/W*) and the decrease of perforation hole diameter ratio (*d/W*). Due to the dominant effect of increased heat transfer over that of increased friction factor, the thermal performance factor was found to be increased as wing depth ratio (*w/W*) increased and hole diameter ratio (*d/W*) decreased.

Acknowledgements

The authors would like to acknowledge with appreciation, the Energy Policy and Planning Office, Ministry of Energy, Thailand (EPPO) for financial support of this research.

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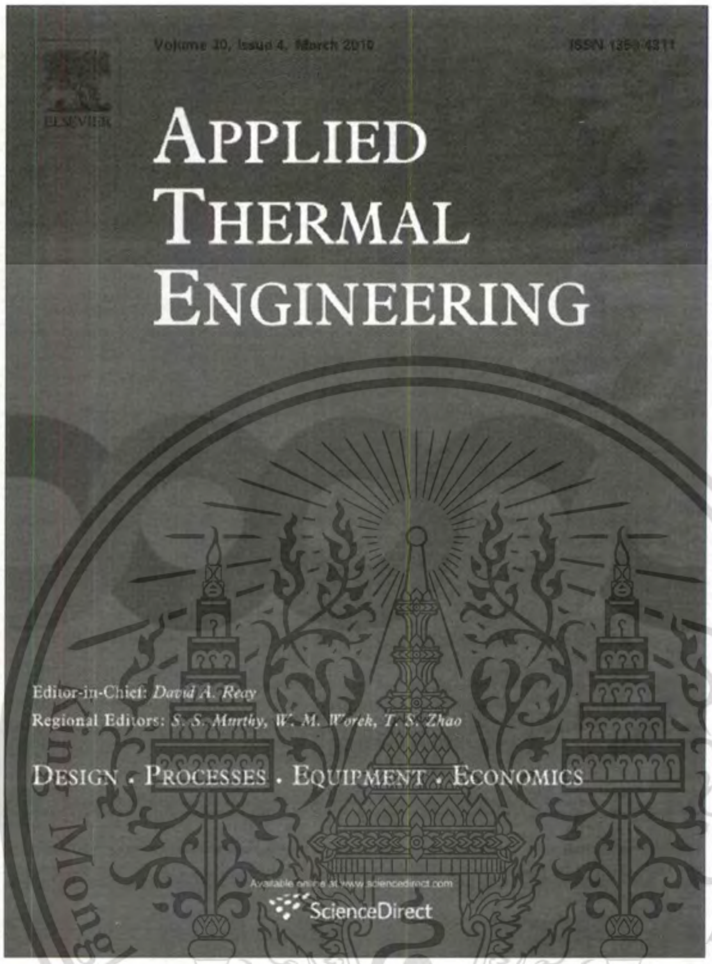
Volume 30, Issue 4, March 2010
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Applied Thermal Engineering 30 (2010) 310–318

Contents lists available at ScienceDirect

Applied Thermal Engineering

journal homepage: www.elsevier.com/locate/apthermeng

Heat transfer enhancement in a tube using delta-winglet twisted tape inserts

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ARTICLE INFO

Article history:
Received 20 March 2009
Accepted 5 September 2009
Available online 10 September 2009

Keywords:
Heat transfer enhancement
Friction factor
Delta-winglet twisted tape
Swirl flow

ABSTRACT

Heat transfer, flow friction and thermal performance factor characteristics in a tube fitted with delta-winglet twisted tape, using water as working fluid are investigated experimentally. Influences of the oblique delta-winglet twisted tape (*O-DWT*) and straight delta-winglet twisted tape (*S-DWT*) arrangements are also described. The experiments are conducted using the tapes with three twist ratios ($y/w = 3$, 4 and 5) and three depth of wing cut ratios ($DR = d/w = 0.11$, 0.21 and 0.32) over a Reynolds number range of 3000–27,000 in a uniform wall heat flux tube. The obtained results show that mean Nusselt number and mean friction factor in the tube with the delta-winglet twisted tape increase with decreasing twisted ratio (y/w) and increasing depth of wing cut ratio (DR). It is also observed that the *O-DWT* is more effective turbulator giving higher heat transfer coefficient than the *S-DWT*. Over the range considered, Nusselt number, friction factor and thermal performance factor in a tube with the *O-DWT* are, respectively, 1.04–1.64, 1.09–1.95, and 1.05–1.13 times of those in the tube with typical twisted tape (*TT*). Empirical correlations for predicting Nusselt number and friction factor have been employed. The predicted data are within ±10% for Nusselt number and ±10% for friction factor.

1. Introduction

High performance heat transfer system is of great importance in many industrial applications. Therefore, the heat transfer enhancement techniques are widely applied in heat exchangers, in order to improve heat transfer coefficient [1–4]. Passive heat transfer augmentation is method to enhance heat transfer without external power. Among the techniques used, insertion of twisted tape in a circular tube is one of the most effective approaches. The inserted twisted tape generates swirling flow and increases turbulence intensity which is major influencing factors for heat transfer enhancement. In fact, using twisted tape increases both desirable heat transfer rate and undesirable friction loss (pressure drop). An appropriate twisted tape modification is a challenge task as a proper design of twisted tape is a main key for heat transfer enhancement at a reasonable friction loss.

Numerous investigations using twisted tape with different geometrics for heat transfer enhancement have been released. Dewan et al. [5] overviewed vast numbers of research works of heat transfer enhancement using this type of the insert. Here, some of the literatures after the review paper are mentioned as follows. In the earlier papers of our research group, heat transfer and friction factor characteristics of a flow through the circular tube fitted with twisted tapes with various free space ratios [6], a conical-ring combined with a twisted tape insert [7], and helical tapes with and without a rod [8], were investigated. The results revealed that both heat transfer coefficient and friction factor increased with a decrease of the free space ratio [6], a tube fitted with the conical-ring and a twisted tape gave higher Nusselt number than that fitted with a simple conical-ring [7], and helical tape with a rod provided a higher heat transfer rate in comparison with the helical tape without a rod [8]. Promvonge [9] used wire coil in conjunction with twisted tape for heat transfer augmentation. The finding was that the combination of wire coil and twisted tape led to a double increase in heat transfer over the use of wire coil/twisted tape alone. It was also observed that, the twisted tape and wire coil with smaller twist and coil pitch ratio gave higher heat transfer rate than those with larger twist and coil pitch ratios. Chang et al. [10,11] investigated the heat transfer enhancement in the tubes fitted with serrated twisted tape and broken twisted tape at different twist ratios. The results of both papers showed similar trends, that the local Nusselt number and Fanning friction factor increased as the twist ratio decreased. Recently, Rahimi et al. [12] reported experimental and computational fluid dynamics (CFD) investigation on the friction factor, Nusselt number and thermal–hydraulic performance of a tube equipped with the typical and three modified twisted tapes (perforated, notched, and jagged twisted tape). Their results demonstrated that the Nusselt number

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1359-4311/$ - see front matter
doi:10.1016/j.applthermaleng.2009.09.006

Nomenclature

A	heat transfer surface area, m^2
C_p	specific heat of fluid, $J\,kg^{-1}\,K^{-1}$
D	inside diameter of the test tube, m
d	depth of wing cut, m
f	friction factor = $\Delta P/((L/D)(\rho U^2/2))$
h	heat transfer coefficient, $W\,m^{-2}\,K^{-1}$
I	current, A
k	thermal conductivity of fluid, $W\,m^{-1}\,K^{-1}$
L	length of the test section, m
M	mass flow rate, $kg\,s^{-1}$
Nu	Nusselt number = hD/k
P	pressure of flow in stationary tube, Pa
ΔP	pressure drop, Pa
Pr	Prandtl number = $\mu C_p/k$
Q	heat transfer rate, W
Re	Reynolds number = $\rho UD/\mu$
t	thickness of the test tube, m
T	temperature, K
$\tilde{T}$	mean temperature, K
U	average axial flow velocity, $m\,s^{-1}$
V	voltage, V
$\dot{V}$	volumetric flow rate, $m^{-3}\,s$
w	tape width, m
y	twisted tape pitch, m

Greek symbols

ρ	fluid density, $kg\,m^{-3}$
δ	twisted tape thickness, m
μ	fluid dynamic viscosity, $kg\,s^{-1}\,m^{-1}$
η	thermal performance factor

Subscripts

b	bulk
c	convection
i	inlet
o	outlet
p	plain
pp	pumping power
s	surface
t	turbulator
w	water

Abbreviations

O-DWT	oblique delta-winglet twisted tape
S-DWT	straight delta-winglet twisted tape
TT	typical twisted tape

and thermal performance of the jagged insert were higher than those of the others.

In the current work, the newly designed *DWTs* are used as heat transfer enhancement devices. It is assumed that, the wing part can enhance degree of turbulence near wall tube and thus, heat transfer rate. The experiments are performed using the tape with three different twist ratios (y/w = 3, 4 and 5) and three depth of wing cut ratios ($DR = d/w$ = 0.11, 0.21 and 0.32) with a Reynolds number range between 3000 and 27,000, where water is used as the working fluid. The effects of the oblique delta-winglet twisted tape (*O-DWT*) and straight delta-winglet twisted tape (*S-DWT*) arrangements on the heat transfer rate, friction factor and thermal enhancement factor characteristics in a round tube are also reported in this paper.

2. Delta-winglet twisted tape

All tapes used in the present work are made of aluminum strip with 0.8 mm thickness (δ) and 19.5 mm width (w). Firstly, aluminum strip was twisted to produce a typical twisted tape. A tape was subsequently modified to obtain the *DWT* by cutting at the edge of the tape with oblique shape and straight shape to produce an oblique delta-winglet twisted tape (*O-DWT*) and a straight delta-winglet twisted tape (*S-DWT*), respectively. Then the outer part of the cut was arranged to 90° (*degree*) relatively to the inner part, forming delta-winglet shape. The length between the two cuts was set equally to the pitch length (y). The *DWTs* were prepared with three different twist ratios (y/w) = 3, 4 and 5, and three different depth of wing cut ratios ($DR = d/w$ = 0.11, 0.21 and 0.32). The geometrical configurations of the delta-winglet twisted tape (*DWT*) inserts are presented in Fig. 1.

3. Experimental details

3.1. Experimental setup

Experiments were carried out in an open loop rig as shown in Fig. 2, using water as working fluid. The heating test tube is made of copper with thickness of 1.5 mm, diameter of 19.5 mm and length of 1000 mm. Cold water was continuously supplied from a water chiller (0.5 m^3) to an overhead water tank (at 3 m elevation) by 0.5 hp centrifugal water pump. The overhead water tank had a volume of 0.5 m^3. The constant head water level in overhead water tank was controlled by overflow discharged through the ball valve to the ground water tank (0.5 m^3). The water main line was 25.4 mm in diameter and the dimension then decreased to 19.5 mm at calming section. Calming section made of copper tube with length of 1500 mm was used to eliminate the entrance effect. The rubber bellows were also used to reduce the flow pulsations and vibration effect from the centrifugal water pump. One end of heating test tube was attached with the calming section, while the other end was attached with the mixing section where three baffles were assembled to the tube at a position of 150 mm ahead of the test tube exit, for efficient mixing of outgoing fluid.

The test tube was heated by continually winding flexible electrical wire to provide a uniform heat flux boundary condition. The heating tube was wound with ceramic beads coated electrical SWG Nichrome heating wire. The terminals of the Nichrome wire were connected to the Varaic transformer, by which heat flux of the tube wall was varied by adjusting the voltage (2–200 V) and current (9–15 A). The electrical output power was controlled via a Variac transformer to obtain a uniform heat flux (UHF) along the entire length of the test section. The K-type thermocouple beads were tapped along the local tube wall for 15 stations for monitoring the temperature of the surface tube wall. The heating tube and thermocouple were covered with insulation to minimize heat loss to surrounding. Calibrated RTD PT 100 type temperature sensors were used to measure the inlet (before the test section) and outgoing water temperature (behind the mixing section) of the heating test tube. At the entrance of the test tube, the volume flow rate of water was measured using rotameter. The water flow rate was adjusted by the globe valve which was located upstream of the rotameter. The experiments were carried out at the Reynolds number ranged between 3000 and 27,000. For each test run, temperature, volumetric flow rate and pressure drop of the bulk air were recorded at steady state conditions, in which the inlet water

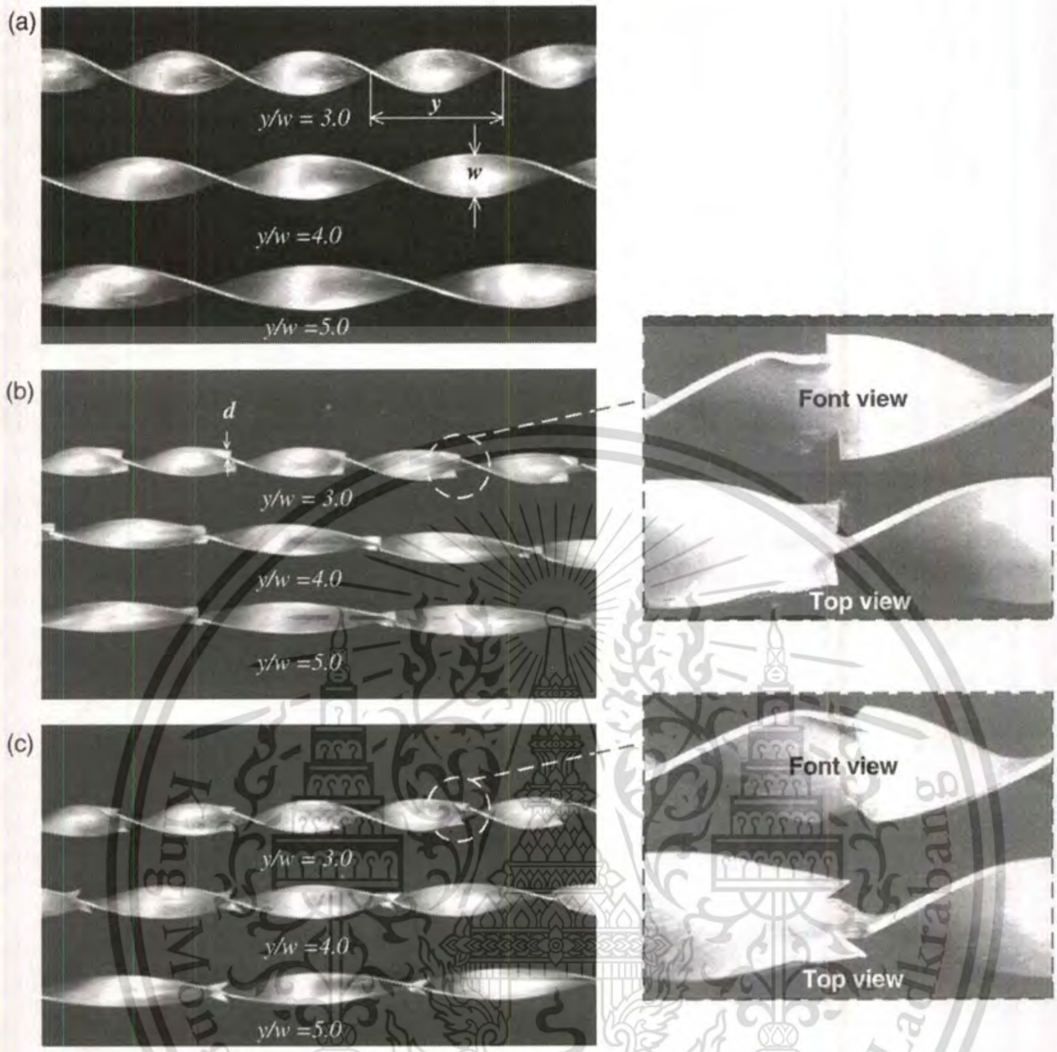

Fig. 1. Twisted tape vortex generator: (a) typical twisted tape (*TT*), (b) straight delta-winglet twisted tapes (*S-DWT*) and (c) oblique delta-winglet twisted tapes (*O-DWT*).

temperature was maintained at 27 °C and the volumetric flow rates were varied from 120 l/h to 1200 l/h.

3.2. Experimental procedure

In the experiments, water was pumped to the overhead water tank then discharged directly to the test tube. The heat flux at the test tube wall was set using Variac transformer and then, the system was allowed to reach steady state for each value of Reynolds number. The friction factor was calculated in term of pressure drop that measured by manometer under isothermal condition or without heating tube. In the experiment, the pressure taps were located around 50 mm upstream and 150 mm downstream from the test section. The length between both tapes was around 1200 mm. In the experiments, the tape was used as swirl generator to produce different swirl/turbulent intensity depends on the twist ratios ($y/w = 3$, 4 and 5), depth of wing cut ratios ($DR = d/w = 0.11$, 0.21 and 0.32) and Reynolds number. Effects of the oblique delta-winglet twisted tape (*O-DWT*) and straight delta-winglet twisted tape (*S-DWT*) arrangements are also examined.

4. Calculation of heat transfer and friction factor

The details of the calculation of heat transfer in term of Nusselt number and friction factor of the present study are described as follows. Heat transferred to the cold water is assumed to be equal to the heat loss from the test section which can be expressed as:

$$Q_w = Q_c \tag{1}$$

where

$$Q_w = MC_{p,w}(T_o - T_i) \tag{2}$$

and the heat balance between the heat input (Q_{VI}) and heat transfer rate of the water (Q_w) is less than 7%

$$(Q_{VI} - Q_w) \times 100\%/Q_{VI} < 7\% \tag{3}$$

The convection heat transfer from the test section can be written by

$$Q_c = hA(\tilde{T}_s - T_b) \tag{4}$$

where A is the internal surface of the tube wall (πDL) and T_b is the bulk/mean fluid (water) temperature ($T_b = (T_o + T_i)/2$) and $\tilde{T}_s$ is the mean wall temperature and evaluated at the outer wall surface of the test tube which can be written as

$$\tilde{T}_s = \sum T_s/15 \tag{5}$$

The mean wall temperature is calculated from 15 points of surface temperatures lined between the inlet and the exit of the test tube. The average heat transfer coefficient, h and the mean Nusselt number, Nu are estimated as follows:

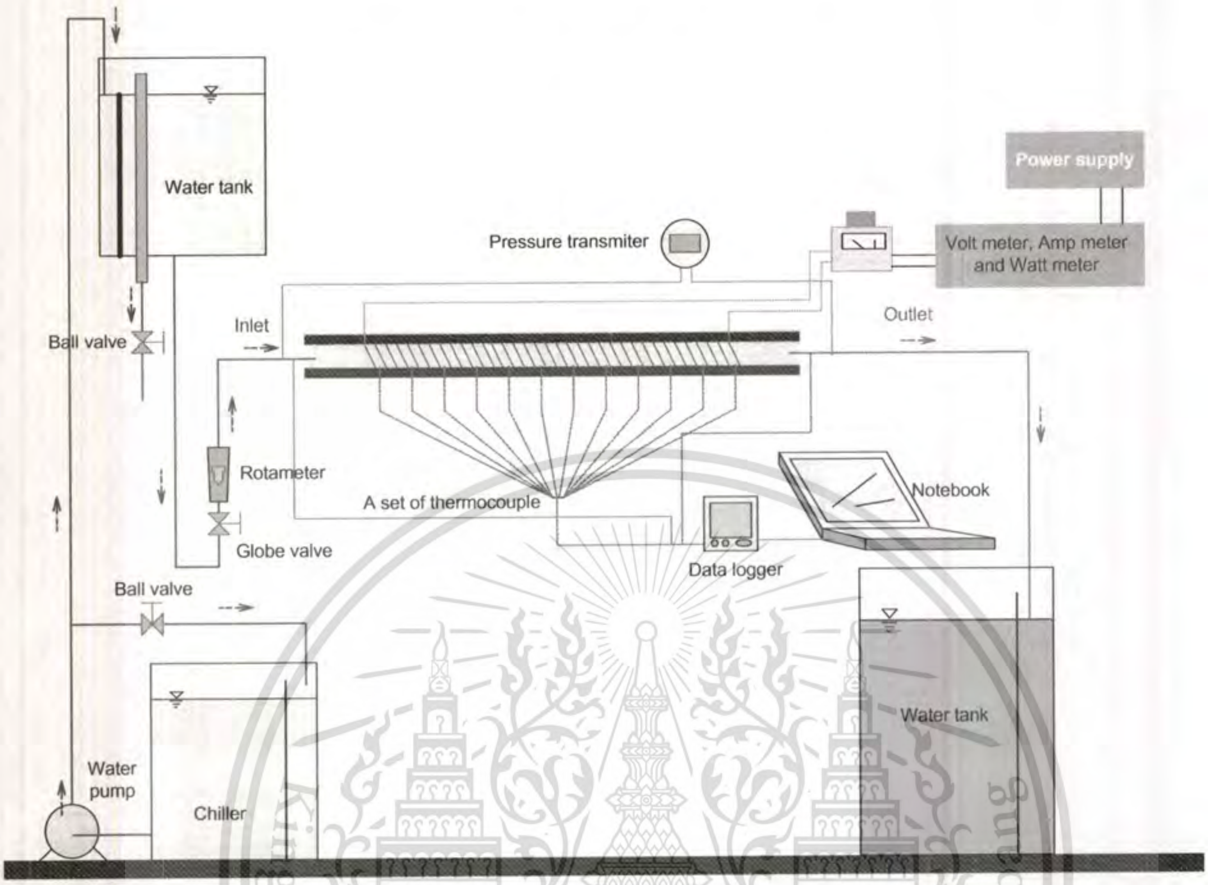

Fig. 2. Schematic diagram of the experimental heat transfer setup.

$$h = Q_w/A(\tilde{T}_s - T_b) \quad (6)$$

The heat transfer is measured by Nusselt number which can be obtained by

$$Nu = hD/k \quad (7)$$

The friction factor is calculated by pressure loss, Δp across the test length, L that defined as

$$f = \frac{\Delta P}{\left(\frac{L}{D}\right)\left(\rho \frac{U^2}{2}\right)} \quad (8)$$

where U is mean axial velocity of the test tube, ρ is the density of the working fluid and D is the inside diameter of the test tube. The situation of the flow can be defined from the Reynolds number that given by

$$Re = \rho UD/\mu \quad (9)$$

All of thermo-physical properties of the water are determined at the overall bulk fluid temperature (T_b). The uncertainties of the reduced data obtained experimentally are determined. Following the procedure based on ANSI/ASME [13] on reporting for the uncertainties in experimental measurements and results, the maximum uncertainty associated with the pressure, temperature, Reynolds number and Nusselt number are estimated to be ±4.8%, ±0.1%, ±8% and ±14%, respectively.

5. Experimental results and discussion

In this section, heat transfer rate, friction factor and thermal performance factor behaviors in the tubes fitted with the delta-winglet twisted tapes (oblique or straight delta-winglet twisted tape) for different twist ratios (y/w) and depth of wing cut ratios (DR) are reported. The results obtained by using plain tube and the tubes equipped with typical twisted tape are also presented for comparison.

5.1. Proof of plain tube and tube fitted with twisted tape

Heat transfer rate and flow friction characteristics in terms of Nusselt number and friction factor for the present plain tube are declared with standard correlations in Fig. 3a and b, for the proof of the plain tube. The data obtained from the experiments, shown in both figures, are in satisfactory agreement with the correlations. As seen in Fig. 3a, the Nusselt number data are within 7.4% error from Dittus and Boelter correlation ($Nu = 0.023Re^{0.8}Pr^{0.4}$) [14]. In Fig. 3b, the isothermal friction factors for the present plain tube are 12% higher than those obtained from the modified correlation of Blasius. In addition, the correlations of present experimental results for Nusselt number and friction factor are obtained as follows:

$$Nu = 0.028Re^{0.79}Pr^{0.4} \quad (10)$$

$$f = 0.448Re^{-0.275} \quad (11)$$

For the proof of the present typical twisted tapes, Nusselt number and friction factor of a tube fitted with the present typical twisted tapes are compared with the results obtained from the correlations by Maglik and Bergles [15] as demonstrated in Fig. 4a and b. Apparently, present results agree well with the available correlations within ±20% for Nusselt number and ±20% for friction factor.

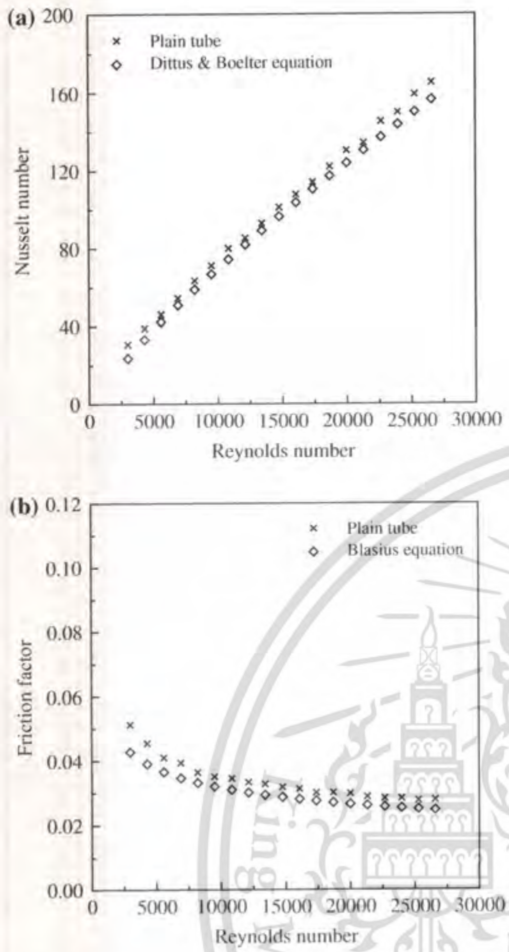

Fig. 3. Proof of the plain tube: (a) Nusselt number and (b) friction factor.

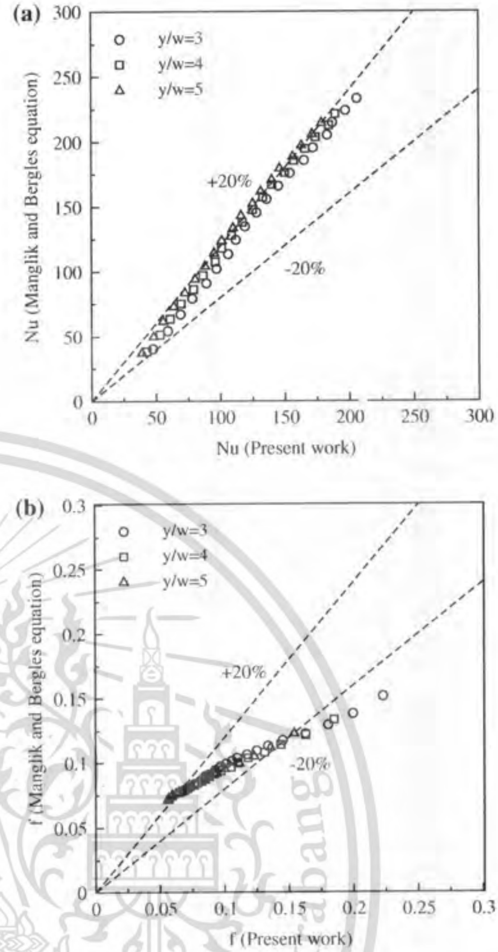

Fig. 4. Proof of the plain tube fitted with twisted tapes: (a) Nusselt number and (b) friction factor.

5.2. Effect of delta-winglet twisted tape (DWT)

Variation of Nusselt number with Reynolds number in the tube fitted with delta-winglet twisted tape (*DWT*), the tube fitted with typical twisted tape (*TT*) and also the plain tube are presented in Fig. 5a and b. For all experiments, Nusselt number increases with increasing Reynolds number. This is attributed to the increase of heat convection and also swirl flow intensity. Apparently, Nusselt numbers in the tube with both oblique and straight delta-winglet twisted tapes are higher than those in the plain tube and the tube with typical twisted tape insert over the considered Reynolds number range. Refers to the past investigations [16–19], typical delta-winglet tapes were applied as vortex generators to amplify turbulence intensities and produce secondary flows near the tube wall. Therefore, the high heat transfer rate in the tube with both oblique and straight delta-winglet twisted tapes (*O-DWT* and *S-DWT*) for the present work may be caused by the synergy effect of (1) vortex circulation together with secondary flow generated by the delta-winglet part and (2) main swirl flow produced by twisted tape. This effect results in superior heat transfer enhancement over that caused by the typical tape (*TT*) which induces only swirling flow. In the range of the present experiments, the Nusselt numbers for tube equipped with *DWT*, are respectively 1.1–2.55 and 1.02–1.64 times of those for plain tube and tube with typical twisted tape insert.

Friction factors are plotted against Reynolds numbers in Fig. 6a and b. Not surprisingly, the friction factors obtained from the tube with delta-winglet twisted tape (*DWT*) are considerably higher than those from the plain tube and also the tube with typical twisted tape insert (*TT*). This is a result of additional blockage and disturbance to the flowing stream generated by the delta-winglet part of *DWT* over those induced by typical twisted tape. Over the range considered, the friction factors for tube equipped with *DWT* are respectively 2.5–7.02 and 1.08–1.95 times of those for plain tube and tube with typical twisted tape insert.

5.3. Effect of twist ratio (y/w)

Fig. 5a and b shows variation of Nusselt number with Reynolds number in tube with oblique delta-winglet twisted tape (*O-DWT*) and straight delta-winglet twisted tape (*S-DWT*) for various twist ratios (y/w = 3, 4 and 5). At the given Reynolds number, the Nusselt number consistently increases with the decrease of twist ratio (y/w). This is due to the fact that, the tape with smaller twist ratio (y/w) induces stronger swirl/turbulent intensity, and also gives longer flowing path, leading to longer residence time and thus more efficient heat transfer compared to that with larger twist ratio (y/w). Mean Nusselt numbers for the *S-DWT* with twist ratio y/w of 3, 4 and 5 are respectively, 1.6, 1.4 and 1.23 times of that for the plain tube and 1.2, 1.16 and 1.1 times of that in the tube equipped with typical twisted tape (*TT*). With a similar trend, mean Nusselt numbers for the *O-DWT* with twist ratios y/w of 3, 4 and 5 are respectively, 1.66, 1.45 and 1.31 times of that for the plain tube and 1.25, 1.2 and 1.17 times of that for the tube with typical twisted tape insert. Depending on the Reynolds number, Nusselt

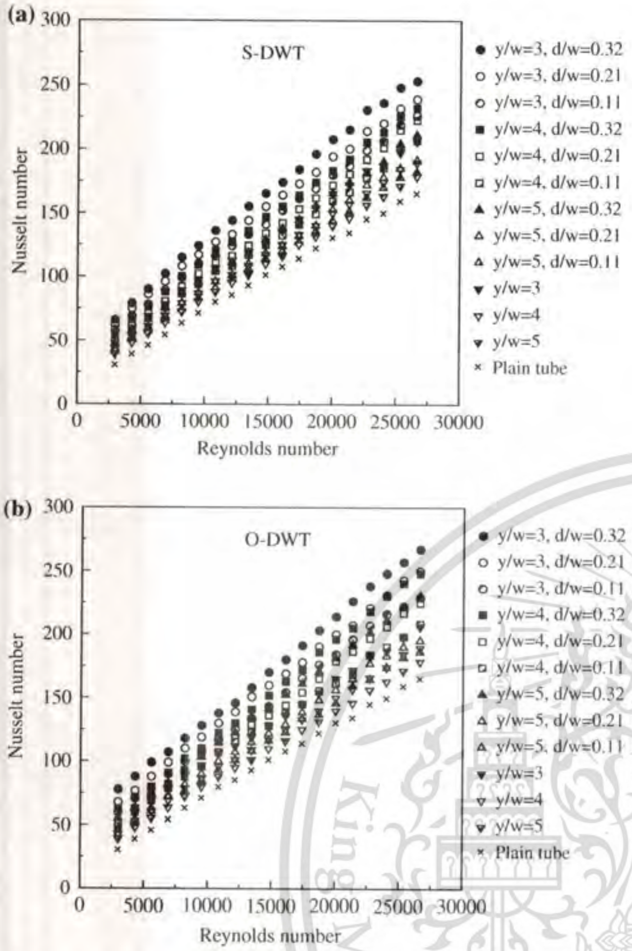

Fig. 5. Nusselt number versus Reynolds number for tubes with *DWT*, *TT*, and plain tube: (a) *S-DWT* and (b) *O-DWT*.

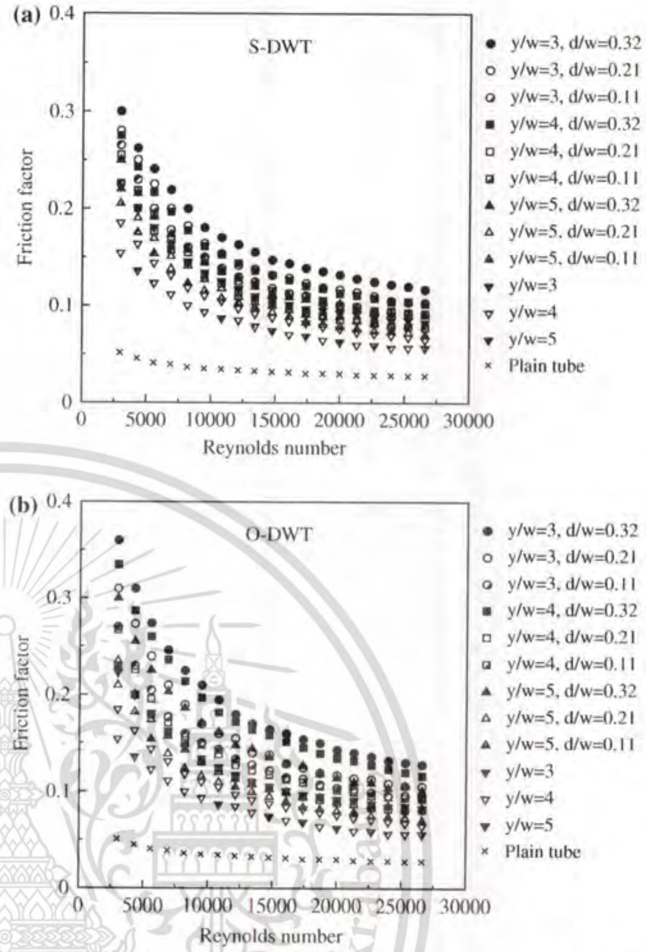

Fig. 6. Friction factor versus Reynolds number for tubes with *DWT*, *TT*, and plain tube: (a) *S-DWT* and (b) *O-DWT*.

numbers from using the *O-DWT* and *S-DWT* with twist ratio (y/w) of 3 are around 7–29% and 15.8–45.6% greater than those from the tapes with twist ratios (y/w) of 4 and 5, respectively.

Fig. 6a and b shows variation of friction factor with Reynolds number for oblique delta-winglet twisted tape (*O-DWT*) and straight delta-winglet twisted tape (*S-DWT*) at various twist ratios (y/w = 3, 4 and 5). As found, the friction factor increases with decreasing twist ratio (y/w). This is a consequence of the reasons mentioned for Nusselt number, in which twisted tape with shorter twist length provides longer flowing path, resulting in larger tangential contact between the flowing stream and tube surface. Over the range studied, mean friction factors for the *S-DWT* with twist ratios y/w of 3, 4 and 5 are respectively, 4.35, 3.86 and 3.4 times of that for the plain tube and 1.25, 1.34 and 1.4 times of that for the tube with typical twisted tape insert. For a tube with *O-DWT*, friction factors with twist ratios y/w of 3, 4 and 5 are respectively, 4.68, 4.18 and 3.67 times of that for the plain tube and 1.33, 1.45 and 1.52 times of that for the tube with typical twisted tape insert. In addition, friction factors for *O-DWT* and *S-DWT* with twist ratio (y/w) of 3 are approximately 3.5–22.8% and 17.9–37.8% over those for the tapes equipped with twist ratios (y/w) of 4 and 5, respectively.

5.4. Effect of depth of wing cut ratios (DR)

Effects of the of depth of wing cut ratios ($DR = d/w = 0.11$, 0.21 and 0.32) on the Nusselt number and friction factor are presented in Fig. 5a and b and Fig. 6a and b. Refer to Fig. 5a and b, for both tubes with *S-DWT* and *O-DWT*, Nusselt number increases with increasing wing cut ratios (*DR*). As the depth of wing cut increases, longer wing part protruded into flowing stream. Consequently, the flow is more seriously disturbed and degree of turbulence intensity is raised. This leads to more effective heat transfer enhancement. The results also show that mean Nusselt numbers for the *S-DWT* and *O-DWT* with the highest depth of wing cut ratios, $DR = 0.32$ are respectively, 1.25–2.16 and 1.35–2.55 times of that for the plain tube and 1.16–1.39 and 1.29–1.64 times of that for the tube with typical twisted tape. In addition, the average Nusselt number for employing the tapes with of depth of wing cut ratio (*DR*) of 0.32 is respectively, 17.3% and 9.7% higher than those for the tapes depth of wing cut ratios of 0.11 and 0.22.

Fig. 6a and b indicates that the tape insert with larger depth of wing cut ratio (*DR*) induces higher friction factor. This is caused by increasing extent of flow obstruction by the wing parts of the twisted tape. Mean friction factors for the *S-DWT* and *O-DWT* with the highest depth of wing cut ratio, $DR = 0.32$ are respectively, 3.22–5.85 and 3.62–7 times of that in the plain tube and 1.31–1.63 and 1.51–1.95 times of that in the tube with typical twisted tape. In addition, the average friction factor for employing the tape with depth of wing cut ratio, $DR = 0.11$ is found to be 10.9% and 25.2% lower than that for the tapes with $DR = 0.21$ and 0.32.

5.5. Effect of straight and oblique wings

Comparisons shown in Figs. 7 and 8 reveal that the oblique delta-winglet twisted tapes (*O-DWT*) give higher Nusselt number and

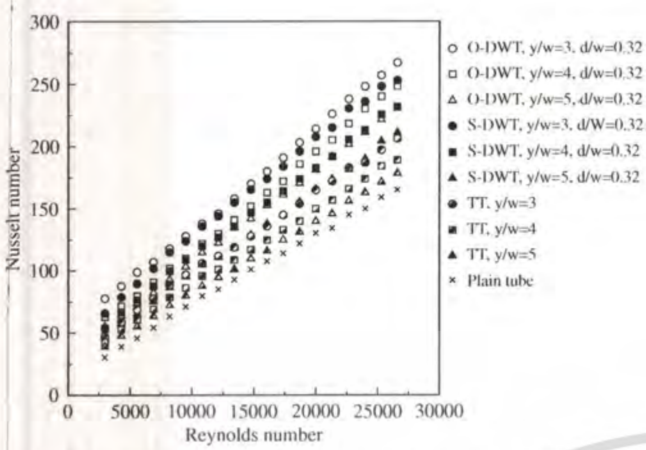

Fig. 7. Effect of straight and oblique wings on Nusselt number.

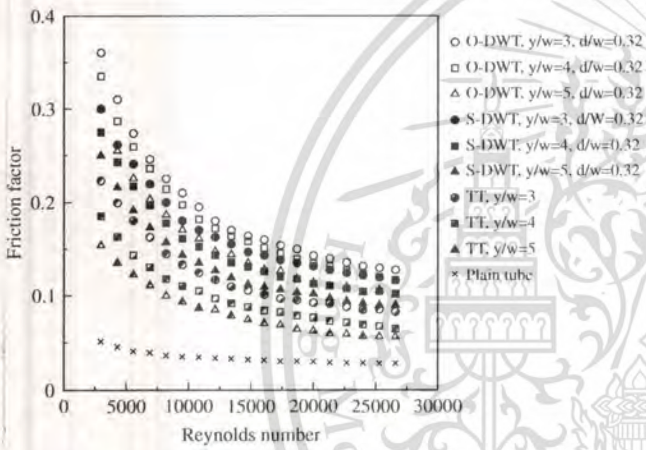

Fig. 8. Effect of straight and oblique wings on friction factor.

also friction factor than the straight delta-winglet twisted tapes (*S-DWT*), when other parameters are controlled. These results are directly related to geometries of both tape arrangements. As depicted in Fig. 1, with the same depth of wing cut, the *O-DWT* can be more effectively twisted, thus the tape possesses larger protruding part than the *S-DWT*. Consequently, greater heat transfer rate and also friction factor are obtained for the *O-DWT*. The calculated data show that the mean Nusselt number and friction factor for the *O-DWT* are respectively, 4.2% and 7.8% higher than those for the *S-DWT*.

5.6. Performance criteria

According to the previous studies [20–23], a comparison of heat transfer coefficients in plain tube (p) and tube fitted with turbulator (t) was made at the same pumping power since it is relevant to operation cost.

For constant pumping power:

$$(\dot{V}\Delta P)_p = (\dot{V}\Delta P)_t \qquad (12)$$

Eq. (12) can be rewritten in terms of friction and Reynolds number as

$$(fRe^3)_p = (fRe^3)_t \qquad (13)$$

Thermal performance factor (η) at equal pumping power is defined as ratio of the convective heat transfer coefficient of the tube with turbulator to that of the plain tube which can be expressed as:

$$\eta = \left.\frac{h_t}{h_p}\right|_{pp} \qquad (14)$$

In general, the thermal performance factor above unity indicates that the effect of heat transfer enhancement due to the turbulator (or enhancing device) is more dominant than the effect of rising friction and vice versa. The thermal performance factors for the oblique delta-winglet twisted tapes (*O-DWT*) and straight delta-winglet twisted tapes (*S-DWT*) calculated from Eqs. (17) and (20) based on the same pumping power, are plotted versus Reynolds number in Figs. 9 (a–b) and 10. The performance factors in the tube equipped with typical twisted tape (*TT*) are also plotted for comparison. At the same Reynolds number, the performance factors for both *O-DWT* and *S-DWT* are found to be greater than those for the typical twisted tape (*TT*). The performance factors for all twisted tapes tend to decrease with increasing Reynolds number. This suggests that both *DWTs* are feasible in terms of energy saving at higher Reynolds numbers compared to typical twisted tape (*TT*) which is suitable only for low Reynolds number as suggested in the previous works [3,20–22].

Similar to the effects found for Nusselt number and friction factor, the performance factor increases with decreasing twist ratio (y/w) and increasing depth of wing cut (DR) for all given Reynolds numbers. In addition, the performance factor for the *O-DWT* is greater than that in the *S-DWT* tubes, for the similar operation test conditions. Over the range considered, the values of performance factors obtained from using *O-DWT* and *S-DWT* are about 0.92–

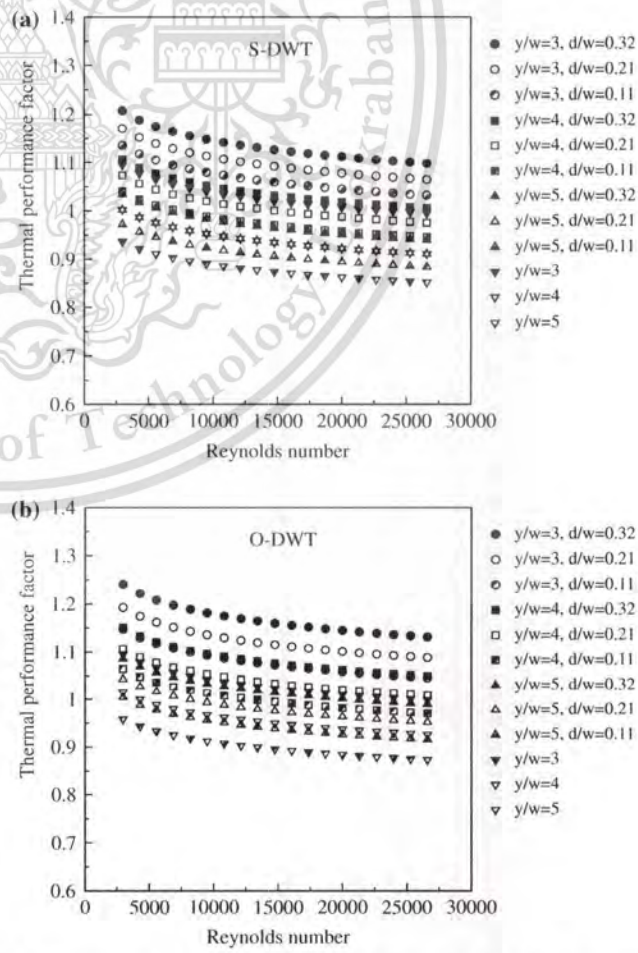

Fig. 9. Thermal performance factor versus Reynolds number for tubes with *DWT*, *TT*, and plain tube: (a) *S-DWT* and (b) *O-DWT*.

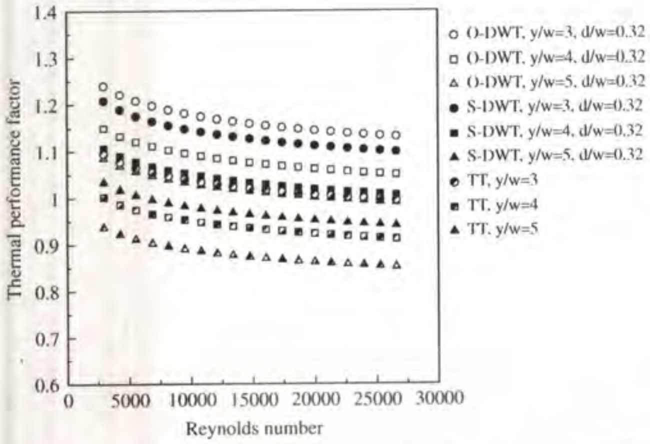

Fig. 10. Effect of straight and oblique wings on thermal performance factor.

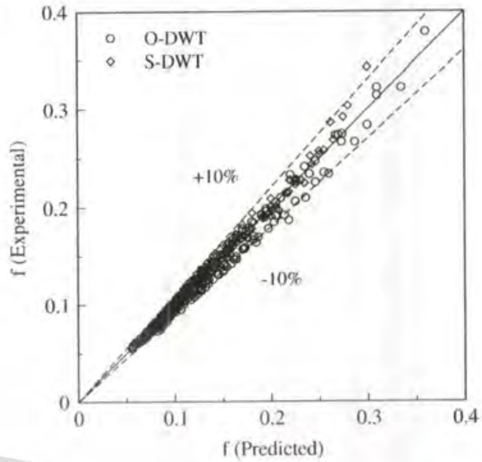

Fig. 12. Comparison of experimental friction factor and predicted results.

1.24 and 0.88–1.21, respectively. The maximum thermal performance factor of 1.24 is achieved by using of *O-DWT* with $DR = 0.32$ and $y/w = 3$ at Reynolds number of 3000.

Empirical correlations for Nusselt number (Nu), friction factor (f), thermal performance factor (η) are developed for the tube with delta-winglet twisted tape inserts in the range of Reynolds number between 3000 and 27,000, $Pr = 4.91–5.57$, twist ratio ($y/w = 3$, 4 and 5), and depth of wing cut ratios ($DR = d/w = 0.11$, 0.21 and 0.32) as follows.

For oblique delta-winglet twisted tapes(O-DWT) :

$$Nu = 0.18Re^{0.67}Pr^{0.4}(y/w)^{-0.423}(1+(d/w))^{0.982} \quad (15)$$

$$f = 24.8Re^{-0.51}(y/w)^{-0.566}(1+(d/w))^{1.87} \quad (16)$$

$$\eta = 2.04Re^{-0.042}Pr^{0.4}(y/w)^{-0.261}(1+(d/w))^{0.45} \quad (17)$$

For straight delta-winglet twisted tapes(S-DWT) :

$$Nu = 0.184Re^{0.675}Pr^{0.4}(y/w)^{-0.465}(1+(d/w))^{0.76} \quad (18)$$

$$f = 21.7Re^{-0.45}(y/w)^{-0.564}(1+(d/w))^{1.41} \quad (19)$$

$$\eta = 2.164Re^{-0.0435}Pr^{0.4}(y/w)^{-0.304}(1+(d/w))^{0.356} \quad (20)$$

The experimental data and predicted data of the correlation by Eqs. (15), (16), (19), (20) are compared in Figs. 11 and 12. It appears that the mean absolute percentage deviation of the present experimental Nusselt number data is within 10% for Eq. (15) and 10% for Eq. (18) from the values predicted by the present correlations while that of the friction factor is within 10% and 10% for Eqs. (16) and (19), respectively.

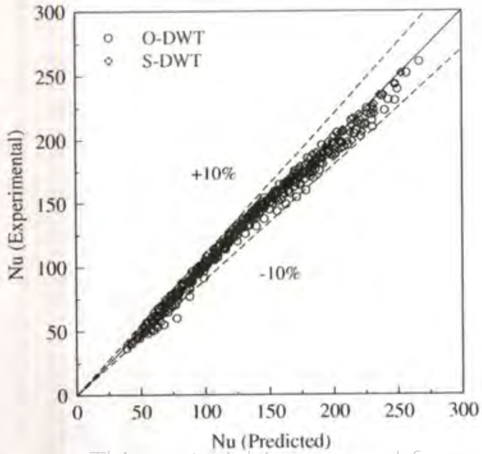

Fig. 11. Comparison of experimental Nusselt number and predicted results.

6. Conclusions

The enhancement of the heat transfer in a tube fitted with delta-winglet twisted tapes which act as swirl generator and turbulator is experimentally investigated. The values of Nusselts number and friction factor in the test tube equipped with delta-winglet twisted tape are noticeably higher than those in the plain tube and also tube equipped with typical twisted tape. Nusselt number and friction factor increase with decreasing of twist ratio (y/w) and increasing depth of wing cut ratio (DR) for all Reynolds numbers studied. In addition, the *O-DWT* gives higher Nusselt number and friction factor than that of the *S-DWT*. The thermal performance factor in the tube with *O-DWT* is greater than that with *S-DWT* and the factor increases with decreasing Reynolds number and increasing twist ratio. Over the range considered, the performance factor in the tubes equipped with the *O-DWT* and *S-DWT* are found to be around 0.92–1.24 and 0.88–1.21, respectively. The results of the present work agree well with the available correlations within ±10% in comparison with experimental data for the friction factor and within ±10% for the Nusselt number. In this study, several *DWT* tapes give the thermal performance factor higher than unity while the typical twisted tape provides lower unity. It is obvious that the *DWT* performs better heat transfer enhancement than that typical twisted tape. It indicates that the heat exchanger fitted with *DWT* is more compacted than one with the typical twisted tape. Again, the *DWT* can be replaced any of the *TT* efficiently to reduce the size of the heat exchanger.

Acknowledgements

The financial support of the present work by the Thailand Research Found (TRF) is gratefully appreciated.

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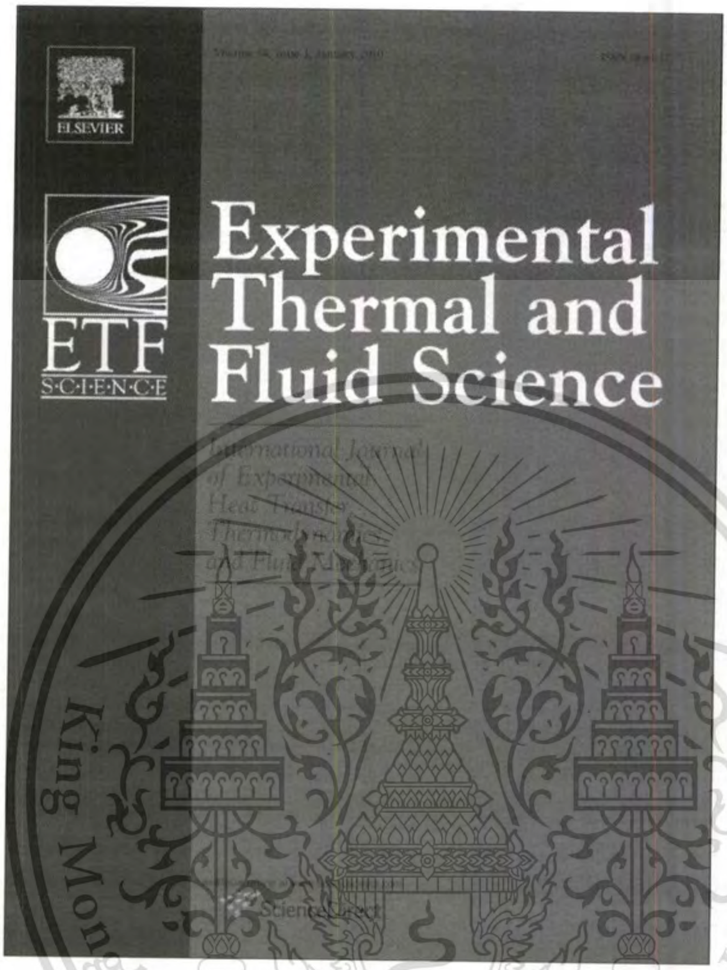
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Experimental Thermal and Fluid Science 34 (2010) 53–62

Contents lists available at ScienceDirect

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Turbulent heat transfer enhancement by counter/co-swirling flow in a tube fitted with twin twisted tapes

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ARTICLE INFO

Article history:
Received 17 April 2009
Received in revised form 20 August 2009
Accepted 7 September 2009

Keywords:
Heat transfer
Friction factor
Enhancement index
Twin counter twisted tapes
Twin co-twisted tapes

ABSTRACT

In the present study, the influences of twin-counter/co-twisted tapes (*counter/co-swirl tape*) on heat transfer rate (*Nu*), friction factor (*f*) and thermal enhancement index (η) are experimentally determined. The twin counter twisted tapes (*CTs*) are used as *counter-swirl flow generators* while twin co-twisted tapes (*CoTs*) are used as *co-swirl flow generators* in a test section. The tests are conducted using the *CTs* and *CoTs* with four different twist ratios (y/w = 2.5, 3.0, 3.5 and 4.0) for Reynolds numbers range between 3700 and 21,000 under uniform heat flux conditions. The experiments using the single twisted tape (*ST*) are also performed under similar operation test conditions, for comparison. The experimental results demonstrate that Nusselt number (*Nu*), friction factor (*f*) and thermal enhancement index (η) increase with decreasing twist ratio (y/w). The results also show that the *CTs* are more efficient than the *CoTs* for heat transfer enhancement. In the range of the present work, 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 indices (η) 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, while those obtained by using the *CoTs* with the same range of y/w are 1.1, 1.03, 0.97 and 0.92, respectively. In addition, the empirical correlations of the heat transfer (*Nu*), friction factor (*f*) and thermal enhancement index (η) are also reported.

1. Introduction

A high cost of energy and material has resulted in an increased effort aimed at producing more efficient heat exchange equipment. The methods of improving convective heat transfer in the tubes of heat exchangers have been widely investigated by many researchers. Heat transfer enhancement techniques can be classified into two groups: (1) active technique that needs external power source and (2) passive technique which does not need external power source. Both active and passive techniques have been applied to improve heat transfer in several areas such as nuclear reactors, chemical reactors and general heat exchangers. The principle of the passive technique involves either surface treatment, such as coated surface, rough surface and extended surface or flow manipulation such as swirl flow and modified flow. One of the most favourable passive techniques is generating swirl flow by insertion of a twisted tape because the tape is inexpensive and can be easily employed to the existing system. The effects of twisted tape insertion have been widely studied for both experimental and numerical simulation works. The presence of twisted tape directs toward reducing the hydrodynamic or thermal boundary layer thickness, leading to greater convective heat transfer. However, in the process, pumping power may increase significantly and ultimately the pumping cost becomes high. Therefore, to achieve a desired heat transfer rate in an existing heat exchanger at an economic pumping power, the design of twisted tape with a proper geometry is necessary.

For several years, research work on heat transfer enhancement in heat exchanger by using twisted tape has grown strongly. After the success of the use of twisted tape for heat transfer augmentation which was early reported by Whitham [1], further improvements of thermal performance for tubes with various geometries of twisted tape inserts have been released. Kidd [2] and Klepper [3] invented the short length twisted tapes for the use in gas cooled nuclear reactor and inferred that the tapes were more efficient than the full length twisted tapes. Saha et al. [4] examined the heat transfer enhancement in a round tube fitted with the regularly spaced twisted tape elements. Date and Gaitonde [5] developed correlations for predicting characteristics of laminar flow in a tube fitted with regularly spaced twisted tape elements. In addition, the experimental investigations of heat transfer enhancement by means of regularly spaced twisted tapes were also reported by Dasmahapatra and Raja Rao [6] and Eiamsa-ard et al. [7].

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doi:10.1016/j.expthermflusci.2009.09.002

Nomenclature

A	heat transfer surface area, m^2
C_p	specific heat at constant pressure, $J\,kg^{-1}\,K^{-1}$
D	inside diameter of the test tube, m
f	friction factor = $\Delta P/((L/D)(\rho U^2/2))$
h	heat transfer coefficient, $W\,m^{-2}\,K^{-1}$
I	current, A
k	thermal conductivity of fluid, $W\,m^{-1}\,K^{-1}$
L	length of the test section, m
M	mass flow rate, $kg\,s^{-1}$
Nu	Nusselt number = hD/k
P	pressure of flow in stationary tube, Pa
Pr	Prandtl number = $\mu C_p/k$
ΔP	pressure drop, Pa
Q	heat transfer rate, W
Re	Reynolds number = $\rho UD/\mu$
t	thickness of the test tube, m
T	temperature, °C
$\tilde{T}$	mean temperature, °C
U	average axial flow velocity, $m\,s^{-1}$
V	voltage, V
$\dot{V}$	volume flow rate, $m^3\,s^{-1}$
w	tape width, m
y	tape pitch length, m
y/w	twist ratio

Greek letters

δ	tape thickness, m
ρ	fluid density, $kg\,m^{-3}$
μ	fluid dynamic viscosity, $kg\,s^{-1}\,m^{-1}$
η	thermal enhancement index

Subscripts

b	bulk
conv	convection
e	empty tube
in	inlet
out	outlet
pp	pumping power
s	swirl generator
w	tube wall

Abbreviations

CTs	twin counter twisted tapes
CoTs	twin co-twisted tapes
ST	single twisted tape

A comparative study of heat transfer and friction factor in the round tubes fitted with single, twin and triple twisted tapes was reported by Chang et al. [8]. For their findings, the heat transfer enhancements achieved for laminar flows in the tubes fitted with single, twin and triple twisted tapes were, respectively, 1.5–2.3, 1.98–2.8 and 2.86–3.76 times of the Dittus–Boelter levels. In the range of $3000 < Re < 14{,}000$, the tubes fitted with twin and triple twisted tapes could offer the higher values of heat transfer augmentation with the similar levels of performance factor as those found in the tube fitted with single twisted tape. Chang et al. [9,10] also presented the promise of using the serrated twisted tapes and broken twisted tapes as heat transfer enhancing devices. As found, the heat transfer rates in the tubes fitted with the serrated and broken twisted tapes were, respectively, 1.25–1.67 and 1.28–2.4 times of that in the tube fitted with typical twisted tape. A recent study by Rahimi et al. [11] addressed the heat transfer, friction factor and thermal performance of a tube fitted with jagged, notched, perforated and classic twisted tapes for both experimental and computational fluid dynamics (CFD) works. Their results showed that the jagged twisted tape yielded higher heat transfer rate than the others. Bharadwaj et al. [12] determined the pressure drop and heat transfer behaviours in a 75-start spirally grooved tube fitted with twisted tape under laminar and moderately turbulent regions. They found that the direction of twist to grooved surface (clockwise and anticlockwise) influenced the thermo-hydraulic characteristics. Jaisankar et al. [13] investigated the heat transfer and friction factor characteristics of thermosyphon solar water heater system with full length twist tape, twist tape fitted with rod and spacer fitted at the trailing edge. It was observed that the overall performance for the twist tape fitted with rod was greater than that for the twist tape fitted with spacer. Apart from above literature review, literature of over hundred papers of heat transfer enhancement by using twisted tapes was described by Dewan et al. [14].

In the present work, twisted tapes with new design, twin-counter/co-twisted tapes (*counter/co-swirl tape*) are proposed for heat transfer enhancement in a circular tube. In the experiments, the twin counter twisted tapes (*CTs*) are used as the *counter-swirl flow generators* while the twin co-twisted tapes (*CoTs*) are used as *co-swirl flow generators*. The results obtained by the use of the newly designed twisted tapes are also compared with those by the use of the single twisted tape (*ST*) and the empty tube.

2. Twisted tapes

Twin twisted tapes are made of aluminium and have tape width (w) of 9 mm, tape thickness (δ) of 0.8 mm, and tape length (l) of 1000 mm. Dimensions of single twisted tape are same as those of the twin tapes, except tape width of 19 mm. Both twin-counter/co-twisted tapes were prepared with four different twist ratios, y/w = 2.5, 3.0, 3.5 and 4.0 where twist ratio is defined as twist length (180°/twist length) to tape width (w). The single tapes (*ST*) were prepared with three twist ratios, y/w = 3.0, 3.5 and 4.0, for comparative investigation. Comparison of geometric details of the twin-counter/co-twisted tapes (*counter/co-swirl tapes*) and also a single twisted tape is shown in Fig. 1 while details of tape geometries are presented in Table 1. The tape thickness of 0.8 mm was chosen to avoid tape twisting difficulty of the thinner tape which was torn easily during the twisting operation. On the other hand, to avoid an additional friction in the system that might be caused the thicker tape. To produce the twisted tape, one end of a straight tape was clamped while another end was carefully twisted to ensure a desired twist length. For twin twisted tapes, two tapes were twisted separately and then welded together with a small aluminium wire. As shown in Fig. 1, for the twin co-twisted tapes (*CoTs*), both tapes were well aligned and positioned to be twisted in same direction to generate identical direction swirl called *co-swirl flow*. On the other hand, the twin counter twisted tapes (*CTs*), two tapes were aligned to be twisted in opposite directions to produce *counter-swirl flow*.

3. Experimental setup

The experiments were carried out using an experimental facility as shown in Fig. 2. The test tube is made of copper and has inner diameter of 19 mm (D), outside diameter of 21 mm (D_o), wall thickness of 1.5 mm (t) and length of 1000 mm (L). In the experi-

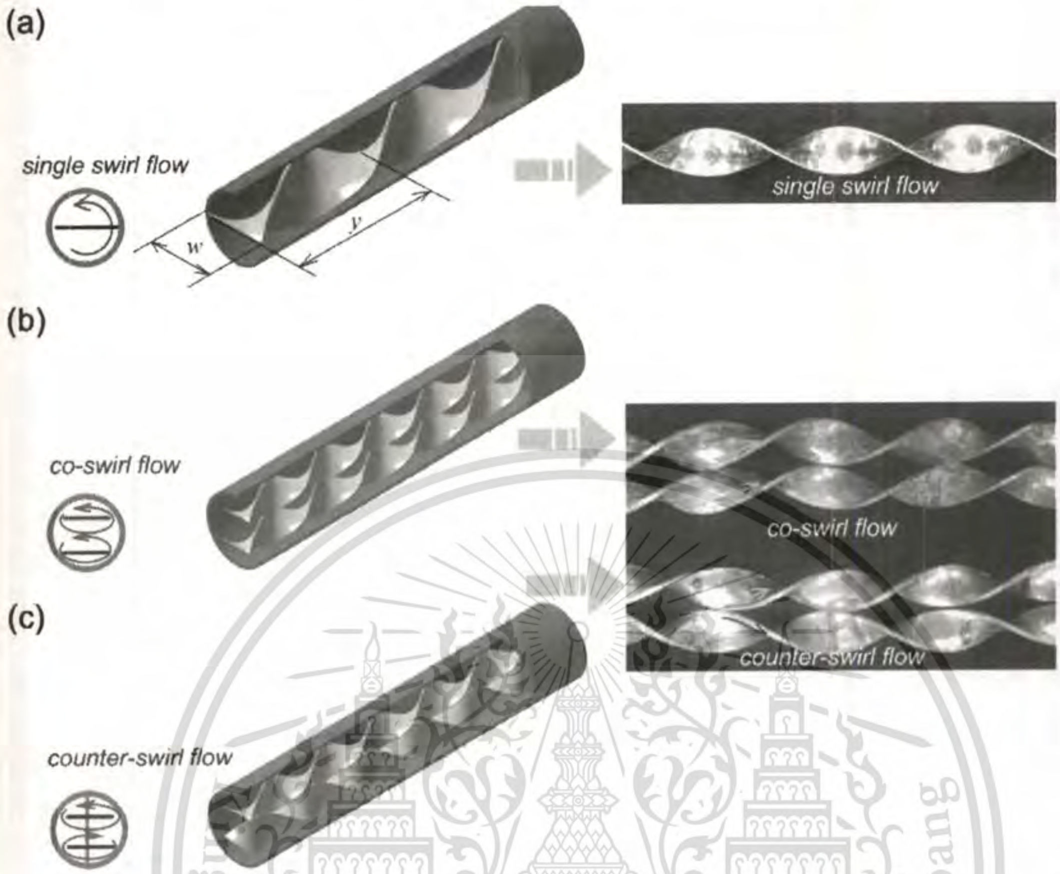

Fig. 1. Test tube with twisted tape inserts: (a) single twisted tape (*ST*), (b) twin co-twisted tapes (*CoTs*) and (c) twin counter twisted tapes (*CTs*).

Table 1
Details of twisted tape insert.

Twisted tape	*ST*	*CTs*	*CoTs*
(a) Tape width (w)	19 mm	9 mm	Same as *CTs*
(b) Tape pitch length (y)	57, 66.5 and 76 mm	22.5, 27, 31.5 and 36 mm	Same as *CTs*
(c) Twist ratio (y/w)	3.0, 3.5 and 4.0	2.5, 3.0, 3.5 and 4.0	Same as *CTs*
(d) Tape thickness (δ)	0.8 mm	Same as *ST*	Same as *ST*
(e) Material	Aluminium	Same as *ST*	Same as *ST*
(f) Swirl type	Single swirl flow	Counter-swirl flow	Co-swirl flow

ments, the single and twin-counter/co-twisted tapes were inserted at the core tube along the test section. The rather long tube provided sufficient contact surface between the tapes and tube wall for the firm attachment of the tapes to the tube without the need of any extra fitting. The tube was wound with electrical heating wire covered with ceramic beads. During the test, the tube was heated by continually winding flexible electrical wire providing a uniform heat flux condition. The electrical output power was controlled by a variac transformer to obtain a constant heat flux along the entire length of the test section and by keeping the current of less than 10 A. The outer surface of the test tube was well insulated to minimize convective heat leak to surroundings. Moreover, necessary precautions were taken to prevent leakages from the system. In the experiment, the heat transfer losses from the test tube are around 3–5% of the total heat input ($Q = IV$). The inner and outer temperatures of the water were measured at certain points with a multi-channel temperature measurement unit in conjunction with the resistance temperature detectors (RTDs). Fifteen thermocouples were tapped on the local wall of the tube and the thermocouples were placed round the tube to measure the circumferential temperature variation, which was found to be negligible. The mean wall temperature was determined by means of calculations based on the reading of copper–constantan thermocouples.

The test loop consists of a water pump, data logger, pressure transmitter, thermocouple/RTD, rotameter and heat transfer test section as depicted in Fig. 2. In the apparatus setting above, the inlet water from a reservoir tank was discharged through the rotameter then the heat transfer test section. For each test run, it was necessary to record the data of temperature, volumetric flow rate and pressure drop of the bulk water at steady state conditions in which the inlet water temperature was maintained constant at 27 °C. The various characteristics of the flow friction and the Reynolds numbers were based on the bulk temperature. During the test, local wall temperature, inlet and outlet water temperatures, pressure drop across the test section and flow velocity were measured. The average Nusselt numbers were calculated and discussed where all fluid properties were determined at the overall bulk mean temperature. Details of the test conditions are shown in Table 2.

The Reynolds numbers of the cold water were varied from 3700 to 21,000. Uncertainties of measurements are estimated based on ANSI/ASME. The uncertainties of axial velocity, pressure and temperature are found to be within ±7%, ±5% and ±0.5%, respectively. In addition, the uncertainties of non-dimensional parameters are ±5% for Reynolds number, ±10% for Nusselt number and ±12% for friction.

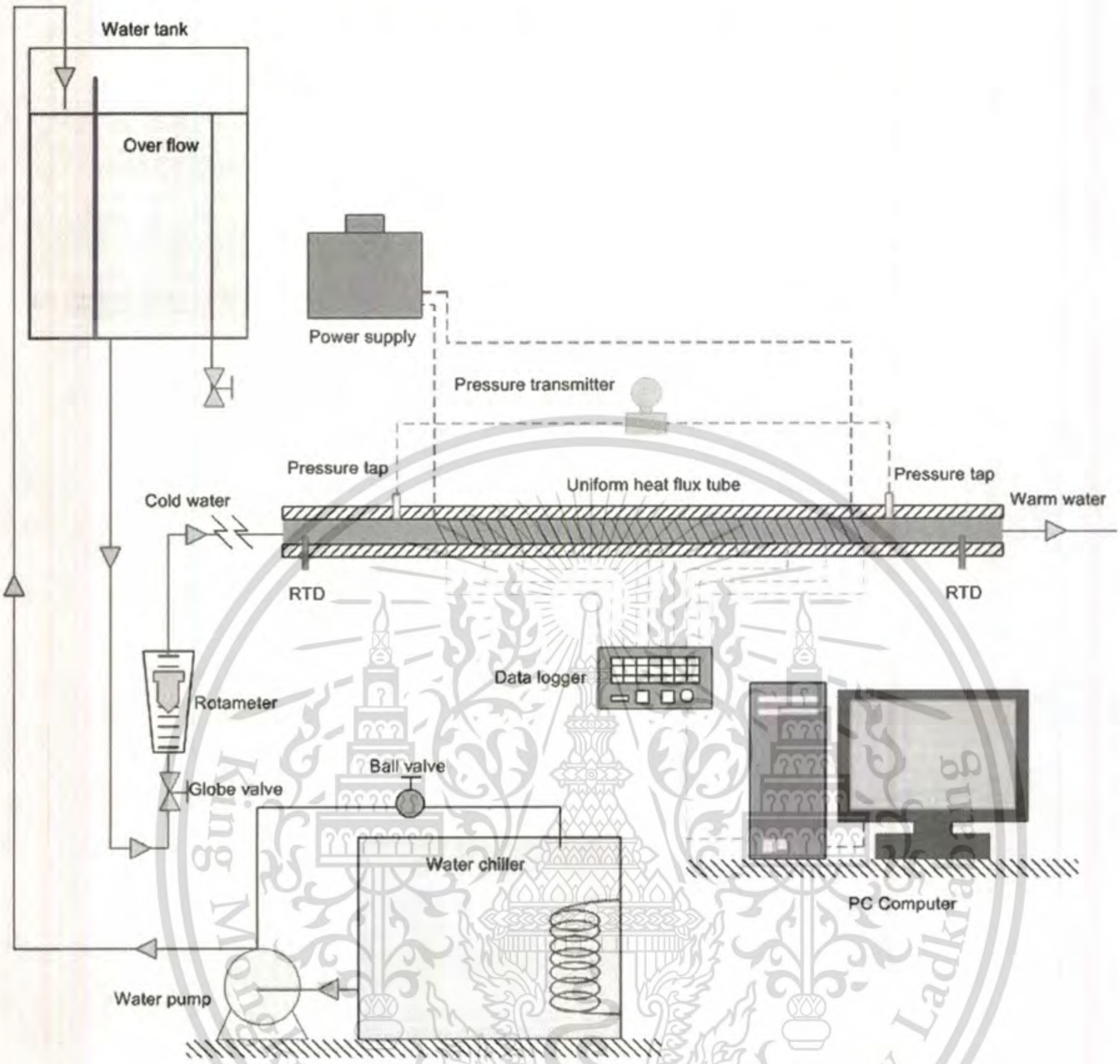

Fig. 2. Schematic diagram of the experimental heat transfer set-up.

Table 2
Details of heating tube and experimental conditions.

(a) Inner diameter	19 mm
(b) Outer diameter	21 mm
(c) Wall thickness	1.5 mm
(d) Length	1000 mm
(e) Material	Copper
(f) Wall condition	Uniform heat flux
(g) Reynolds number	3700–21,000
(h) Working fluid	Water

4. Heat transfer and friction factor measurements

In experiments, water is used as a working fluid and flowed through a test tube under uniform heat flux condition. The heat transfer rate at steady state is assumed to be equal to the heat loss from the test section which may be expressed as:

$$Q_{water} = Q_{conv} \tag{1}$$

Then the heat transferred from the water flow can be drawn as,

$$Q_{water} = MC_p(T_{out} - T_{in}) \tag{2}$$

The heat transferred from the tube wall by convection may be also written as,

$$Q_{conv} = hA\left|\tilde{T}_w - \left(\frac{T_{out} + T_{in}}{2}\right)\right| \tag{3}$$

where T_{out} is the outlet temperature of water flow, T_{in} is the inlet water temperature, T_w is the temperature of the locations in the centerline of the tube wall and A is the total surface area of the inside tube wall, which can be expressed as:

$$A = \pi DL \tag{4}$$

where D is the inner diameter of the test tube. The average heat transfer coefficient (h) for the test tube can be calculated by the combination of Eqs. (2) and (3),

$$h = \frac{MC_p(T_{out} - T_{in})}{A\left|\tilde{T}_w - \left(\frac{T_{out}+T_{in}}{2}\right)\right|} \tag{5}$$

In the experimental results, the average Nusselt number is defined as follow,

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

where k is the local thermal conductivity of the fluid which calculated from the fluid properties at the local mean bulk fluid temperature ($T_b = (T_{out} + T_{in})/2$). The friction factor for tube with or without twisted tape can be calculated using pressure loss, ΔP, across the test length, L, via following equation:

$$f = \frac{1}{2}\frac{\Delta P}{L}\frac{D}{\rho U^2} \tag{7}$$

where ρ is the density at the mean bulk temperature, and U is the average velocity based on the inner diameter.

The Nusselt number, Prantl number, Reynolds number, and all of thermo-physical properties of the fluid were calculated of the basis of water properties corresponding to the bulk fluid temperature (T_b). The Reynolds number based on the total flow rate at the inlet of the test section is expressed as:

$$Re = \rho UD/\mu \tag{8}$$

where μ is the dynamic viscosity of the working fluid.

It is very important to validate the present empty tube data of Nusselt number and friction factor in fully developed straight/axial flow with the correlations from the previous studies. In the present work, Nusselt number and friction factor from the experimental data are compared with the correlations recommended by Dittus–Boelter and Blasius [15].

5. Experimental results

In this section, the heat transfer and friction factor characteristics and also thermal enhancement index in a tube fitted with twin-counter/co-twisted tapes (*counter/co-swirl tape*) are presented. The experiments are performed using twin twisted tapes with four different twist ratios, $y/w = 2.5$, 3.0, 3.5 and 4.0, in the range of Reynolds number between 3700 and 21,000. The results obtained for the tube fitted with the single twisted tapes (*ST*) with three different twist ratios, $y/w = 3.0$, 3.5 and 4.0, and the empty tube are used as the reference data for the performance evaluation of the modified twisted tapes.

5.1. Validation test

Empty tube data serve as a qualification for the facility and the procedure used over the range of Reynolds number studied. Qualification of the heat transfer rate (Nu) and friction factor (f) of the empty tube is evaluated by comparing the experimental data with the previous correlations [15] under similar conditions. The results shown in Fig. 3a and b, reveal that the data for present empty tube are in good agreement with the previous reports for both the Nusselt number (Nu) and the friction factor (f) correlations. As found, the mean absolute percentage deviations of the present experimental Nusselt number data are ±6% from the values predicted by Dittus–Boelter correlation, while the average absolute percentage deviations for friction factor data are ±10.4% from the values predicted by Blasius correlation. The empirical correlations of the Nusselt number and friction factor for the present empty tube can be expressed as follows:

$$Nu = 0.04Re^{0.75}Pr^{0.4} \tag{9}$$

$$f = 0.376Re^{-0.259} \tag{10}$$

Comparisons between the present results and correlations by Manglik and Bergles [16] using the tube fitted with single twisted tapes (*ST*) at three different twist ratios, $y/w = 3.0$, 3.5 and 4.0 are demonstrated in Fig. 4a and b for Nusselt number and Fig. 5a

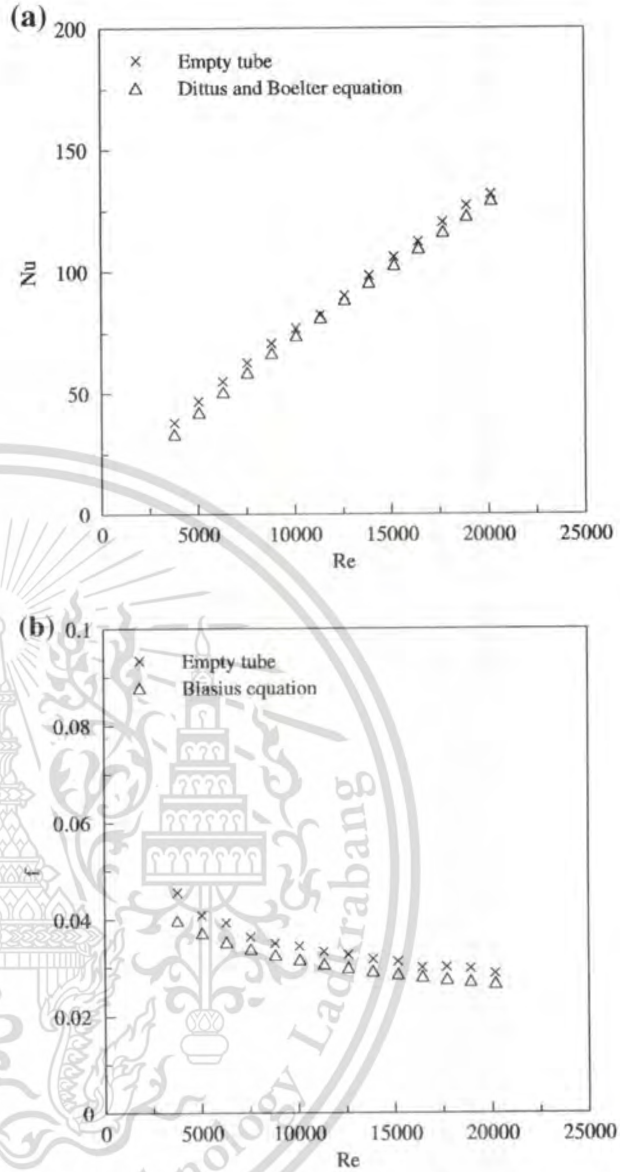

Fig. 3. Confirmatory test of empty tube: (a) Nusselt number and (b) friction factor.

and b for friction factor. Compared to the results obtained from the past investigation [16], the present Nusselt numbers are found to be slightly lower while the present friction factors are noticeably higher at the low Reynolds number region and then become comparable at the high Reynolds number. In addition, it is obvious that the present Nusselt numbers and friction factors agree well with the available correlations within ±15–20% for both Nusselt number and friction factor.

5.2. Effect of twin-counter/co-twisted tape inserts

Effects of the twin counter twisted tapes (*CTs: counter-swirl flow generators*) and the twin co-twisted tapes (*CoTs: co-swirl flow generators*) on the heat transfer rate in a uniform heat flux tube are shown in Fig. 6. The results obtained for the tube fitted with the single twisted tape (*ST*), and the empty tube are also plotted for comparison. General trend found in Fig. 6 is that, the Nusselt number increases with the rise of Reynolds number. The influence for applying the *CTs* on heat transfer rate is significant for all Reynolds numbers.

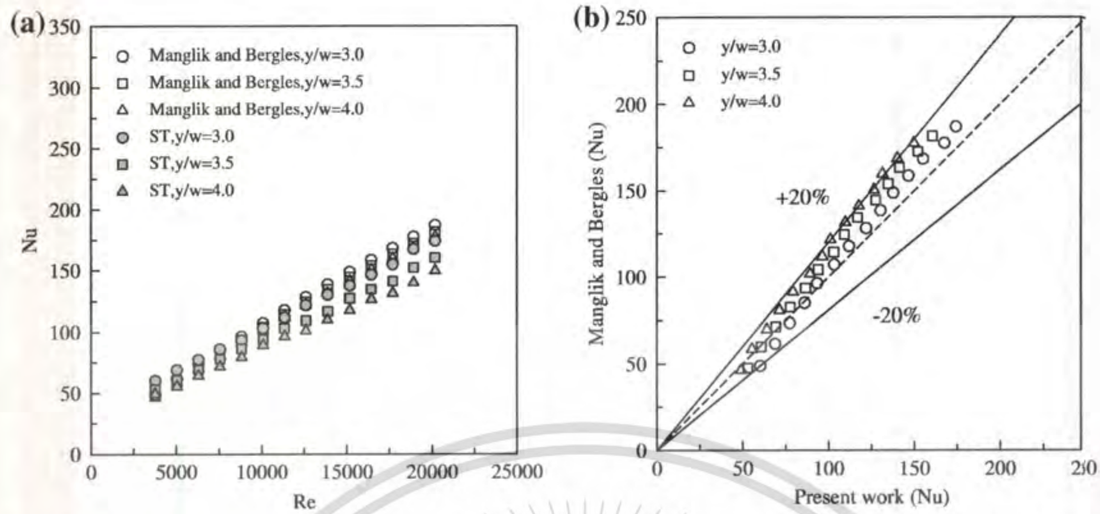

Fig. 4. Confirmatory test of Nusselt number of single twisted tape: (a) *Nu* and *Re* and (b) deviation of the present *Nu* from those by Manglik and Bergles [16].

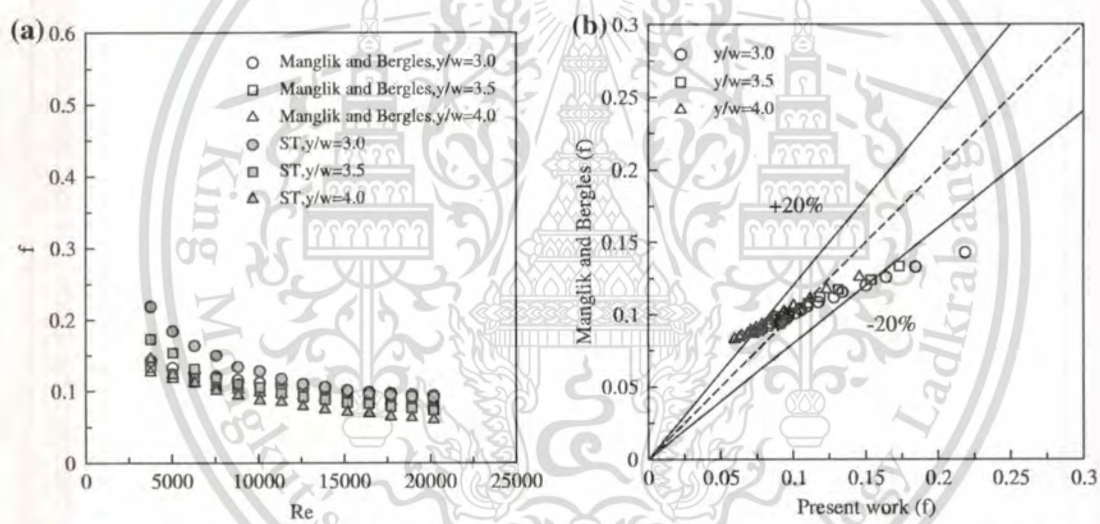

Fig. 5. Confirmatory test of friction factor of single twisted tape: (a) *f* and *Re* and (b) deviation of the present *f* from those by Manglik and Bergles [16].

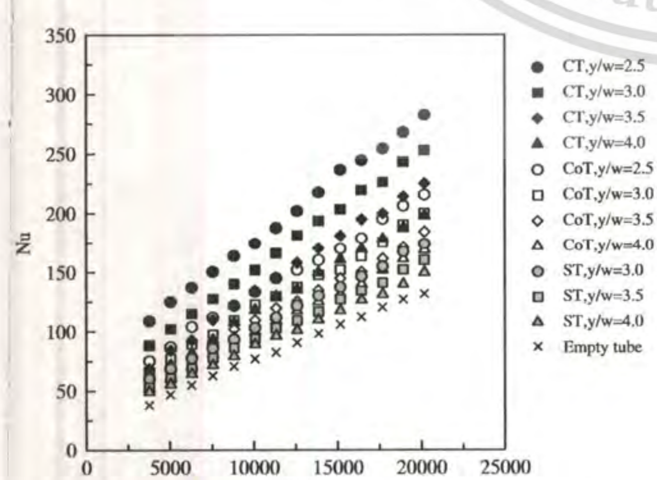

Fig. 6. Variations of Nusselt number with Reynolds number for various twisted tape inserts.

In the present study, the phenomena of the swirling flow in the tube fitted with the *ST*, *CoTs* and *CTs* are demonstrated in Fig. 7a–c using finite volume method [17]. The results are based on the computational conditions as shown in Table 3, for the flow at the constant Reynolds number, $Re = 10{,}000$. Apparently, twin swirl flows are produced for both *CoTs* and *CTs* while only single swirl flow is generated by the *ST*. The higher number of streams generated by both twin tapes is responsible for a better mixing and thus higher heat transfer rate (*Nu*) as experimental results shown in Fig. 6. For all twisted tapes, each stream is induced along each side of tape with axisymmetric profile in direction led by the tape geometry. With the close look for twin co-twisted tapes (*CoTs: co-swirl flow generators*), recirculation zones appear on the top/bottom of vortex core and the generated swirl flows show just small contact zone in the clearance between two twisted tapes, indicating little interaction between them. Therefore, the mixing in the tube equipped with the *CoTs* is dominantly enhanced by rather independent swirl flows. On the other hand, the swirls generated by twin counter twisted tapes (*CTs: counter-swirl flow generators*) are found to be converged in the clearance between two twisted tapes, this apparently provides higher intensity of vortex strength than that gener-

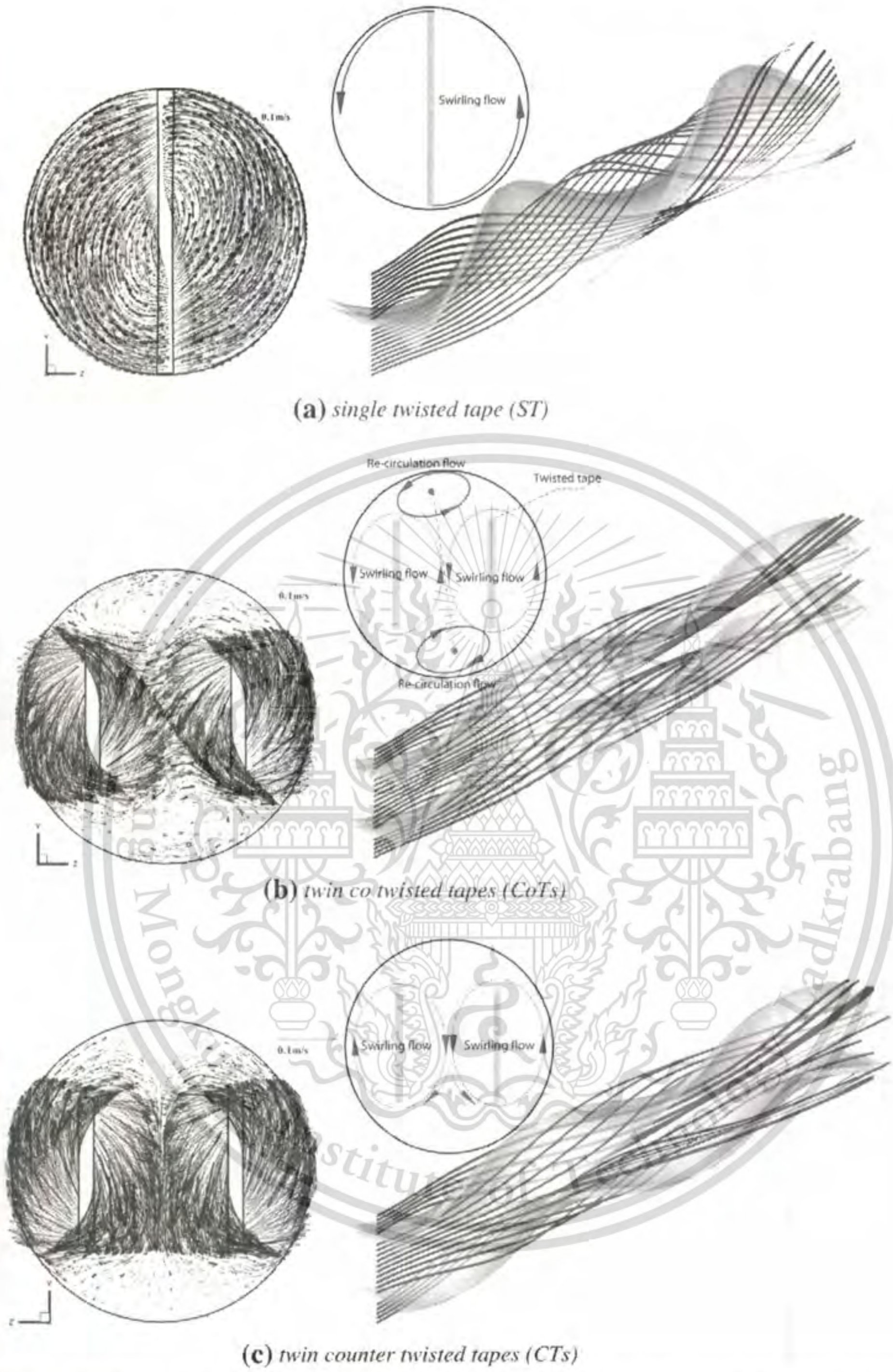

Fig. 7. Path line of flow through a tube fitted with twisted tape: (a) single twisted tape (*ST*), (b) twin co-twisted tapes (*CoTs*) and (c) twin counter twisted tapes (*CTs*).

ated by the *CoTs*. In addition, the recirculation zone is not induced in the tube with the *CTs* as the strong swirl flows impinge to the tube wall. Therefore, it can be addressed herein that the higher interaction between swirl flows (or high vortex strength) induced by the *CTs* leads to superior fluid mixing, resulting in uniform fluid temperature in tubes, and thus more efficient heat transfer in comparison with the *CoTs*. This confirmed by the obtained results in which, the heat transfer rates in the tube fitted with the *CTs* (counter-swirl flow) are noticeably higher than those in the tube fitted with the *CoTs* (co-swirl flow) for all range Reynolds numbers studied (Fig. 6). From the figure, it is also found that the influence of the *CTs* on the heat transfer rate is more intense at higher Reynolds number. Over the range considered, heat transfer rate in the tubes fitted with *CTs* are, respectively, 12.5–44.5% and 17.8–50% higher than those in tubes equipped with the *CoTs* (*co-swirl flows*) and the *ST* (*single swirl flow*).

Fig. 8 presents the friction factor in the corresponding tubes as shown in Fig. 6. Basically, the friction factor decreases with increasing Reynolds number. At the same Reynolds number, the friction factors in the tube fitted with the *CTs* are higher than those in tube fitted with the *CoTs*, tube fitted with the single twisted tape (*ST*) and the empty tube (*non-swirl flow*). This is a consequence of

Table 3
Details of computational conditions.

Twisted tape	*ST*	*CTs*	*CoTs*
(a) Dimensional	3D	3D	3D
(b) Number of grid generation (nodes)	261,495	255,712	254,671
(c) Method	Finite volume	Same as *ST*	Same as *ST*
(d) Turbulence modeling	k–ε model with QUICK	Same as *ST*	Same as *ST*
(e) Flow condition	Isothermal flow	Same as *ST*	Same as *ST*
(f) Fluid	Water	Same as *ST*	Same as *ST*
(g) Inlet temperature	25 °C	Same as *ST*	Same as *ST*
(h) Reynolds number	10,000	Same as *ST*	Same as *ST*
(i) Flow condition	Isothermal flow	Same as *ST*	Same as *ST*

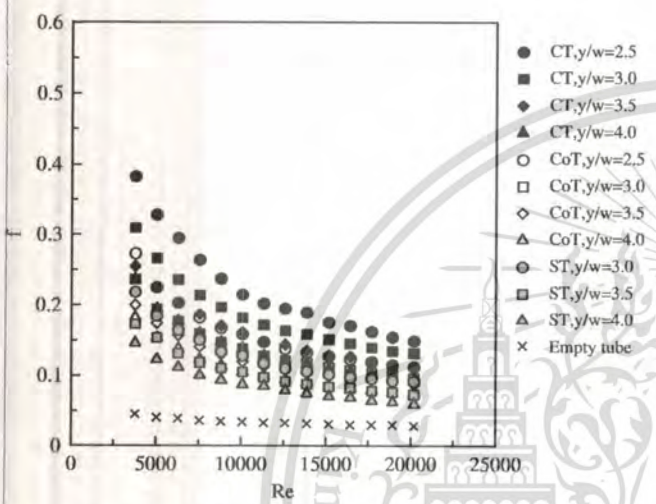

Fig. 8. Variations of friction factor with Reynolds number for various twisted tape inserts.

the repeated acts of strong intensity of vortex in the tube with *CTs*. Over the range studied, the friction factors for tube fitted with *CTs* are higher than those in tube fitted with the *CoTs* around 26.5–45.5%.

5.3. Effect of twist ratios

Effect of twist ratios (y/w = 2.5, 3.0, 3.5 and 4.0) on the heat transfer rate in the tube fitted with *CTs* and *CoTs* is demonstrated in Fig. 6. From the experimental results, it can be observed that heat transfer enhancement increase as twist ratio decreases. In common, the smaller twist ratio generates stronger swirl intensity, leading to more efficient interruption of boundary layer along the flow path. Hence, heat can be transferred efficiently over thin boundary layer. Moreover, the residence time of the flow increases with the increasing swirl flow intensity [7]. This extend the duration of heat transfer between the working fluid and heat source (tube wall). For the entire conditions studied, the *CTs* and *CoTs* with the smallest twist ratio (y/w = 2.5) provide the heat transfer rates around 7.8 to 74.6% and 59.4–187% higher than those for the tube with larger twist ratios and the empty tube, respectively.

As seen in Fig. 8, friction factor tends to increase with decreasing twist ratio. This is in the same trend found for Nusselts number. This can be explained by the fact that the use of a twisted tape with a smaller twist ratio leads to a higher viscous loss near the tube wall regions caused by a stronger swirl flow. Over the range studied, the friction factors for the tube fitted with *CTs* with y/w = 2.5, 3.0, 3.5 and 4.0, are, respectively, 6.37, 5.33, 4.57 and 3.95 times of those in the empty tube.

In the present investigation, the tests are made in a uniform heat flux tube with water as working fluid and correlations are applicable to turbulent region of Reynolds number (Re) between 3700 and 21,000. The empirical correlations from the experimental results of the empty tube fitted with the *ST*, *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 ST:

$$Nu = 0.224Re^{0.66}Pr^{0.4}(y/w)^{-0.6} \tag{11}$$

$$f = 65.4Re^{-0.52}(y/w)^{-1.31} \tag{12}$$

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

$$Nu = 0.473Re^{0.66}Pr^{0.4}(y/w)^{-0.9} \tag{13}$$

$$f = 72.29Re^{-0.53}(y/w)^{-1.01} \tag{14}$$

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

$$Nu = 0.264Re^{0.66}Pr^{0.4}(y/w)^{-0.61} \tag{15}$$

$$f = 41.7Re^{-0.52}(y/w)^{-0.84} \tag{16}$$

Comparisons between the present data with those calculated by the present correlations are for Nusselt number and friction factor portrayed in Figs. 9 and 10. Evidently, the majority of the heat transfer data falls within ±10% for the present correlations of Eqs. (11), (13) and (15). Eqs. (12), (14) and (16) provide the correlative

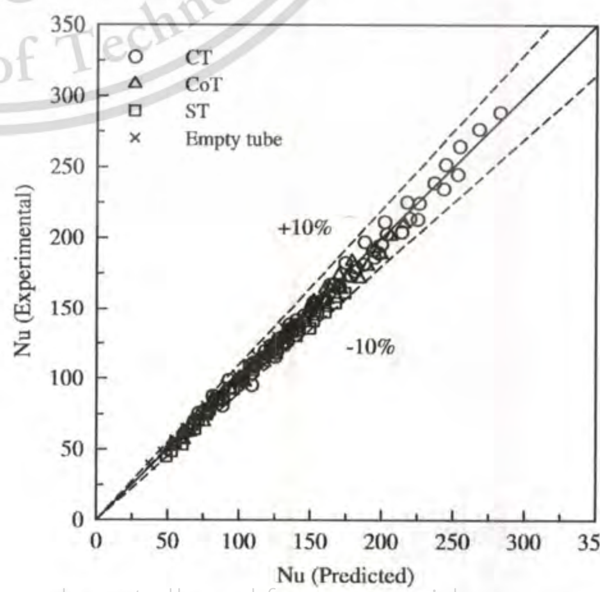

Fig. 9. Validation of empirical correlations for Nusselt number.

results of the friction factor with maximum discrepancies of ±6% with the experimental measurements.

6. Enhancement index

In order to assess the heat transfer enhancement index or thermal enhancement index of the tube fitted with the *CTs* and the *CoTs*, a equal pumping power comparison is performed [18–20]. At the constant pumping power, the relationship between volumetric flow rate and the pressure loss between the empty tube and the tube fitted with the *CTs*/*CoTs* is written as:

$$(\dot{V}\Delta P)_e = (\dot{V}\Delta P)_s \tag{17}$$

and the relationship between friction and Reynolds number is expressed as:

$$(fRe^3)_e = (fRe^3)_s \tag{18}$$

The thermal enhancement index is defined as:

$$\eta = \left.\frac{h_s}{h_e}\right|_{pp} \tag{19}$$

where h_s is the heat transfer coefficient for the empty tube with the *CTs* or *CoTs* insert while h_e is the heat transfer coefficient for the empty tube, at identical pumping power.

Performance evaluation is determined to asses the benefits of using the *CTs* and the *CoTs* in term of enhancement index at the same pumping power. Applying Eqs. (10), (12/14/16) and (18), the Reynolds number for the empty tube (Re_e) can be expressed as the function of the Reynolds number of the tube fitted with twisted tape (Re_s):

For the tube fitted with the ST (Re_s):

$$Re_e = 6.57Re_s^{0.9}(y/w)^{-0.48} \tag{20}$$

For the tube fitted with the CTs (Re_s):

$$Re_e = 6.81Re_s^{0.9}(y/w)^{-0.37} \tag{21}$$

For the tube fitted with the CoTs (Re_s):

$$Re_e = 5.57Re_s^{0.9}(y/w)^{-0.31} \tag{22}$$

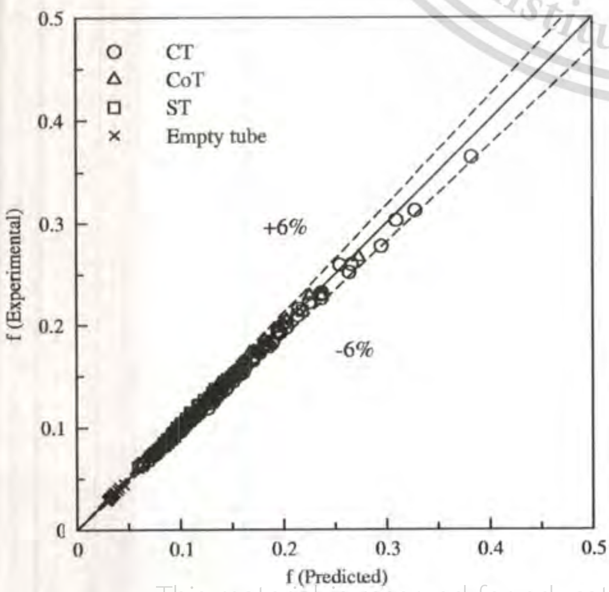

Fig. 10. Validation of empirical correlations for friction factor.

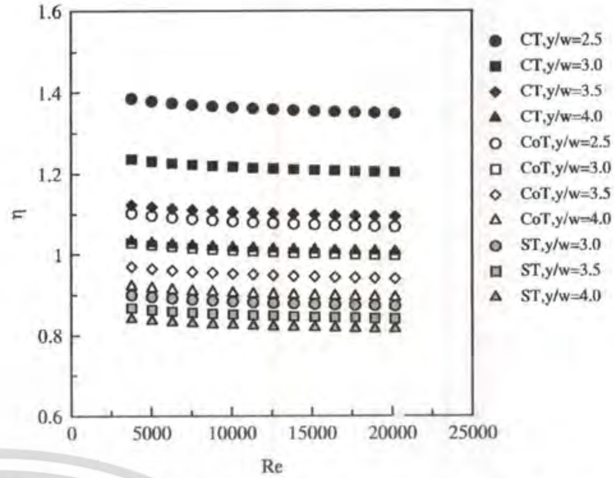

Fig. 11. Variations of thermal enhancement index with Reynolds number for various twisted tape inserts.

Using Eqs. (9), (11), (19) and (20), the thermal enhancement index for tube fitted with *ST* can be expressed as:

$$\eta = 1.365Re_s^{-0.0185}(y/w)^{-0.24} \tag{23}$$

Similarly, the thermal enhancement index for the tube fitted with the *CTs* or *CoTs* can be obtained by the combination of Eqs. (9), (13/15), (19) and (21/22), as follows:

For the tube fitted with the CTs:

$$\eta = 2.8Re_s^{-0.016}(y/w)^{-0.624} \tag{24}$$

For the tube fitted with the CoTs:

$$\eta = 1.82Re_s^{-0.0186}(y/w)^{-0.38} \tag{25}$$

Fig. 11 shows the effect of various single/twin twisted tapes at different twist ratios (y/w) on the thermal enhancement index in Reynolds number range of 3700–21,000. The comparative data reveal that the thermal enhancement index increases with decreasing Reynolds number. In additions, thermal enhancement indices are varied between 1.01 and 1.39 for the *CTs*, 0.89 and 1.1 for the *CoTs*, and 0.81 and 0.9 for the *ST*, depending upon the Reynolds number and the twist ratio (y/w). For the *CTs*, the mean enhancement index for the smallest twist ratio ($y/w = 2.5$) is, respectively, 12%, 23.4% and 34% higher than those for $y/w = 3.0$, 3.5 and 4.0. The above data suggest that the highest enhancement index can be obtained at the low twist ratio (y/w), and relatively low Reynolds number. It is noteworthy that the enhancement indices in the tube fitted with the *CTs* (*counter-swirl tapes*) are around 12–25.7% higher than those in the tube fitted with the *CoTs* (*co-swirl tapes*).

7. Conclusions

The present paper shows the feasibility of convection heat transfer enhancement by inserting the *CTs* (*counter-swirl flow generators*) and the *CoTs* (*co-swirl flow generators*) for turbulent flow with a uniform heat flux. Heat transfer coefficient and friction factor are determined experimentally for the tube fitted with the *CTs*, *CoTs* and *ST* using water as working fluid with Reynolds number ranging from 3700 to 21,000. Based on the obtained results, key findings of this investigation can be summarized as follows:

(I) For general observation, it is found that heat transfer, friction factor, and thermal enhancement index increase as the twist ratio (y/w) decreases. In addition, the Nusselt number increases with increasing Reynolds number while the opposite trends are found for the friction factor and the thermal enhancement index.

(II) The *CTs* (*counter-swirl tapes*) can enhance heat transfer more efficiently than the *CoTs* (*co-swirl tapes*). The quantitative results show that, heat transfer rates for the *CTs* are around 12.5–44.5% higher than those for the *CoTs* and around 17.8–50% higher than those for the *ST*. The maximum thermal enhancement indices (η) obtained at the constant pumping power by using 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, while those obtained by using the *CoTs* with the same range of y/w are 1.1, 1.03, 0.97 and 0.92. The achieved results indicate that the modified swirl flow (*counter-swirl flow*) generated by the *CTs* is a promising approach for heat transfer enhancement.

(III) The empirical correlations developed in the present work, provide the correlative results of the Nusselt number and friction factor. The maximum discrepancies between the correlative results and experimental results for Nusselt number and friction factor are found to be ±10% and ±6%, respectively.

Acknowledgement

The first author would like to gratefully acknowledge the Thailand Research Fund (*TRF*) for the financial support of this research.

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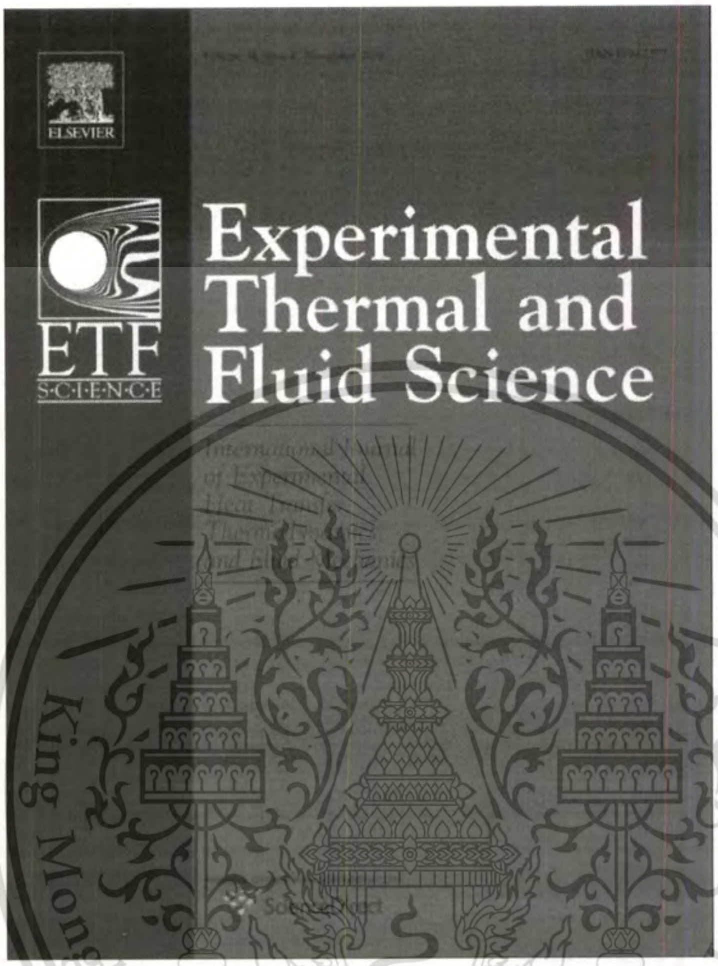
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Experimental Thermal and Fluid Science

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Thermohydraulic investigation of turbulent flow through a round tube equipped with twisted tapes consisting of centre wings and alternate-axes

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ARTICLE INFO

Article history:
Received 4 January 2010
Received in revised form 19 April 2010
Accepted 19 April 2010

Keywords:
Enhanced heat transfer
Twisted tape
Wing
Alternate axis
Swirl flow
Vortex

ABSTRACT

The effects of the twisted tapes consisting of centre wings and alternate-axes (WT-A) on thermohydraulic properties in a round tube, were investigated. The effects of other three types of twisted tapes including: (1) the twisted tape with wings alone (WT), (2) the twisted tape with alternate axes alone (T-A), and (3) the typical twisted tape (TT), were also studied for comparison. All twisted tapes used were twisted at constant twist length (y) of 57 mm, corresponding to a constant twist ratio (y/W) of 3.0. The wings were generated along the centre line of the tape with three different angles of attack, ($\beta = 43°$, $53°$ and $74°$). Test runs were conducted using water as a testing fluid with Reynolds number range between 5200 and 22,000. Under the similar condition, the heat transfer rate in the tube fitted with the WT-A was consistently higher than those in the tube equipped the WT, T-A and plain tube. It is also found that the heat transfer rate increased with increasing angle of attack. Over the range studied, the use of WT-A at $\beta = 74°$ was found to be the most effective for heat transfer enhancement, giving thermal performance factor of up to 1.4. Mean values of Nusselt number (Nu), friction factor (f), thermal performance factor (η) provided by the WT-A (at $\beta = 74°$) were respectively, 17.7%, 30.6% and 7.8% higher than those in the tube with WT (at $\beta = 74°$), 20.8%, 53% and 4.9% higher than those in the tube with T-A, and 62%, 123% and 24% higher than those in the tube with TT. The superior performance of the WT-A over those of the other tapes could be attributed to the combined effects of the following actions: (1) a common swirling flow by the twisted tape (2) a vortex generated by the wing (3) a strong collision of the recombined streams behind each alternate point. For a better understanding on flow phenomena, flow-visualization by smoke wire technique is also presented. In addition, the experimental correlations of Nusselt number, friction factor and thermal performance factor were also developed.

1. Introduction

Enhancement of heat transfer in a heat exchanger is widely applied in industries due to the need of more compact heat exchanger, a lower operating cost, energy saving as well as ecological benefit. Among many heat transfer enhancement techniques, utilization of twisted tape and deta-wing/delta-winglet vortex generators is a promising method. The approach possesses not only an effective heat transfer enhancement but also the advantage of a low cost and an ease of installation.

Twisted tapes have been extensively used as heat transfer enhancing devices in heat exchangers. The important effects induced by the tapes are: (1) swirl flow which improves fluid mixing, (2) helically twisting fluid motion which offers an effectively longer flow path, and (3) partitioning and blockage of the tube flow cross section which leads to a higher flow velocity [1]. All the effects mentioned above are directly responsible for the improvement of heat transfer within heat exchanger. The performances of twisted tape swirl generator have been intensively investigated by many researchers [1–14] for different tape geometries (e.g. broken, serrated, delta-wing, perforated, notched and jagged), working fluid types (e.g. water, air, oil, servotherm medium oil, turbine oil, ethylene glycol, nitrogen, and R134a), wall conditions (uniform heat flux and constant wall temperature) and flow regimes (laminar, transition and turbulent flows). Regarding to the results reported in the mentioned papers, it can be observed that the shape of twisted tape is an important key signifying the tape performance, for example serrated twisted tape [3], the modified tape with protrusive part could improve heat transfer rate significantly, however the tape caused an unavoidably dramatic increase of friction within system and this prohibits its practical applications. On

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doi:10.1016/j.expthermflusci.2010.04.004

Nomenclature

A	heat transfer surface area, m^2
C_p	specific heat of fluid, J kg^{-1} K^{-1}
D	inside diameter of the test tube, m
f	friction factor = $\Delta P/((L/D)(\rho U^2/2))$
h	heat transfer coefficient, W m^{-2} K^{-1}
I	current, A
k	thermal conductivity of fluid, W m^{-1} K^{-1}
L	length of the test section, m
M	mass flow rate, kg s^{-1}
Nu	Nusselt number = hD/k
P	pressure of flow in stationary tube, Pa
ΔP	pressure drop, Pa
Pr	Prandtl number = $\mu C_p/k$
Q	heat transfer rate, W
Re	Reynolds number = $\rho UD/\mu$
t	thickness of the test tube, m
T	temperature, K
$\tilde{T}_s$	mean temperature, K
U	mean axial flow velocity, m s^{-1}
V	voltage, V
W	twisted tape width, m
Y	twisted tape pitch, m

Greek symbols

β	angle of attack, degree
ρ	fluid density, kg m^{-3}
δ	twisted tape thickness, m
μ	fluid dynamic viscosity, kg s^{-1} m^{-1}
η	thermal performance factor

Subscripts

b	bulk
c	convection
i	inlet
o	outlet
p	plain
s	surface
t	turbulator
w	water

Abbreviations

TT	typical twisted tape
T-A	twisted tape with alternate-axes
WT	twisted tape with centre wings
WT-A	twisted tape with centre wings and alternate-axes

the other hand, the twisted tapes with space such as perforated, notched [2], and regularly-spaced [6] twisted tapes generated lower friction loss but their heat transfer enhancement and thus thermal performance factor are unattractive. Differently, the twisted tapes consisting of protrusive parts together with spaces on the tapes such as broken [4] and delta-wing twisted tapes [8] offered promising outcomes in both viewpoints of heat transfer enhancement and friction loss since the protrusion effectively improved mixing while the space brought down the friction to satisfactory level. This resulted in thermal performance factor above unity which is beneficial for energy saving and hence made the twisted tape suitable for industrial applications.

Delta-wing and delta-winglet vortex generators were used for heat transfer augmentation by many researchers [15–23]. Tiwari et al. [15] and Fiebig [16] described that vortices enhanced the mixing of fluid in the periphery region with that in the core region of the flow field by the swirling motion as a secondary flow. This effect led to the destabilization of primary flow and thermal boundary layer thinning, and thus heat transfer improvement. The numerical investigation by Biswas et al. [17] and the experimental finding by Valencia et al. [18] indicated that the proper use of vortex generators resulted in an effective heat transfer augmentation with low or moderate pressure drop penalty. Vortex generators can be incorporated into a plate, a fin surface or a channel with an attack angle by means of punching, embossing, stamping, or attachment process [19]. There are two types of vortices: transverse vortex and longitudinal vortex. The axis of a transverse vortex lies perpendicular to the flow direction while that of the longitudinal vortex (also called streamwise vortex) is parallel to the main flow. Practically, longitudinal vortex and transverse vortex are induced simultaneously and the dominance of the one over the another depends on the attack angle. The longitudinal vortex is governing at small attack angle whereas the transverse vortex becomes more influential with increasing attack angle [20]. This issue gains significance in design of delta-wing for specific applications [21]. It was also mentioned that a longitudinal vortex offered less friction loss and more efficient heat transfer than a transverse vortex [20,22]. Regarding to results by Fiebig [23], delta-wings were more efficient than rectangular wings, and punched wings were slightly better than mounted wings for the same vortex generator area.

Most of the delta-wing and delta-winglet vortex generator appear in a flat plate. In our previous report [8], the delta-winglet was located on the periphery of twisted tape. Our results indicated that under the similar operating conditions, the Nusselts numbers in the test tube equipped with peripheral delta-winglet twisted tape were considerably higher than those in the tube equipped with typical twisted tape. Besides, our evaluation revealed that the thermal performance factor in the tube with the modified twisted tape was enhanced up to 1.24. This points out that the presence of the winglet on twisted tape gives a promising result for heat transfer enhancement.

In present work, the modified twisted tape was alternatively designed by generating wings on the centre line of the twisted tape and then forming alternate axes (twisted tape with wings and alternate axes, WT-A). The modification has been assumed to induce the combined effects of the common swirl flow by the twisted tape, vortex by wings and also additional flow fluctuation by alternate axes, leading to excellent results for heat transfer enhancement. The effects of other three types of twisted tapes including (1) the twisted tapes with wings alone (WT), (2) the twisted tape with alternate axes alone (T-A), and (3) the typical twisted tape (TT), were also studied for comparison. The experiments were conducted to investigate the effects of twisted tapes used on the heat transfer rate, friction factor as well as thermal performance factor in a turbulent tube flow with Reynolds number between 5200 and 22,000 using water as a working fluid.

2. Wing twisted tape with/without alternate axis

All twisted tapes used in the present work, were made of aluminum strips with thickness (δ) of 0.8 mm, width (W) of 19 mm and length (L) of 1000 mm. The twist ratio y/W defined as ratio of twist length (y, 180°/ twist length) to tape width of twisted tapes (W), was kept constant at 3.0. All tapes were initially formed as typical twisted tapes. The twisted tape with centre wings (WT) was modified from the typical twisted tape (TT) by punching delta-wings on the TT along the centre line (one delta-wing per twist length). The adjacent wings were open to the different sides of the twisted tape (left to right and vice versa) as shown in Fig. 1a and b. The attack angle of the wing (β) was varied at 43°, 53° and 74°. Another modified twisted tape, twisted tape with

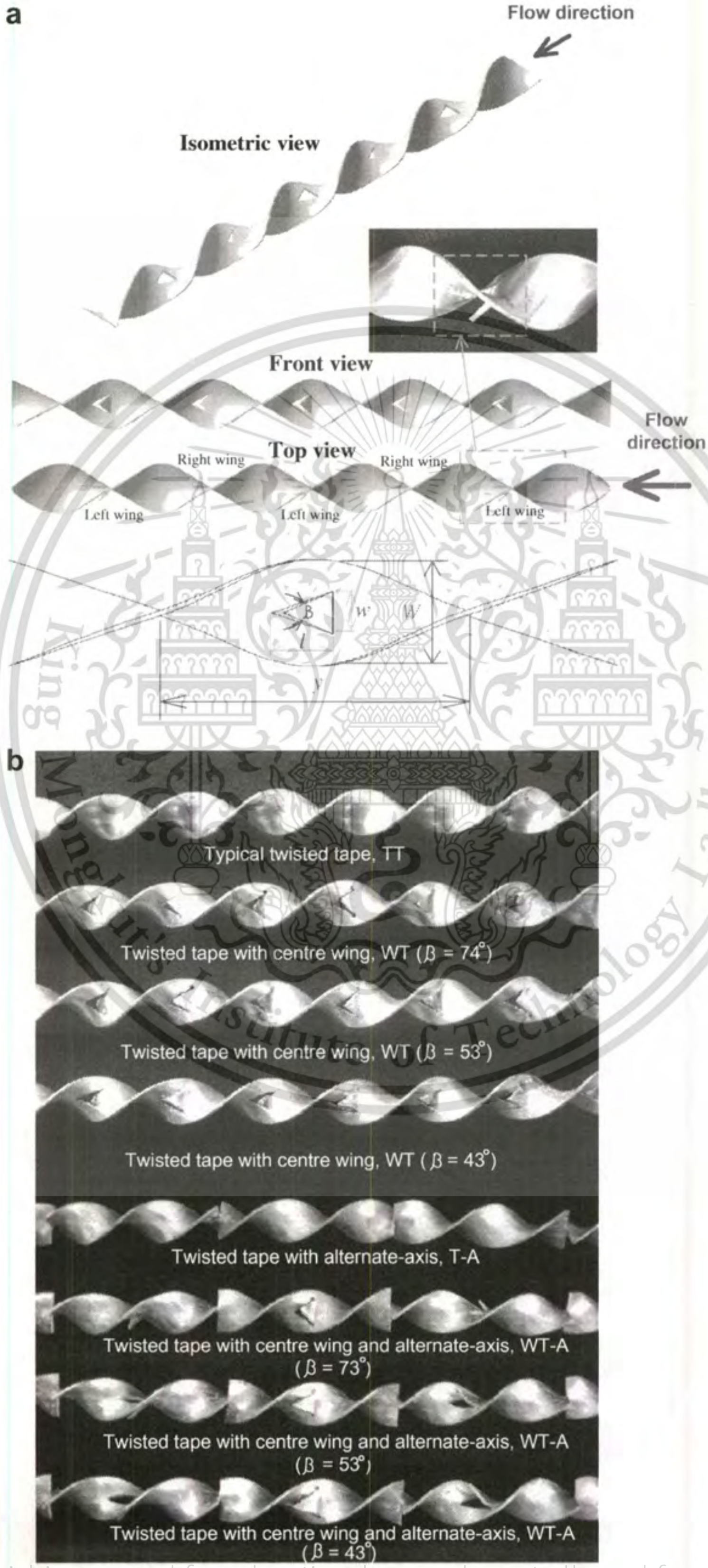

Fig. 1. (a)Geometry of twisted tapes. (b)Photograph of twisted tapes.

Table 1
The ranges of operation parameters.

Test tube conditions	
Reynolds number	5200–22,000
Inlet water temperature	28 °C
Inner tube diameter, D	19.5 mm
Outer tube diameter	22.5 mm
Tube thickness, t	1.5 mm
Test tube material	Copper
Twisted tape	
Twist ratio, y/W	3.0
Tape width, W	19 mm
Twist length, y	57 mm
Twist tape length, L	1000 mm
Tape thickness, δ	0.8 mm
Angle of attack, β	43°, 53° and 74° (for WT and WT-A)
Width of wing, w	6, 9 and 11.5 mm
Length of wing, l	9 mm
Tape material	Aluminum

alternate axes alone (T-A) was formed by cutting on both sides of the TT, with the depth of cut of 4 mm on each side. Subsequently, the tape was twisted to angle difference of 90° to produce an alternate axis. The distance between each pair of alternate points was set at two twist lengths. The last modified twisted tape, the twisted tape with centre wings and alternate axes (WT-A) was formed by creating wings on the TT (one wing per two twist lengths) and then generating alternate axis in the middle between each pair of the wings. Note that the consequence of generating the alternate points on the WT-A, made wings appeared on four sides of the tape (left–right–top–bottom). The photograph of all tapes used in the present work, including one TT, one T-A, three different WT and three different WT-A, is shown in Fig. 1b and their dimensions are also summarized in Table 1.

3. Experimental facility

3.1. Heat transfer set-up

An apparatus used in the present work, is shown in Fig. 2a and b. The heat transfer set-up facility consists of: (1) water reservoir (2) measuring and recording devices (3) a calm section and a test section which were made from copper. The length and inner diameter of the calm section were respectively, 1500 mm and 19.5 mm while those of the test section were respectively 1000 mm and 19.5 mm. The dimension of the calm section met a requirement for fully developed turbulent flow in which $L/D \approx 75$. In the experiment, a testing fluid (water) was continuously supplied from a water reservoir at ground level to an overhead tank. Water was directed from the overhead tank to the calm section and then the test section. The test tube was spirally wounded by electrical heating wire with a constant wire pitch on its outer surface. Due to the sufficient thermal insulation over the test section, only around 3–7% of total heat provided by heating wire was lost from test tube during the test. Heat supplied via the electrical heating wire, was controlled by the variac transformer. Local surface wall temperatures (T_s) at fifteen different axial stations were measured using K-type thermocouples while the inlet and outlet temperatures of cold water and hot water were measured using two RTDs. All temperature data were collected by data logger (Yokogawa: FX122-4-2) and directly transferred to a personal computer.

The inlet water (at temperature of about 28 °C) flow rate was set to the desired value varying from 4 to 16 L/min. This range corresponded to a range of Reynolds number from 5200 to 22,000. Pressure drop across the test section were measured via pressure taps which connected to manometer and located approximately 50 mm upstream and 150 mm downstream of the test section. Note that the pressure drop was measured under an isothermal flow condition without switching on the heater. All data were recorded after attaining steady state. More details of the experimental set-up are described elsewhere [8]. The test tube conditions are summarized in Table 1.

3.2. Flow-Visualization set-up

To gain a better understanding on flow phenomena induced by each twisted tape, the flow visualization was conducted. The smoke wire flow-visualization set-up is schematically illustrated in Fig. 2b. The facility consists of (1) a copper tungsten wire with diameter of 0.2 mm (2) a paraffin oil dropper (as smoke source), (3) a twisted tape insert, (4) a light bulb to lighten the test section, (5) a voltage supplier, (6) a digital camera (Nikon D80), (7) an inverter, (8) a hot wire anemometer, (9) a blower and test section made from acrylic tube, and (10) a personal computer. The twisted tape was made from the steel with width of 35 mm, and the tape was twisted at constant twist ratio (y/W) of 4.0. In the experiment, the paraffin oil was dropped on the copper tungsten wire. Then, heat was supplied to the oil through the voltage supplier to generate smoke and air was fed into the tunnel as a smoke carrier. The velocity of air in the tunnel was measured using a hot wire anemometer and air flow rate was kept constant, corresponding to the constant Reynolds number of 2500.

4. Data reduction

In the experiment, electrical power input (Q_{VI}) was supplied to the tube wall via heating wire:

$$Q_{VI} = V \times I \tag{1}$$

Due to heat loss from the test section to surrounding, the net heat transfer rate from the inner tube surface to water (Q_w) at the steady-state condition can be expressed as

$$Q_w = Q_{VI} - Q_{loss} = MC_{p,w}(T_o - T_i) \tag{2}$$

where Q_{loss} is the rate of heat loss, $M = \rho U(\pi D^2/4)$ is the mass flow rate of water through the test tube, T_i and T_o is the inlet and outlet temperatures of the working fluid, respectively. According to the heat balance at the steady state flow condition, Q_{loss} was found in between 3% and 7% of the input electrical power (Q_{VI}).

It is assumed that the rate of heat gained by water is equal to the rate of convective heat transfer (Q_c) as

$$Q_c = Q_w \tag{3}$$

The convective heat transfer is defined in term of the nominal inside surface area as

$$Q_c = hA(\tilde{T}_s - T_b) \tag{4}$$

where T_b is bulk flow temperature which was calculated from $T_b = (T_i + T_o)/2$. $\tilde{T}_s$ is the mean inside tube-wall temperature of the test section which was computed by taking the average temperatures of 15 axial stations as

$$\tilde{T}_s = \sum (T_{s1} + T_{s2} + \cdots + T_{s15})/15 \tag{5}$$

The mean heat transfer coefficient h was then evaluated from the following equation:

$$h = MC_{p,w}(T_o - T_i)/A(\tilde{T}_s - T_b) \tag{6}$$

It should be mentioned that the water thermo physical properties were evaluated at the bulk flow temperature (T_b).

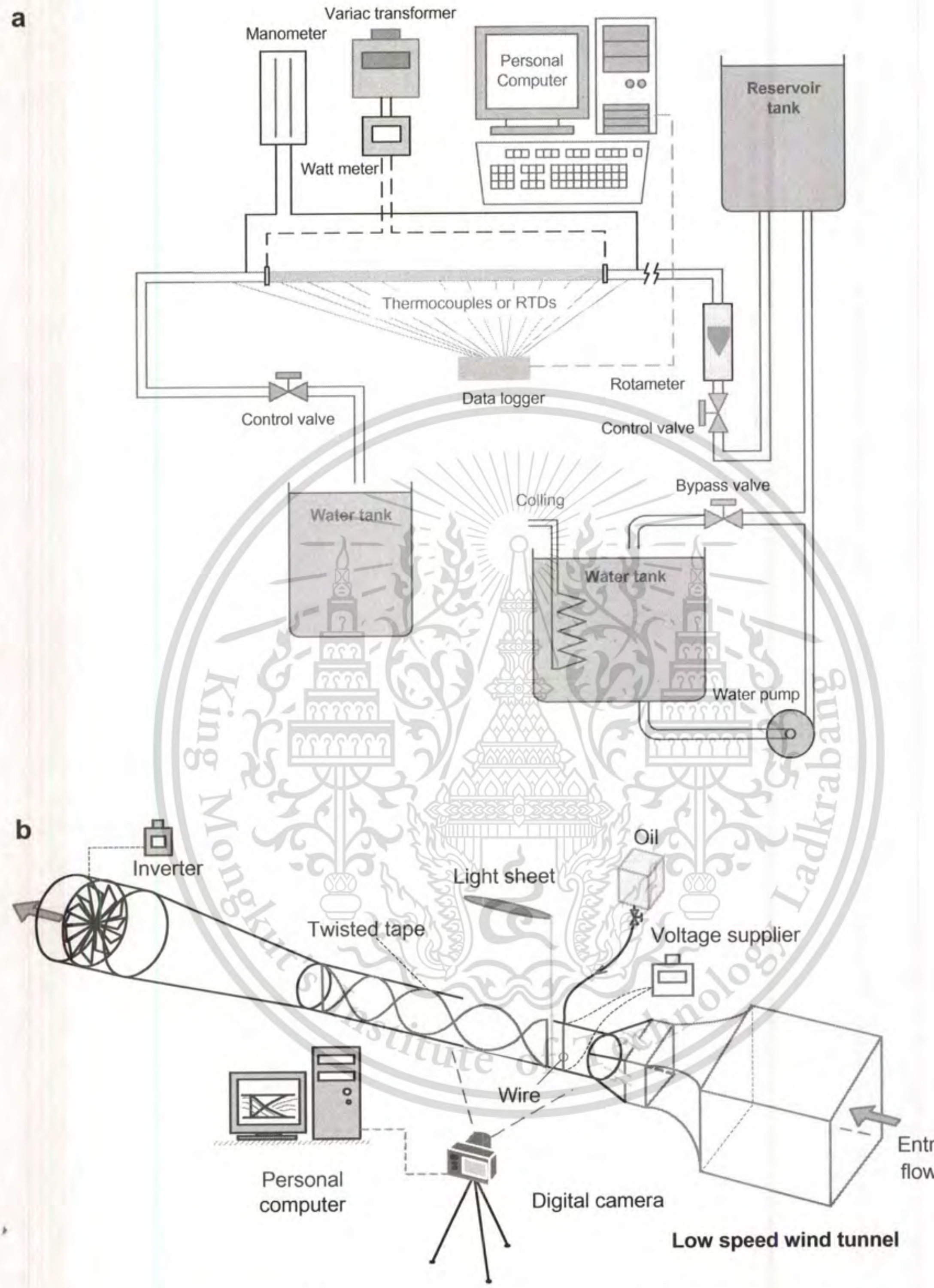

Fig. 2. Schematic diagram of experimental apparatus: (a) heat transfer set-up and (b) smoke wire set-up.

The heat transfer coefficient inside the test tube is reported in term of mean Nusselt number which can be written as

$$\mathrm{Nu} = hD/k \tag{7}$$

Another important parameter characterizing the fluid flow through the tube is the friction factor

$$f = 2(\Delta PD)/(\rho U^2 L)) \tag{8}$$

where ΔP is pressure drop across the test section which was evaluated under an isothermal flow condition. Reynolds number stated in the present work is based on the average flow inlet velocity and the tube inlet diameter as

$$\mathrm{Re} = UD/\nu \tag{9}$$

Finally, thermal performance factor at constant pumping power which is a factor for evaluating a potential of a twisted tape for actual application, can be expressed as

$$\eta = (Nu_t/Nu_p)/(f_t/f_p)^{1/3} \quad (10)$$

In the present experiment, the evaluations for experimental uncertainties of the present data reduction process were also performed by following the method described elsewhere [24]. The maximum uncertainties for Nu, Re and f were found at about 8%, 5% and 7%, respectively.

5. Experimental results

The experimental results including Nusselt number (Nu), friction factor (f) and the thermal performance factor (η) behaviors in the tube with different twisted tapes (TT, T-A, WT, and TW-A) are presented and discussed in the following subsections.

5.1. Validation test

To evaluate the reliability and precision the present experimental facility, Nusselt number and friction factor in the present plain tube were taken and then compared with those in the previously published correlations from the open literature including, Gnielinski correlation and Dittus and Boelter correlation for the Nusselt number and Petukhov correlation and Blasius correlation for the friction factor [25]. In addition, the Nusselt numbers and friction factors in the tube equipped with the typical twisted tape were also verified with those reported by Manglik and Bergles [1].

Fig. 3a–c presents the comparison between the measured data (Nusselt number, friction factor and thermal performance factor) and those calculated from the previous correlations [1,25]. For the plain tube, the present mean Nusselt number was respectively, 5% and 3% lower than those from Gnielinski equation and Dittus & Boelter equation while the present mean friction factor data was respectively 9% and 10% higher than those of Blasius equation and Petukhov equation. For the tube fitted with typical twisted tape, the present mean Nusselt number was 7% lower and friction factor was 9% higher than those from Manglik and Bergles equation. According to the comparative results mentioned above, the reliability of the present facility is considered acceptable. Results of the thermal performance factor (η) of the present plain tube fitted with typical twisted tape (TT) are compared with those of the

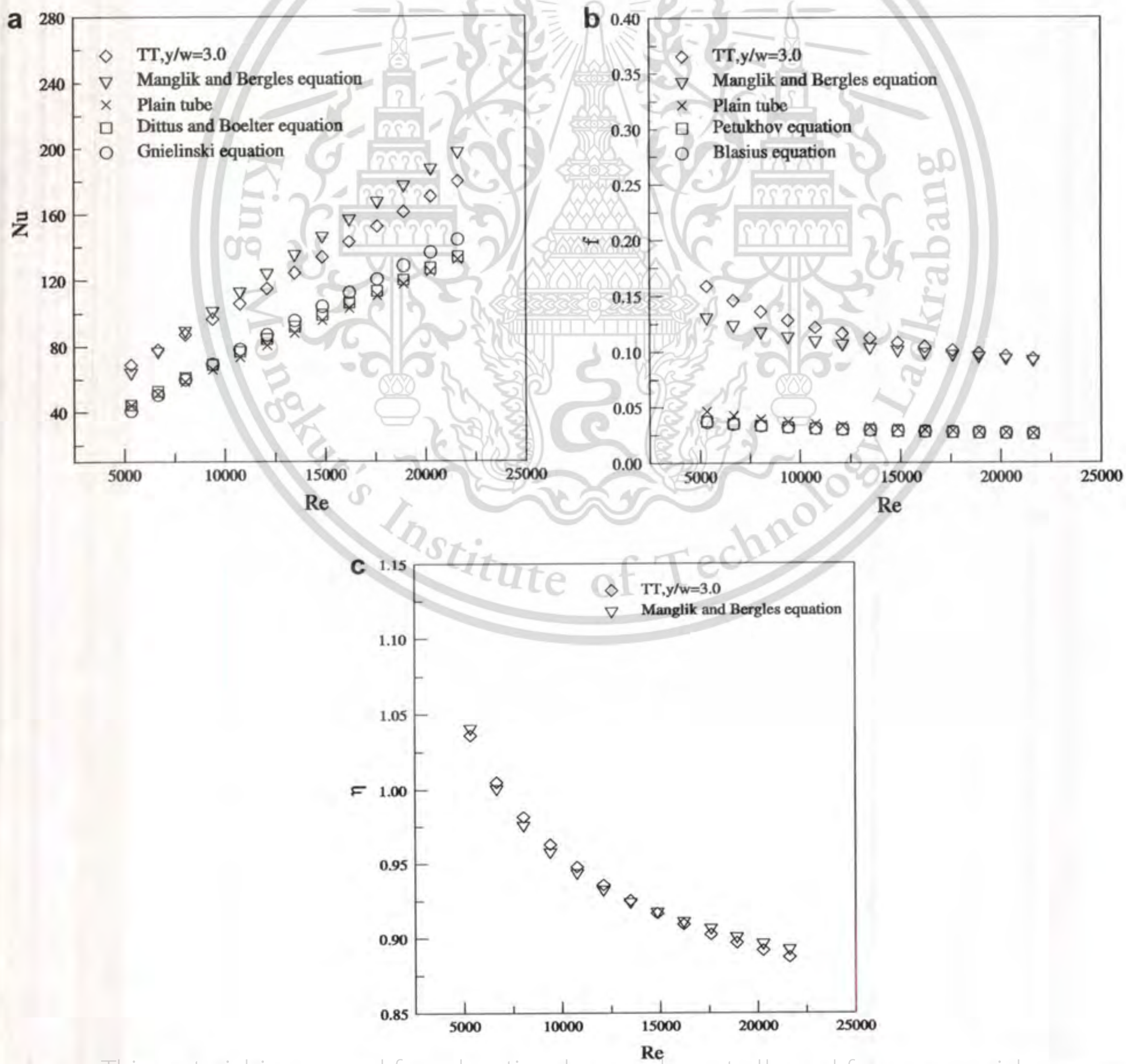

Fig. 3. Validation of the plain tube with/without typical twisted tape inserts for: (a) Nusselt number, (b) friction factor and (c) thermal performance factor.

previous correlations [1,25] by using Eq. (10). For the previous studies, the values of the Nusselt number (Nu_t) and friction factor (f_t) of the tube fitted with TT were calculated from the Manglik and Bergles correlation while the Nusselt number (Nu_p) and friction factor (f_p) values of the plain tube were calculated using correlations of Gnielinski and Petukhov, respectively. It is obvious that the present results of thermal performance factor reasonably agree well with the previous work.

5.2. Effect of wing twisted tape

5.2.1. Heat transfer results

The comparison of Nusselt numbers of the tube with typical twisted tape (TT) and twisted tapes with centre wings (WT) are demonstrated along with those of the plain tube in Fig. 4. At the similar operating conditions, Nusselt numbers (Nu) in the tube with WT were found to be consistently higher than those in the tube with TT and plain tube. Depending on Reynolds number and angle of attack the Nusselt numbers in the tube with WT, were 16% to 52% and 58% to 138% higher than those in the tube with TT and the plain tube, respectively. The superior performance of the WT compared to that of the TT, can be attributed the combined effect of common swirl flow induce by the twisted tape and the vortex generated by the centre wings in the WT. This can be supported by the fact that the vortex provides the additional disturbance into the typical swirl stream generated by the twisted tape. This action directly improves heat transfer rate with respect to that offered by the swirl flow alone.

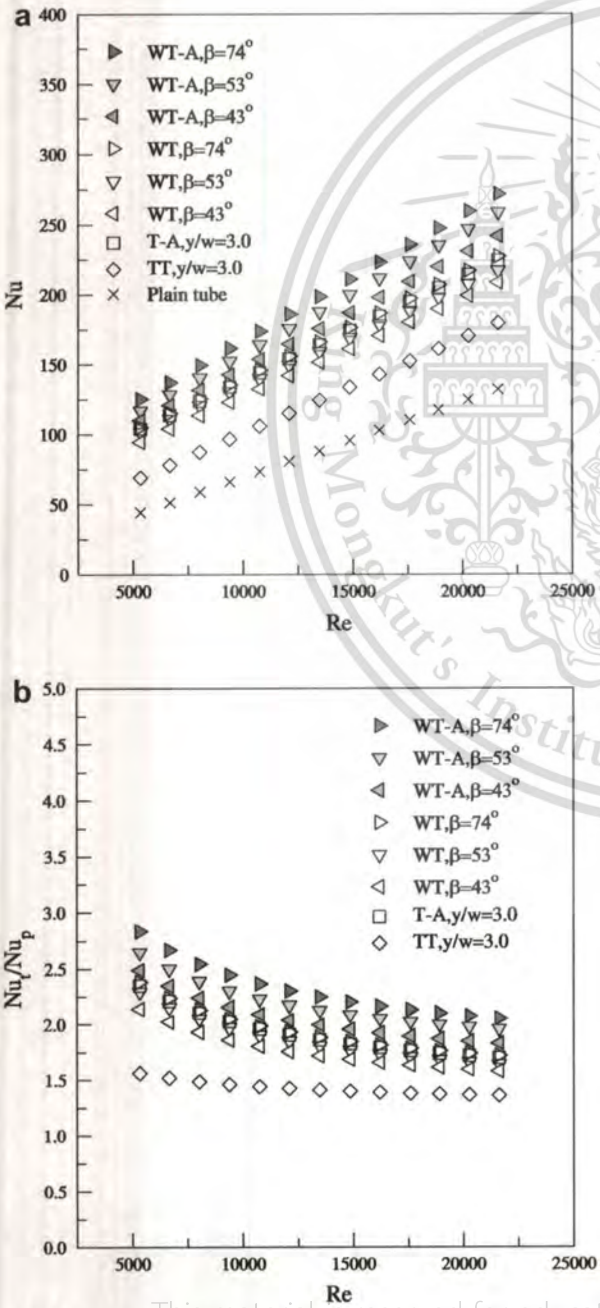

Fig. 4. Effect of twisted tape on Nusselt number: (Nu) and (Nu_t/Nu_p).

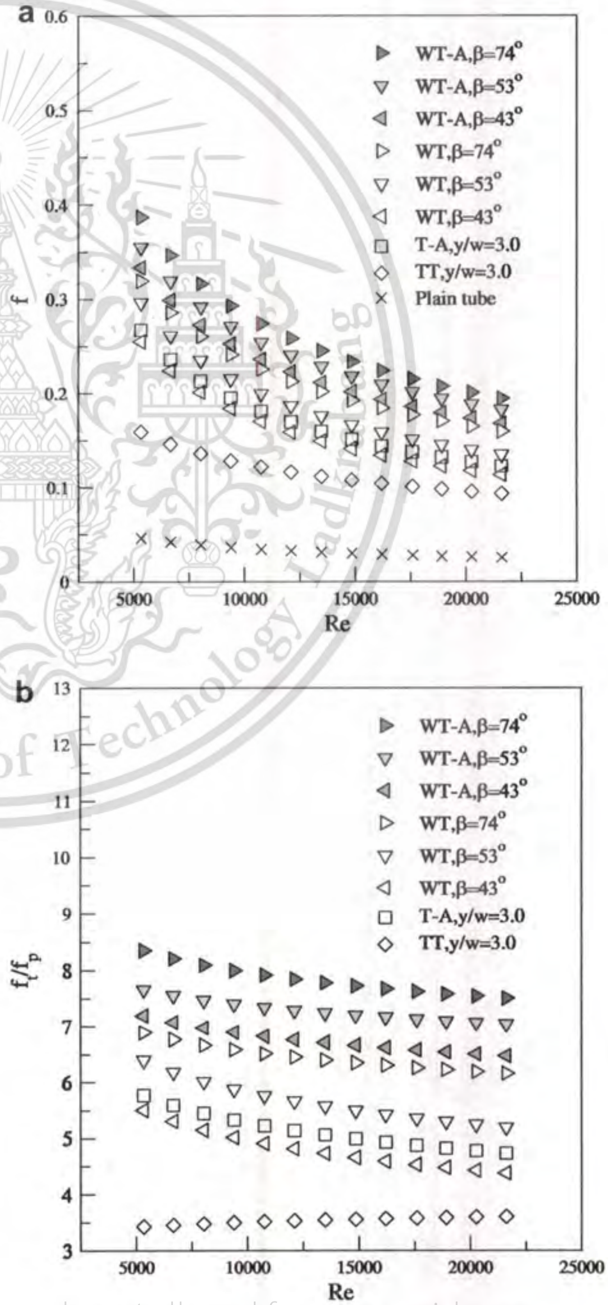

Fig. 5. Effect of twisted tape on friction factor: (f) and (f_t/f_p).

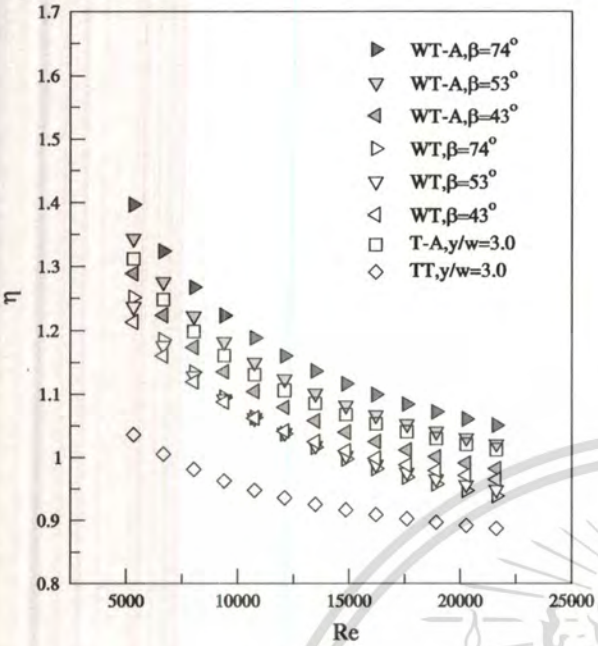

Fig. 6. Effect of twisted tape on thermal performance factor.

5.2.2. Friction factor results

The experimental results for friction factor in the tube fitted with the TT and the WT as well as a plain tube under an isothermal flow condition are shown in Fig. 5. Apparently, the friction factors in the tube fitted with the twisted tape consisting of centre wings (WT) were found to be higher than those in the tube with the typical twisted tape around 21.5–101% and higher than those in plain tube from 4.4 to 6.9 times. This is due to an extra blockage by the wings appears on the WT and the additional forces exerted by a vortex flow, giving rise to a more pronounced dissipation of the dynamical pressure of a working fluid.

5.3. Effect of angle of attack

5.3.1. Heat transfer results

The effect of angle of attack (β) on heat transfer characteristics in the tube fitted with the WT is also reported in Fig. 4. The result revealed that Nusselt number (Nu) increased with the increase of angle of attack. Over the range studied, the WT with the largest angle of attack ($\beta = 74°$) provided the highest heat transfer rate. The maximum increase in Nusselt number obtained in the tube with this tape was 11%, 4.7% and 138% compared to those in the tube with the WT at angles of attack, $\beta = 43°$ and 53° and the plain tube, respectively. This can be explained that the wing with the larger

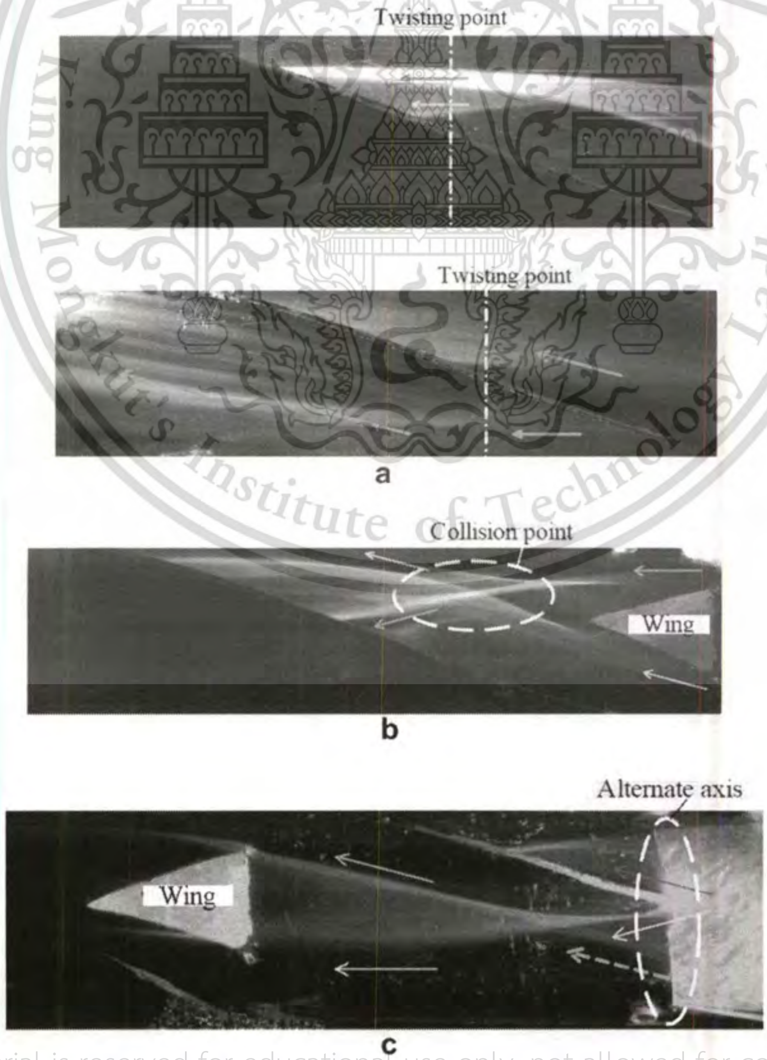

Fig. 7. Visualization of flow through tube with twisted tape inserts by smoke wire technique: (a) TT, (b) WT and (c) WT-A.

attack angle gives the more efficient mixing which provides more disruption to a thermal boundary layer, leading to a better heat transfer.

5.3.2. Friction factor results

Fig. 5 presents the effect of angle of attack (β) on friction factor which was taken under an isothermal condition. As expected, under the similar conditions, friction factor increased with the increasing angle of attack. The mean friction factors in the tube with the WT at angles of attack, $\beta = 43°$, $53°$ and $74°$ were respectively 4.8, 5.6 and 6.4 times of those in the plain tube and 1.4, 1.6 and 1.8 times of those in the tube with the TT. This is simply attributed to a greater flow blockage by the centre wings at the larger angle of attack.

5.4. Effect of an alternate axis and a combination of wing and alternate axis

Determination from Figs. 4–6, it is found that under the similar conditions, the heat transfer coefficient, friction factor and thermal performance factor in the tube fitted with the WT-A were consistently higher than those in the tube equipped the WT, T-A and plain tube. The heat transfer enhancement (HTE) data is also presented in term of Nusselt number ratio (Nu_t/Nu_p) which is plotted against the Reynolds number as shown in Fig. 4b. Obviously, the HTE ratio is high at low Reynolds number and then decreased with increasing Reynolds number and became nearly constant at high Reynolds number. Fig. 5b shows the variation of the friction factor ratio (f_t/f_p) with the Reynolds number value for various twisted tapes. The trend with Reynolds number is found to be similar to that for the HTE ratio. Over the range investigated, WT-A with $\beta = 74°$ gave maximum thermal performance factor of 1.4 at Reynolds number of 5200. Mean values of Nusselt number (Nu), friction factor (f), thermal performance factor (η) provided by WT-A with $\beta = 74°$ were respectively, 17.7%, 30.6% and 7.8% higher than those in the tube with WT-A with the same angle of attack, 20.8%, 53% and 4.9% higher than those in the tube with T-A, and 62%, 123% and 24% higher than those in the tube with TT.

The visualization of flow by smoke wire technique through a tube with various twisted tape inserts, relating to their performance mentioned above, is presented in Fig. 7a–c. Evidently, the TT induced the swirl flow as depicted in Fig. 7a. The swirl flow is a basic flow commonly generated by any twisted tape. The photograph of fluid flowed through the tube with WT in Fig. 7b demonstrated that fluid stream was split into two streams and then recombined behind the wing, producing double longitudinal vortices. The flow behavior caused by the alternate axis in the WT-A is presented in Fig. 7c. According to the tape geometry, two streams of fluid (in front and back sides of tape) were split into four streams around the alternate axis. Then they recombined into two streams with the stream collision behind the alternate point. The flow caused by the centre wing induced by the WT-A is similar to that induced by the WT as shown in Fig. 7b.

The sketch of flow phenomena induced by each twisted tape is demonstrated in Fig. 8, which are: (1) the TT generates only single direction-swirl flow, (2) the WT provides a single-direction-swirl flow caused by the twisted tape together with vortices induced by the wings (3) the T-A generates double-direction-swirl flows and (4) a WT-A offers combined actions from both WT and T-A twisted tape. As mentioned in Section 5.2, the wings on WT appear on one side/twist ratio (left or right) of the tape while those on WT-A appear on two sides (left and right) as a consequence of the generating the alternate axis. Therefore, the additional flow disturbance by the WT-A over that by the WT is not only caused by the periodic change of swirl direction but also a better distribution of the wings in the tube. As described above, the WT-A thus gives a better fluid mixing and superior heat transfer rates as well as thermal performance factor over the other twisted tapes do. The thermal performance factor above unity offered by the WT-A indicates the potential of the WT-A in view point of pumping energy saving.

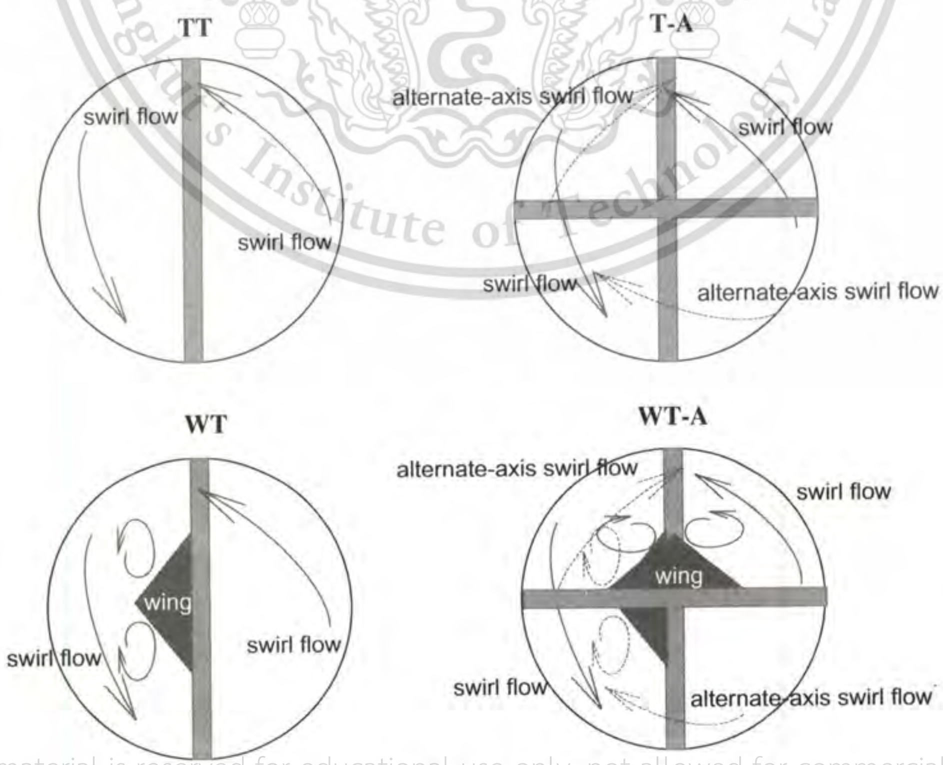

Fig. 8. Sketch of flow phenomena in the front view of the tube with various twisted tapes.

Table 2
Correlations of Nusselt number, friction factor and thermal performance factor.

Nusselt number	
Plain tube	$Nu = 0.025Re^{0.791}Pr^{0.4}$
Twisted tape with centre wings	$Nu = 0.232Re^{0.595}Pr^{0.4}(1+\tan\beta)^{0.202}$
Twisted tape with centre wings and alternate axes	$Nu = 0.385Re^{0.568}Pr^{0.4}(1+\tan\beta)^{0.129}$
Friction factor	
Plain tube	$f = 1.645Re^{-0.416}$
Twisted tape with centre wings	$f = 14.039Re^{-0.505}(1+\tan\beta)^{0.406}$
Twisted tape with centre wings and alternate axes	$f = 20.445Re^{-0.504}(1+\tan\beta)^{0.283}$
Thermal performance factor	
Twisted tape with centre wings	$\eta = 4.629Re^{-0.166}(1+\tan\beta)^{0.067}$
Twisted tape with centre wings and alternate axes	$\eta = 6.772Re^{-0.194}(1+\tan\beta)^{0.035}$

From the experimental results above, a least-squares regression was utilized to yield the equations summarized in Table 2. These equations are expressed in terms of Reynolds number (Re), Prandtl number (Pr), and angle of attack (β). Fig. 9a–c shows a comparison of experimental values of Nusselt number (Nu), friction factor (f) and thermal performance factor (η) with the predicted values, obtained from those equations. Apparently, the predicted data were in good agreement with the experimental data, within ±10% (WT) and ±6% (WT-A) for Nusselt number, ±10% (WT) and ±9% (WT-A) for friction factor and ±7% (WT) and ±4% (WT-A) for thermal performance, respectively.

6. Conclusions

The present study explored the effects of the wing and alternate axis of the modified twisted tapes on the heat transfer and fluid friction characteristics in a heat exchanger tube. For comparison, the twisted tapes used in the present work included one TT, one T-A, three different WTs and three different WTs-A. The angle of attack was varied at 43°, 53° and 74°. The main findings in the present work are concluded as follows.

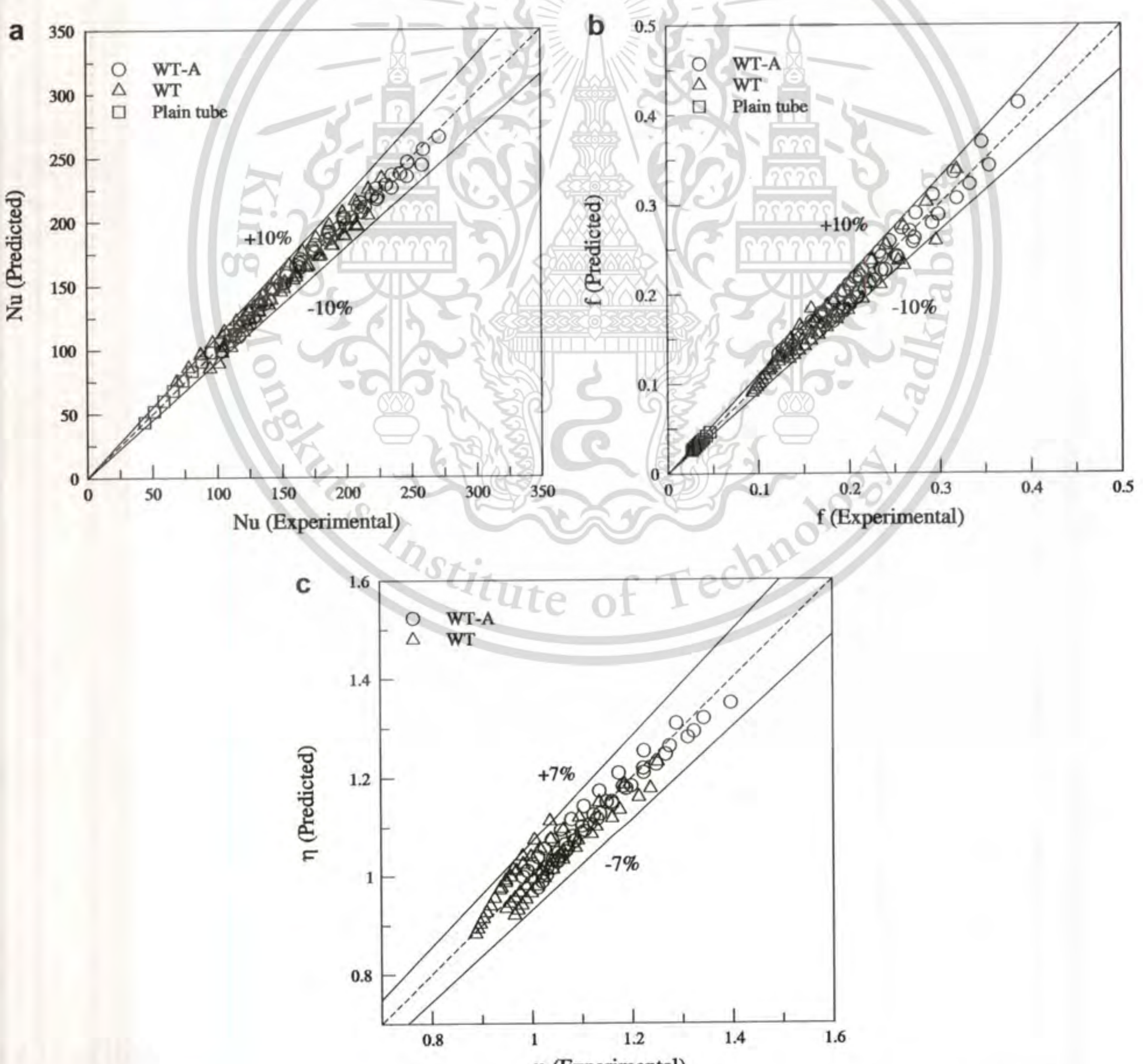

Fig. 9. Experimental data versus predicted data for: (a) Nusselt number and (b) friction factor and (c) thermal performance factor.

1. The combined actions of the wing and alternate axis in the WT-A resulted in a better fluid mixing and thus heat transfer enhancement compared to those induced by wing alone (WT) or alternate axis alone (T-A). The results also revealed that Nusselt number (Nu) increased with the increasing angle of attack.
2. The WT-A generated higher pressure drop within the tube than WT, T-A and also TT, due to the flow disturbance caused by both the wing and the alternate axis and pressure drop became larger as the angle of attack increased.
3. The Nusselt number ratios (Nu_t/Nu_p) were in the range of 1.6–2.4 and 1.8–2.8 while the friction factor ratios (f_t/f_p) were in range of 4.4–6.9 and 6.48–8.4 for the WT and the WT-As, respectively.
4. The similar trend also found for the effects of the twisted tape form and the angle of attack on thermal performance factor (η) which is the consequential results from both heat transfer rate and pressure drop.
5. In the present range, the WT-A with the largest angler of attack ($\beta = 74°$) gave the highest Nusselt number (Nu), friction factor (f) as well as thermal performance factor (η). Mean values of these three parameters provided by the WT-A with ($\beta = 74°$ were respectively, 17.7%, 30.6% and 7.8% higher than those in the tube with WT, 20.8%, 53% and 4.9% higher than those in the tube with T-A, and 62%, 123% and 24% higher than those in the tube with TT.
6. The predicted data obtained from the developed correlations for Nusselt number, friction factor and thermal performance factor were in good agreement with the experimental data.

Acknowledgment

The financial support of this work by the Thailand Research Fund (TRF) and the Thailand Toray Science Foundation (TTSF) is gratefully appreciated.

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Turbulent Heat Transfer in a Circular Tube Fitted with Twisted-Tape Swirl Generator

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Abstract—***This paper describes heat transfer enhancement associated with twisted tape (TT) swirl generator in a circular tube. The influences of twist ratio (y/W = 3, 4 and 5) on heat transfer enhancement, friction factor and thermal performance factor characteristics are reported for the Reynolds number between 5500 and 20200. Heat transfer enhancement is evaluated by comparing the present experimental results with the results of present plain tube (non-swirl flow) and also those obtained from standard correlations. Evidently, the tube with TT consistently possesses higher heat transfer and friction factor than the plain tube. In addition, the tape with the smaller twist ratio gives higher heat transfer rate, friction factor as well as thermal enhancement factor than the one with larger twist ratio as a result of a larger contact surface area, longer residence time, stronger swirl intensity and thus better fluid mixing which leads to a thinner thermal boundary layer. In the range determined, the tubes with TTs at twist ratios, y/W = 3, 4, 5 respectively give Nusselt number over that of the plain tube by around 37.3%, 27.5% and 19.9%, which are accompanied with friction loss penalties over that of the plain tube by around 230%, 194% and 170%, respectively.***

***Index Terms*— Heat Transfer Enhancement, Heat Exchanger, Twisted Tape, Turbulence.**

I. Introduction

For almost a century, the swirl flow devices have been used for enhancing heat transfer rate in heat exchangers. It is regarded that the swirl tangential velocity increases the composite velocity, thins the boundary layer, enhances the tangential and radial turbulent fluctuation, and therefore causes the increase in pressure drop and heat transfer inside a tube. Generally, the swirl flow in pipes can be classified into two major types: the continuous swirl flow and the decaying swirl flow. In continuous swirl flow, the swirling motion persists over the whole length of the pipe while in decaying swirl flow, the swirl is generated at the entry section of the pipe and decays along the flow path. It is well known that the twisted tape is one of the most widely used devices for producing compact heat exchangers and upgrading the thermal performance of the existing heat exchanger due to its low cost and ease of manufacture installation [1-3]. Twisted tape brings several mechanisms for heat transfer enhancement including: (1) augmenting flow velocities caused by partial blockage of the tube flow cross section (2) promoting heat transfer coefficient as a result of hydraulic diameter reduction (3) lengthening flow path in consequence of a helically twisting fluid motion (4) improving fluid mixing by secondary fluid motion and (5) possible fin effect by a metallic tape with a good contact with a tube wall. Regarding to its promising characteristics, twisted tape inserts have been applied in several researches for heat transfer improvement [4-7].

The work aims to improve heat exchanger performance by introducing secondary swirl flow into the primary air flow. This paper deals with a preliminary study on heat transfer enhancement, friction factor and thermal performance factor characteristics in a heat exchanger with twisted tape swirl generators. The effect the twist ratio on the thermal performance factor behaviors is also examined. In the experiments, twisted-tapes at three different twist ratios (y/W = 3, 4 and 5) were inserted along the test section. Test runs were carried out under uniform heat flux for Reynolds numbers between 5500 and 20200, using air as the working fluid.

II. Twisted Tape

Twisted tapes (TTs) were made of aluminium sheet with tape width (W) of 19 mm, tape thickness (δ) of 0.8 mm. and tape length (l) of 1000 mm. Twisted tapes (TT) were prepared with three different twist ratios, y/W=3, 4 and 5 where twist ratio is defined as twist length (180°/twist length) to tape width (W). They were fabricated by twisting straight tapes, about their longitudinal axis, while being held under tension. Figure 1 demonstrates the twisted tapes at different twist ratios (y/W) and their arrangements.

III. Experimental Setup

The experiments were carried out using an experimental facility as shown in Fig. 2. The test tube was made of copper with inner diameter of 19.5 mm (D), outside diameter of 21 mm (D_o), wall thickness of 1.5 mm (t), and length of 1000 mm (L). In the experiments, the twisted tapes were inserted at the core tube along the test section. The rather long tube provided sufficient contact surface between the tapes and tube wall for the firm attachment of the tapes to the tube without the need of any extra fitting. The tube was wound with electrical heating wire covered with ceramic beads. During the test, the tube was heated by continually winding flexible electrical wire providing a uniform heat flux condition. The electrical output power was controlled by a variac transformer to obtain a constant heat flux along the entire length of the test section and by keeping the current less than 10 A. The outer surface of the test tube was well insulated to minimize convective heat leak to surroundings. Moreover, necessary precautions were taken to prevent leakages from the system. In the experiment, the heat transfer losses from the test tube were found to be around 3 to 5% of the total heat input ($Q = IV$). The inner and outer temperatures of the water were measured at certain points with a multi-channel temperature measurement unit in conjunction with the resistance temperature detectors (RTDs). Fifteen thermocouples were tapped on the local wall of the tube and the thermocouples were placed round the tube to measure the circumferential temperature variation, which was found to be negligible. The mean local wall temperature was determined by means of calculations based on the reading of Copper-Constantan thermocouples.

The test loop consists of a water pump, data logger, pressure transmitter, thermocouple/RTD, rotameter, and heat transfer test section as depicted in Fig. 2. In the apparatus setting above, the inlet water from a reservoir tank was discharged through the rotameter to measure the volumetric flow rate and passed to the heat transfer test section. For each test run, it was necessary to record the data of temperature, volumetric flow rate and pressure drop of the bulk water at steady state conditions in which the inlet water temperature was maintained constant at 27°C. The evaluations of Reynolds number and friction factor were based on bulk mean temperatures.

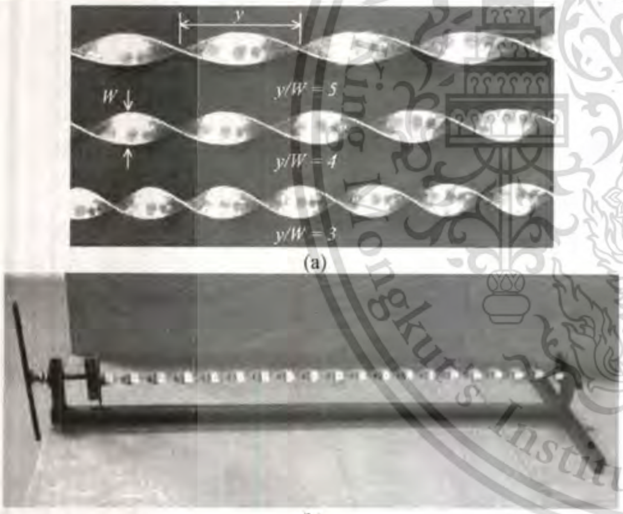

Figure 1. Twisted tapes at different twist ratios (y/W) and and their arrangements

IV. Data reduction

In the present work, the water was used as a working fluid in a uniform heat flux tube. The steady state of the heat transfer rate is assumed to be equal to the convective heat transfer from the test section which can be expressed as:

$$Q_{water} = Q_{conv} \tag{1}$$

in which

$$Q_{water} = \dot{m}C_{p,water}(T_o - T_i) = VI \tag{2}$$

The convective heat transfer from the test section can be written as:

$$Q_{conv} = hA(\widetilde{T}_w - T_b) \tag{3}$$

whereas,

$$T_b = (T_o + T_i)/2 \tag{4}$$

and

$$\widetilde{T}_w = \sum T_w / 15 \tag{5}$$

where T_w is the local wall temperature at the outer surface of the inner tube. The averaged wall temperatures are calculated from 15 points, lined between the inlet and the exit of the test pipe. The averaged heat transfer coefficient, h and the mean Nusselt number, Nu are estimated as follows:

$$h = \dot{m}C_{p,water}(T_o - T_i) / A(\widetilde{T}_w - T_b) \tag{6}$$

$$Nu = hD/k \tag{7}$$

The Reynolds number is given by

$$Re = UD/\nu \tag{8}$$

Friction factor, f can be written as:

$$f = \frac{\Delta P}{\left(\frac{L}{D}\right)\left(\rho\frac{U^2}{2}\right)} \tag{9}$$

in which U is mean velocity of the tube.

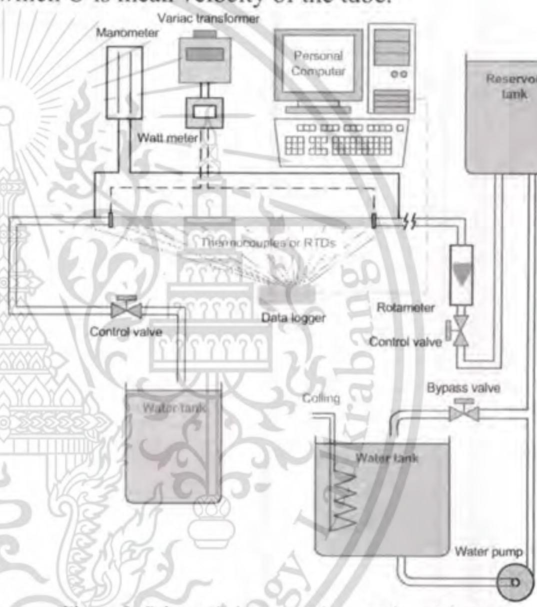

Figure 2. Schematic heat transfer experimental setup

The thermal performance factor of the tube fitted with TT is evaluated by comparison the results of the tube with and without twisted tape under an equivalent pumping power. The equivalent pumping power constraint for the plain tube and the tube fitted with TT can be expressed as:

$$(\dot{V}\Delta P)_t = (\dot{V}\Delta P)_p \tag{10}$$

where index t denotes tape and index p denotes plain tube. Another equation for the same constraint can be expressed as:

$$(f\,Re^3)_t = (f\,Re^3)_p \tag{11}$$

The thermal performance factor is defined as:

$$\eta = \left.\frac{h_t}{h_p}\right|_{pp} \tag{12}$$

Where h_t is the heat transfer coefficient for the tube with TT insert while h_p is the heat transfer coefficient for the plain tube, at equivalent pumping power. These equations result in the explicit form for thermal performance factor at equivalent pumping power which indicates the potential of twisted tape for the actual applications, in the following equation.

$$\eta = (Nu_t/Nu_p)/(f_t/f_p)^{1/3} \qquad (13)$$

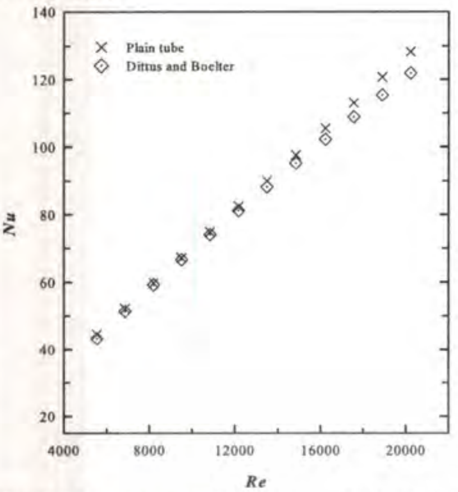

Figure 3. Verification of Nusselt number for plain tube

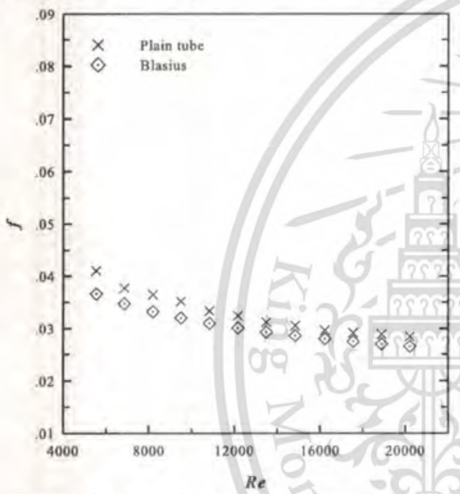

Figure 4. Verification of friction factor for plain tube

V. Results and Discussion

A. *Validation of plain tube*

To ensure reliability of the experimental facility and measurement, the results of the present plain tube, were validated with the previous correlations of Dittus-Boelter and of Blasius [8] for the fully developed turbulent flow. Figures 3 and 4 show the comparisons between the present results those obtained from the correlations reported in the previous work. In the figures, the present friction factors are within ±7.9% of those from Blasius correlation while the present Nusselt numbers are within ±2.5% of those obtained from Dittus and Boelter correlation. Determine from the validation, it can be concluded that the present data are sufficiently accurate. Therefore, the facility was subjected to study on heat transfer and friction factor in a tube with twisted tapes.

B. *Heat transfer results*

Figure 5 demonstrates the variation of the heat transfer rate (Nu) with the Reynolds numbers in the tube with/without twisted tape (TT) swirl generators. For all cases, Nusselt number (or heat transfer rate) tends to increase with incresing Reynolds number, as a result of an increased turbulence intensity. At given Reynolds number, the tubes with TTs, consistently possess superior heat transfer to the one without TT (the plain one). The augmented heat transfer is directly responsible by the continuous swirl flow generated by a twisted tape. The swirl flow repeatedly disrupts thermal boundary layer along the whole test length, resulting in consistent thermal boundary thinning and thus more efficient heat transfer through the tube wall. It is also found that Nusselt number increases as twist ratio (y/W) decreases (or twist number increases). This primarily caused by the promoted swirl violation, enlargement of fluid travel distance and also tangential contact area between a fluid and tube wall. The experimental results reveal that the mean Nusselt numbers attained by the uses of the tubes with TTs at y/W = 3, 4 and 5 are respectively 37.3%, 27.5% and 19.9%, higher than that of the plian tube. In the other word, the TT with smaller (y/W = 3) yields higher heat transfer rate than the ones with greater twist ratios (y/W = 4 and 5) by around 7.7% and 14.5%, respectively. It is noteworthy that for the range investigated, the increased percentage of Nusselt number is larger at higher Reynolds number.

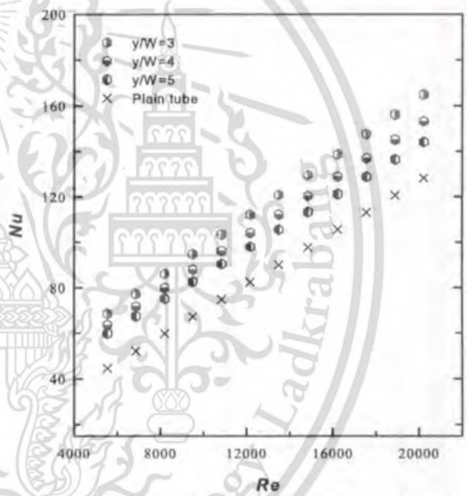

Figure 5. Effect of twist ratio on heat transfer rate

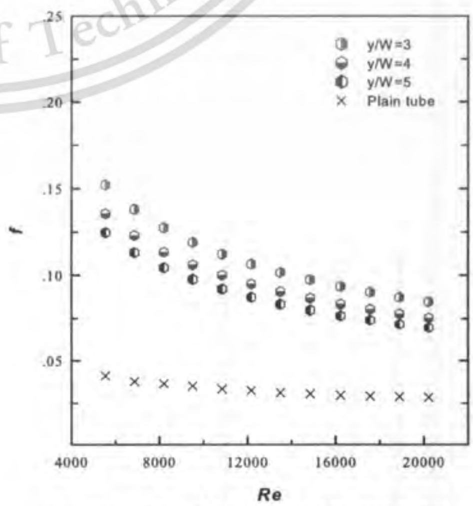

Figure 6. Effect of twist ratio on friction factor

C. *Friction factor results*

Figure 6 presents the relationship between the friction factor and Reynolds number for tubes with/without twisted tape insert. For the tubes with all TTs, friction factor is diminished as Reynolds number increases, which is in manner similar to that of the plain tube. This is basically due to the dissipation of dynamic pressure caused by tube blockage and promoted interaction between the pressure forces and inertial forces in the boundary layer. It is also found that TT with small twist ratio induces higher friction factor than the one with large twist ratio. For the present range, friction factors associated by the TTs at y/W = 3, 4 and 5 are respectively 230%, 194% and 170%, higher than that of the plian tube.

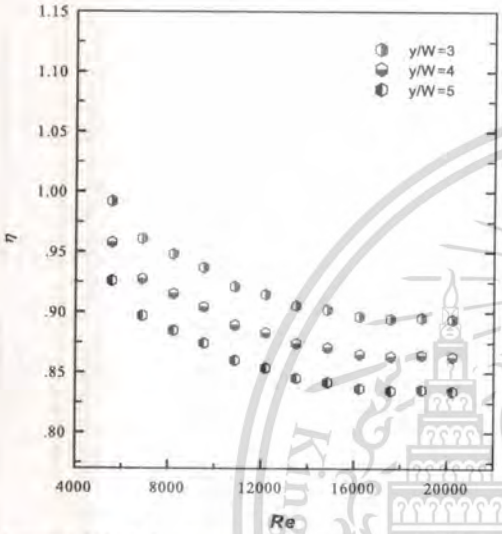

Figure 7. Effect of twist ratio on thermal performance factor

D. *Thermal performance results*

Figure 7 presents the thermal enhancement factor for air flowing in the heat exchanger tube with twisted tape inserts with different twist ratios. Obviously, the effect of Reynolds number on thermal enhancement factor is primarily governed by that of friction factor since thermal enhancement factor tends to decrease with the rise of Reynolds number. In contrast, the effect of twist ratio is influential by that of Nusselt number. The twisted tape with the smallest twist ratio y/W = 3 offers the highest thermal enhancement factor while that with the largest twist ratio y/W = 5 yields the lowest one. The maximum thermal performance factors provided by the tapes with twist ratios, y/W = 3, 4 and 5, are found to be 0.99, 0.96 and 0.92, respectively.

VI. Conclusion

The twisted tape (TTs) twist ratio (y/W = 3, 4 and 5) have been utilized in circular tube to enhance heat transfer for the Reynolds number between 5500 and 20200, under uniform heat flux condition. With the presence of twist tape, heat transfer and friction factor are promoted over those found in the plain tube, case. Heat transfer rate, friction factor and thermal enhancement factor increase tape twist ratio decreases. In the range determined, the tubes with TTs at twist ratios, y/W = 3, 4, 5 give Nusselt number over that of the plain tube by around 37.3%, 27.5% and 19.9%, respectively which are accompanied with friction loss penalties over that of the plain tube by around 230%, 194% and 170%, respectively. The highest thermal enhancement factor of 0.99 is obtained by utilizing the tape with the smallest twist ratio (y/W = 3) at Reynolds number of 5500.

VII. Acknowledgement

Financial support from the Energy Policy and Planning Office, Ministry of Energy, Thailand (EPPO) for this research is greatly acknowledged.

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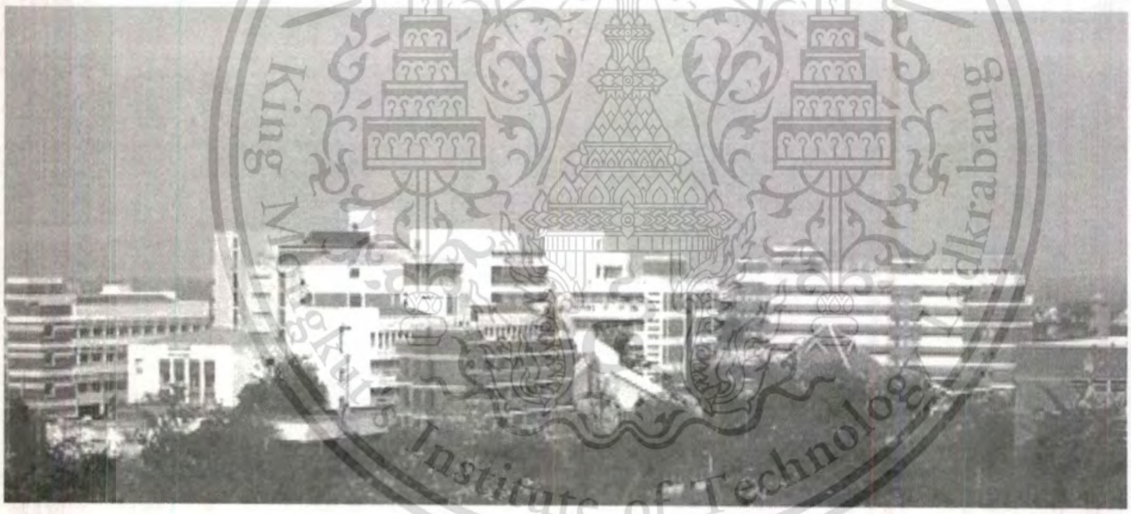

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Heat Transfer Enhancement in a Heat Exchanger Tube Fitted with Twisted-Tape Consisting of Triangular-Wing and Alternate-Axis

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Abstract— ***Heat transfer, friction factor and thermal performance characteristics in a tube equipped with twisted-tapes consisting of triangular-wings and alternate-axes (TT-TWs) are experimentally investigated. The experiments conducted using TT-TWs with three different wing-depths (d/W = 0.1, 0.2 and 0.3) and constant twist ratio of y/W = 4.0. The typical twisted tape (TT) and twisted-tape with alternate-axis (TA) were also tested for evaluation. The experiments were conducted in a turbulence region with Reynolds numbers ranging from 5500 to 20,200 using water as the test fluid. The experimental results reveal that heat transfer rate and friction factor increase with increasing wing-depth ratio. Heat transfer rates associated with TT-TW at d/W = 0.1, 0.2 and 0.3 are enhanced by around 73%, 82% and 92% while friction factors for the corresponding twisted tapes are increased by 300%, 337% and 376% over those of the plain tube. In addition, the empirical correlations of the heat transfer (Nu) and pressure drop (f) are also presented.***

Index Terms— **Heat Transfer Enhancement, Heat Exchanger, Twisted Tape, Turbulence.**

I. Introduction

Heat transfer augmentation (HTA) techniques are frequently used in a heat exchanger system in order to augment heat transfer and increase the thermal performance of the system. HTA techniques can be classified into two categories: active and passive methods. In the active methods, heat transfer is improved by supplying extra energy to the fluid or the equipment. In contrast, the passive enhancement can be attained without any external energy. The approaches of this category take account of rough surface, surface area extension, coated surface, and turbulator/swirl generator device. Within the passive category, insertion of swirl generator device is one of the most promising techniques. The major function of swirl generator device is reducing the thickness of the boundary layer as a consequence of an improved fluid mixing. Among the inserts, twisted tapes have gained great attention due to their low cost and acceptable performance. Twisted tapes are usually installed at the core tube to promote the fluid transfer between the core tube and fluid in the near wall tube.

As reported in several research papers, twisted tapes were combined with several enhancement devices for further improving heat transfer rate, for example: twisted with spirally grooved tube, converging-diverging tube, corrugated tube, dimpled tube, conical-ring, wire coil turbulators and twisted tape with wire-coil. On the other hand, twisted tapes were modified to improve the fluid mixing near a tube wall aiming for better heat transfer rates, such as peripherally-cut twisted tape [1], broken twisted tape insert [2], serrated twisted tape [3], twisted-tapes with oblique teeth [4], delta-winglet twisted tape [5], trapezoidal-cut twisted tape [6], square-cut twisted tape [7], twisted tape with wire-nail [8], and twisted tapes with alternate-axes and triangle, rectangular and trapezoidal wings [9]. Recently, twisted tapes were applied simultaneously with several nanofluids in the heat exchanger tubes. Mostly, heat transfer rates associated by the combined techniques or the modified twisted tapes are higher than those the typical twisted tape. However, an improve heat transfer is unavoidably accompanied by an increase of pressure drop. To obtain optimum conditions, the proper geometries of twisted tape are necessary.

In the present investigation, the heat-transfer, flow friction and thermal performance factor characteristics in a heat exchanger tube with triangular-wing twisted-tape (TT-TW) for three different wing-depths (d/W = 0.1, 0.2 and 0.3) have been studied. The experiments were tested with Reynolds number range of interest from 5000 to 20,200. The tube wall was maintained under a uniform heat flux condition.

II. Twisted Tape Consisting of Triangular-Wings

All twisted tapes used in the present work are made from an aluminum strip with thickness (δ) of 0.8 mm, width (W) of 20 mm and length (L) of 1000 mm. Initially, the typical twisted tape (TT) was fabricated by twisting the strip at constant twist ratio (y/W) of 4.0, where y is twist length per 180 degree. Three different twisted-tapes consisting of triangular-wings and alternate-axes (TT-TWs) were formulated by cutting in triangular shape of the TTs with three different wing-depth ratios of d = 2 mm (d/W = 0.1), 4 mm (d/W = 0.2) and 6 mm (d/W = 0.3) while the wing width-length was kept constant at 4 mm. At each cut, both sides of the tape were twisted simultaneously to angle difference of 90 degree. To form the alternate axis, the tape was cut on both sides at every twist length (180 degree). The appearance of the tapes used in the present work is shown in Fig. 1a.

III. Experimental Apparatus

Experiments were carried out in an open loop rig using water as working fluid as shown Fig. 1a. The heating test tube is made of copper with thickness of 1.5 mm, diameter of 19.5 mm and length of 1000 mm. Cold water was continuously supplied from a water chiller (0.5 m^3) to an overhead water tank (at 3 m elevation) by 0.5 hp centrifugal water pump. The overhead water tank had a volume of 0.5 m^3. The constant head water level in overhead water tank was controlled by overflow discharged through the ball valve to the ground water tank (0.5 m^3). Calming section made of copper tube with length of 1500 mm was used to eliminate the entrance effect. One end of heating test tube was attached with the calming section, while the other end was attached with the mixing section. The test tube was heated by continually winding flexible electrical wire to provide a uniform heat flux boundary condition. The K-type thermocouple beads were tapped along the local tube wall for 15 stations for monitoring the temperature of the surface tube wall. At the entrance of the test tube, the volume flow rate of water was measured using rotameter. The water flow rate was adjusted by the globe valve which was located upstream of the rotameter. All experiments were carried out at the room temperature of 300 K. For each test run, temperature, volumetric flow rate and pressure drop of the bulk air were recorded at steady state conditions, in which the inlet water temperature was maintained at 27°C. More details of the experimental set-up can be found elsewhere [5].

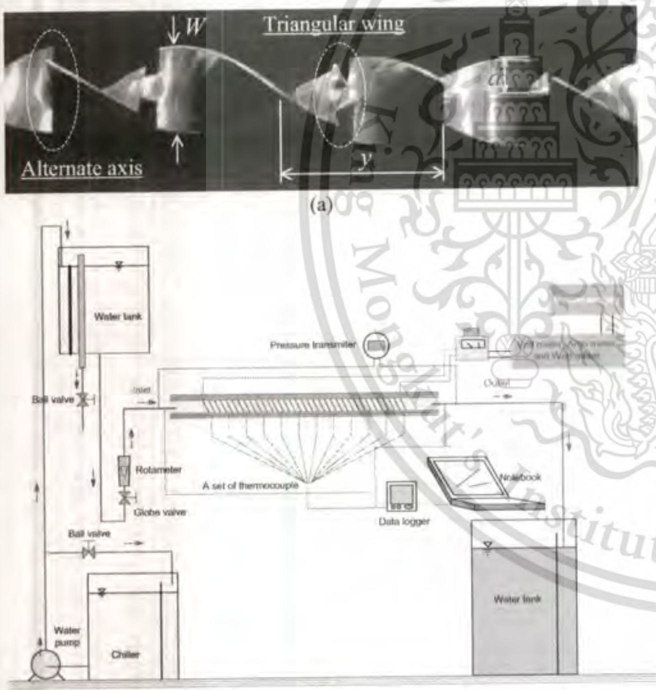

Figure 1. Photograph of triangular-wing twisted-tape and details of experimental setup

IV. Results and Discussion

The experimental investigations of heat transfer in a copper tube fitted with TT-TW of three different depth-ratios, d/W = 0.1, 0.2 and 0.3, are presented.

A. Heat transfer

The experimental results obtained under turbulent flow and uniform heat flux conditions are presented. Under similar conditions, Nusselt numbers of the tubes with TT-TWs are consistently higher than those of the tubes with TT/TA and the plain tube. This is a consequence of a further enhancement of the turbulent flow near a tube wall which results in a more efficient disruption of a thermal boundary layer. The effect of d/W on the heat transfer characteristic in term of Nusselt number are showed in Fig. 2. The results indicate that the heat transfer rate of the tube with TT-TW insert increases with an increase in wing-depth ratio, (d/W) since the tape with a larger wing-depth induces higher turbulence intensity and thus better fluid mixing near the tube wall. The heat transfer rate associated with the tape at the highest depth ratio (d/W = 0.3), are augmented by around 5.3% and 10.9% over those associated by tapes with d/W = 0.2 and d/W = 0.1, respectively.

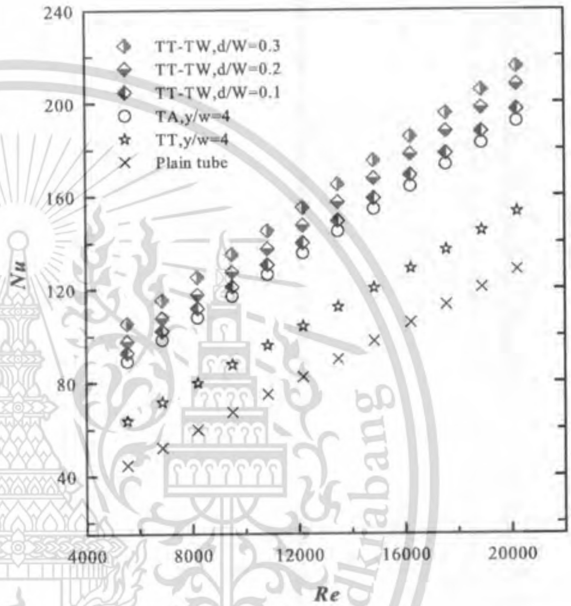

Figure 2. Variation of Nusselt number with Reynolds number

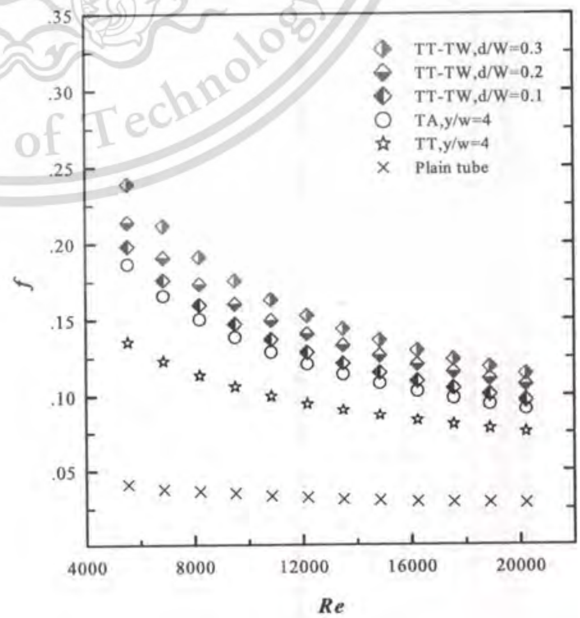

Figure 3. Variation of friction factor with Reynolds number

B. Friction factor

Figure 3 depicts the variation of the friction factor with Reynolds number (*Re*). For the range of interest, friction factor decreases with incresing Reynolds number for all cases. Apparently, friction factors of the tubes with TT-TWs are higher than those of the tubes with TT/TA and the plain tube, under similar conditions. This is due to the augmented dissipation of dynamic pressure of the fluid due to an extra interaction of the pressure forces with inertial forces in the boundary layer caused by TT-TWs. The application of the tape with larger wing-depth ratio (*d/W*) yields a higher friction factoras a consequence of the higher turbulence intensity near a tube wall. Over the range examined, the TT-TW with the largest wing depth ratio (*d/W* = 0.3) generates higher friction factor than the ones with *d/W* = 0.2 and *d/W* = 0.1 by around 8.8% and 19%, respectively.

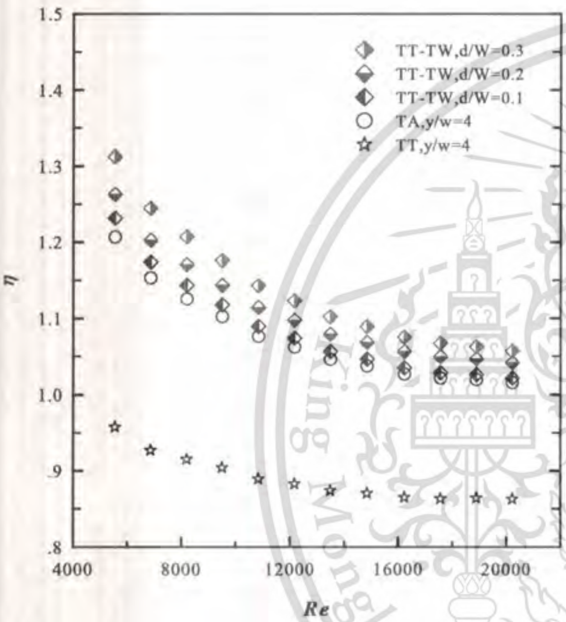

Figure 4. Variation of thermal performance factor with Reynolds number

C. Thermal performance factor

The relationship between thermal performance factor at the same pumping power and Reynolds number is presented in Fig. 4. For the range determined, thermal performance factor monotonically decreases with increasing Reynolds number. Under similar conditions, thermal performance factors given by TT-TWs are consistently higher than those provided by TT and TA. This indicates that an advantage the use of TT-TWs (an increase of Nusselt number) is more prominent than a disadvantage one (an increase of friction factor). For TT-TWs, thermal performance factor increases as wing-depth ratio increases. The TT-TWs with *d/W* = 0.1, 0.2 and 0.3 give thermal performances in the ranges between 1.02 and 1.23, 1.04 and 1.26, and 1.06 and 1.31, respectively. The results mentioned above suggest that TT-TWs are feasible in terms of energy saving at low Reynolds number and large wing-depth ratio.

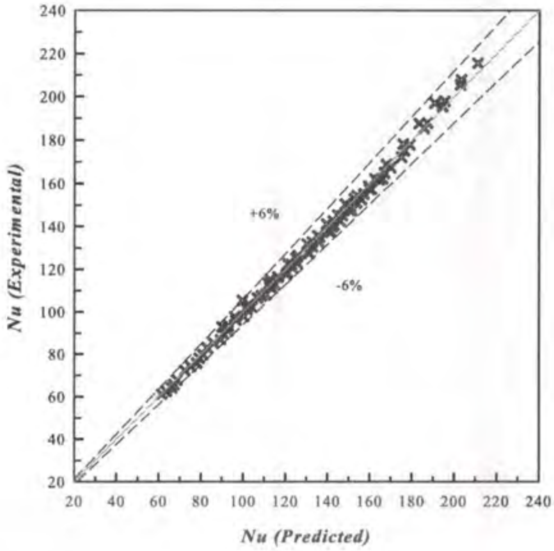

Figure 5. Validation test of Nusselt number correlation

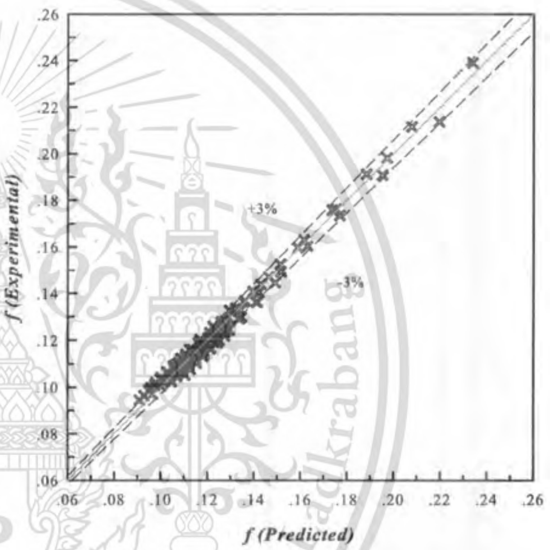

Figure 6. Validation test of friction factor correlation

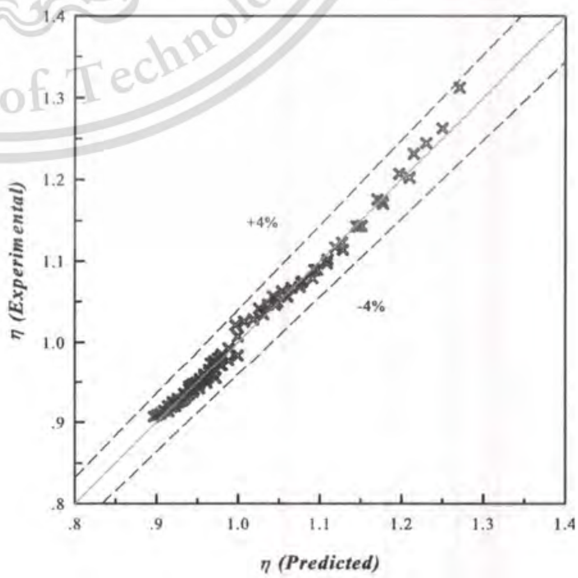

Figure 7. Validation test of thermal performance factor correlation

D. Empirical correlations

The experimental results of Nusselt number (Nu), friction factor (f) and thermal performance factor (η) are fitted, using least square regression analysis, to evaluate the reliability of the developed empirical correlations. The predicted data from equations (1) to (3) of the Nu_{pred}, f_{pred} and η_{pred} are plotted against experimental data of the Nu_{exp}, f_{exp} and η_{exp} in Figs. 5, 6, and 7, respectively. As seen from these figures, the deviations for Nusselt number, friction factor and thermal performance factor are ±6%, ±3%, and ±4%, respectively.

Nusselt number correlation:

$$\mathrm{Nu} = 0.382\,\mathrm{Re}^{0.58}\,\mathrm{Pr}^{0.4}(\mathrm{d/W})^{0.092} \quad (1)$$

Friction factor correlation:

$$\mathrm{f} = 33.16\,\mathrm{Re}^{-0.553}(\mathrm{d/W})^{0.155} \quad (2)$$

Thermal performance correlation:

$$\eta = 4.994\,\mathrm{Re}^{-0.153}(\mathrm{d/W})^{0.041} \quad (3)$$

V. Conclusion

Augmentation of heat transfer rate in heat exchanger tubes by means of TT-TW inserts is investigated experimentally. Effect of the wing-depth ratio *(d/W)* on the Nusselt number, friction factor and thermal performance factor characteristics are reported. Major findings can be summarized as follows:

- Under similar conditions, Nusselt numbers and friction factors of the tubes with TT-TWs are consistently higher than those of the tubes with TT/TA and the plain tube.
- Thermal performance factors given by TT-TWs are higher than those provided by TT and TA due to the prominence of an advantage by the use of TT-TWs (an increase of Nusselt number) over a disadvantage one (an increase of friction factor).
- Heat transfer rate, friction factor and thermal performance factor of the tube with TT-TW insert increases with an increase in wing-depth ratio *(d/W)*.
- TT-TWs are feasible in terms of energy saving at low Reynolds number and large wing-depth ratio.
- For the present work, deviations of experimental data from the predicted ones are ±6%, ±3%, and ±4% for Nusselt number, friction factor and thermal performance factor, respectively.

VI. Acknowledgement

Financial support from the Energy Policy and Planning Office, Ministry of Energy, Thailand (EPPO) for this research is greatly acknowledged.

VII. References

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Effect of perforated twisted-tapes with parallel wings on heat transfer enhancement in a heat exchanger tube

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Abstract

This article reports an experimental investigation on heat transfer and pressure drop characteristics of turbulent flow in a heating tube equipped with perforated twisted tapes with parallel wings (PTT) for Reynolds number between 5500 and 20500. The design of PTT involves the following concepts: (1) wings induce an extra turbulence near tube wall and thus efficiently disrupt a thermal boundary layer (2) holes existing along a core tube, diminish pressure loss within the tube. The parameters investigated were the hole diameter ratio (d/W = 0.11, 0.33 and 0.55) and wing depth ratio (w/W = 0.11, 0.22 and 0.33). A typical twisted tape was also tested for an assessment. Compared to the plain tube, the tubes with PTT and TT yielded heat transfer enhancement up to 208% and 190%, respectively. The evaluation of overall performance under the same pumping power reveal that the PTT with d/W = 0.11 and w/W = 0.33, gave the maximum thermal performance factor of 1.32, at Reynolds number of 5500. Empirical correlations of the heat transfer, friction factor and thermal performance for tubes with PTTs were also developed. In addition, the swirling/axial flow patterns of tube with PTT were visualized using dye injection technique.

Keywords: Heat transfer enhancement; heat exchanger; twisted tape; thermal performance

1. Introduction

Heat transfer enhancement techniques are widely used in many engineering applications for example heat recovery process, shell-and-tube heat transfer exchanger, air conditioning and refrigeration systems, nuclear energy industry, chemical reactors, high power laser systems and chemical process plants, etc [1-2].

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i:10.1016/j.egypro.2011.12.887

The most significant variables in reducing the size and cost of a heat exchanger are basically the heat transfer coefficient and pressure drop. An increase in the heat transfer coefficient generally leads to another advantage of reducing the temperature driving force, which increases the second law efficiency and decreases entropy generation. Several enhancement of heat transfer devices have been introduced and improved for increasing the heat transfer rate and thermal performance in heat exchangers by both of active and passive methods. Active method is the approach that needs the extra power source for example mechanical aids, surface-fluid vibration, injection and suction of the fluid, jet impingement, and use of electrostatic fields. On the other hand, passive method does not require an extra power source. The devices in this category are surface coating, rough surfaces, extended surfaces, turbulent/swirl flow devices, tube insert (wire, spiral spring, porous, static mixer, twisted tape, louvered strip, miniature hydraulic turbine, mesh, and internal fin inserts), convoluted (twisted) tube, additives for liquid and gases. Application of twisted tape in heat exchanger tube [3-5] is the one of key heat transfer enhancement techniques. Convective heat transfer assisted by modified twisted tapes with different geometries have been extensively investigated such as twin twisted tapes, dual twisted tape elements in tandem, alternate clockwise and counter-clockwise twisted-tape, twisted tape consisting of centre wings and alternate-axes, peripherally-cut twisted tape and serrated twisted-tape. As compared to a typical twisted tape, heat transfer enhancement by the modified twisted tapes with such geometries is achieved by extra turbulence and thus better fluid mixing. However, an improvement in heat transfer rate is generally achieved at an expense of substantially increased friction. The most desired twisted tape is the one yields excellent heat transfer with minimum increase in friction.

To deal with an antagonistic requirement as mentioned above, the newly design twisted tape called perforated twisted tape with parallel wings (PTT) is proposed. The design of PTT involves the following concepts: (1) wings induce an extra turbulence near tube wall and thus efficiently disrupt a thermal boundary layer (2) holes existing along a core tube, diminish pressure loss within the tube. To evaluate practical applications, the overall energy performance in term of thermal performance factors under the same pumping power is evaluated. The investigation was performed for fully developed flow in turbulence regime (5500≤Re≤20500). Empirical correlations for heat transfer, friction factor and thermal performance factor are also reported.

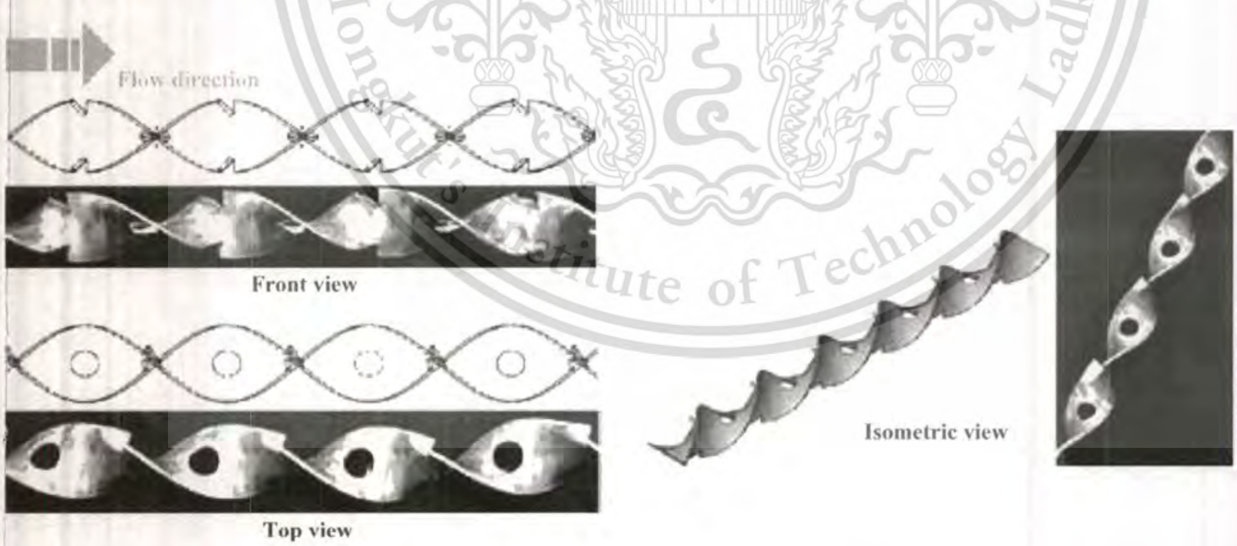
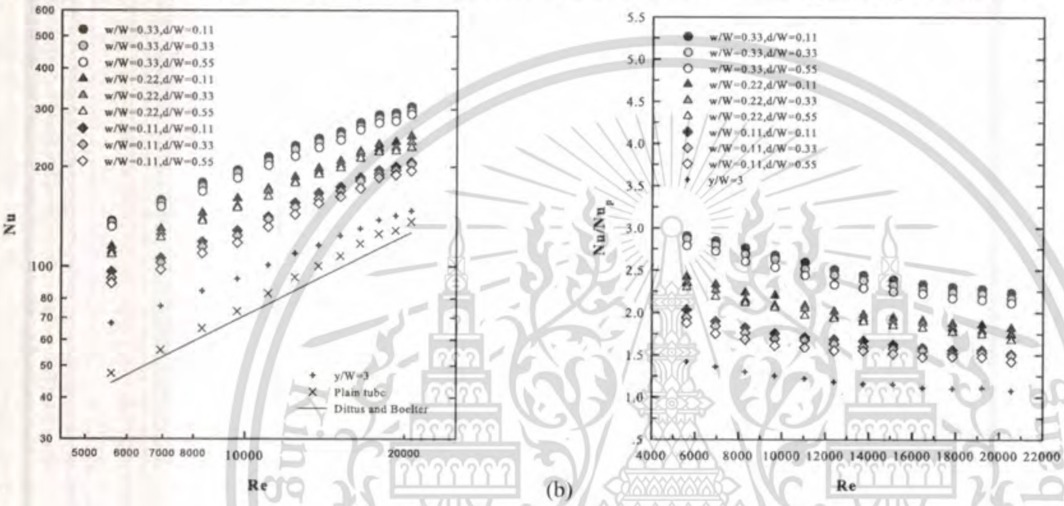


Fig. 1. Photograph and sketch of PTT with parallel wings

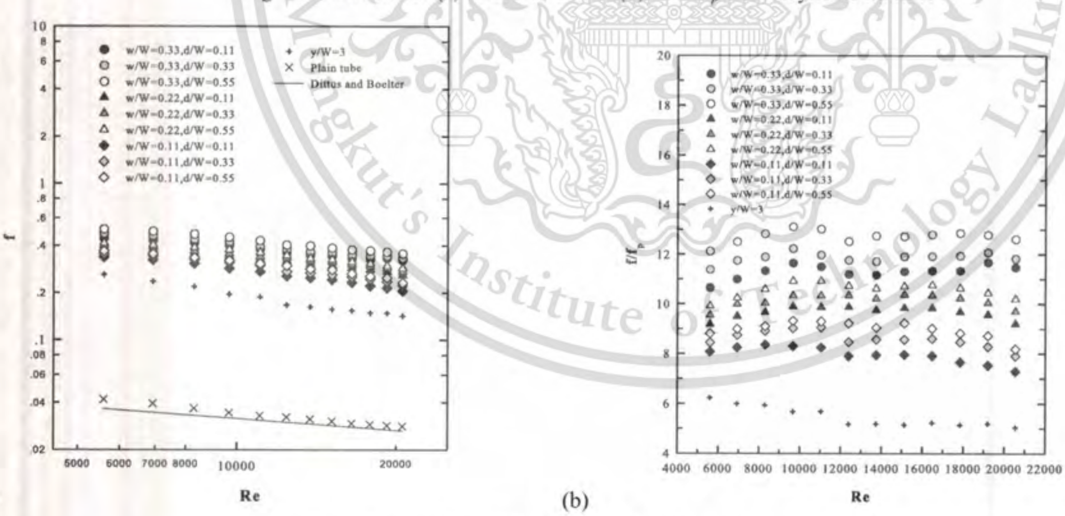
2. Perforated twisted tape with parallel wing

The tapes were made from the aluminum strip sheet having width (W) of 18 mm and thickness (δ) of 1.0 mm. All tapes were twisted at constant twist length of y = 56 mm which corresponds to twist ratio (y/W) = 3 to formulate typical twisted tapes (TTs). Then, TTs were modified by periodic generating holes along a core tube in straight line at every pitch length (180°/twist length). Furthermore, each tape was cut at the edge on both sides between adjacent holes, each cut was subsequently arranged in 45° to the axial flow in the same direction, so called parallel-wings the tap. The modified twisted tape (PTT) is depicted in Fig. 1. The parameters investigated were the hole diameter ratio (d/W = 0.11, 0.33 and 0.55) and wing depth ratio (w/W = 0.11, 0.22 and 0.33). The typical twisted tape (TT) was also subjected to the test, for comparison. All of the tapes were inserted at the core tube along the test section. More details of the twisted tape and experimental set-up can be found in the previous work given by Eiamsa-ard *et al.* [5].



(a) (b)

Fig. 2. Variations of (a) Nusselt number (b) Nu/Nu_p with Reynolds number



(a) (b)

Fig. 3. Variations of (a) friction factor (b) f/f_p with Reynolds number

3. Results and discussion

The experimental investigations of heat transfer, friction factor and thermal performance behaviors in a heat exchanger tube fitted with the PTT with parallel wings of varying wing-cut ratio (*w/W*) and hole diameter ratio (*d/W*) are described. Prior to the main experiment, the plain tube was tested and validated. Figures 2 and 3 show the validations of the present experimental results and with those available in the earlier works. As shown, the present friction factors agree well with those obtained from Blasius equation within ±8.5% and the present Nusselt numbers are within ±8% of those calculated from Dittus-Boelter equation. The present results of the plain tube are therefore used as the bases for the evaluation of the effects of twisted tape on heat transfer enhancement.

3.1. Flow visualization

Since it is difficult to distinct the main flow from the secondary flows induces by TT and PTT via the dye visualization technique in turbulent regime, thus the visualization was performed in laminar regime instead, only for qualitative comparison. Figure 4 shows the flow visualization of the flows through tubes with and without twisted tape. Only axial flow was observed in the plain tube (Fig. 4a) while common swirl flow was found with the presence of TT (Fig. 4b). In case of PTT (Fig. 4c), apart from a common swirl flow, there was attack of dye streams on the wings of PTT leading to an extra turbulence as well as collision among dye streams and thus superior fluid mixing to the case of TT. It should be mentioned that the dye stream positioned around the mid of the tape was directed through a hole to another side of the tape, and the stream direction was between those of an axial flow and a swirl flow.

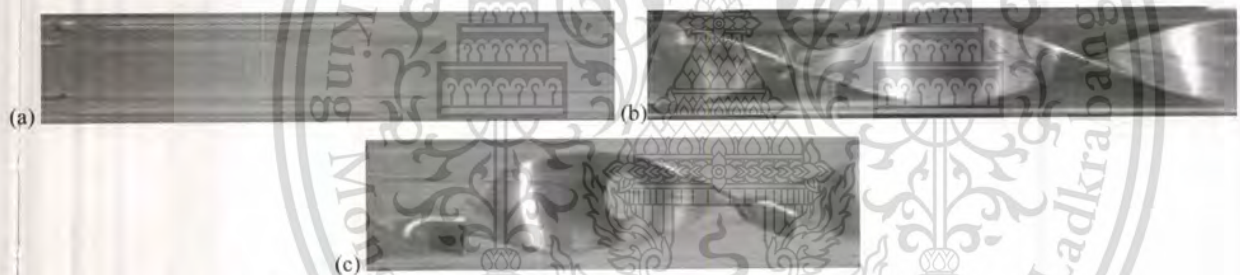


Fig. 4. Flow visualization of flows through (a) plain tube, (b) TT and (c) PTT

3.2. Effect of the presence of wing and wing depth ratio (w/W)

The experimental results of the tubes equipped with PTTs and TT as well as the plain tube obtained under turbulent flow conditions presented in Fig. 2(a-b). Influence of the presence of wing on heat transfer was significant, notified by the considerably higher Nusselt number given by PTT compared to those provided by TT. The superior heat transfer is responsible by the induction of extra turbulent flows near the tube wall as detected by the dye visualization (Fig. 4b). This efficiently disturbs a thermal boundary layer by generating periodic disruption of the viscous boundary layer. At similar operating conditions, it was found that Nusselt number increased with increasing wing depth ratio *(w/W)*. It can be explain by the fact that the larger depth ratio causes higher turbulence intensity and thus better mixing fluid near the tube wall. For the tape with the largest depth ratio (w/W = 0.33), the increase in heat transfer rate was up to 49% and 23% over those of the ones with w/W = 0.11 and w/W = 0.22, respectively. Figure 3(a-b) shows the variation of the friction factor with Reynolds number. Obviously, the use of the tape with larger wing depth ratio generated higher friction factor. This is directly related to the higher turbulence intensity as mentioned above. As found the tape with the largest depth ratio (w/W = 0.33) yielded 40.8% and 18.3% higher mean friction factor than the ones with w/W = 0.11 and w/W = 0.22, respectively. The result of thermal performance factor in which both heat transfer and friction are taken into account based in the same pumping power, is illustrated in Fig. 5. It is found that PTTs consistently gave higher thermal performance factor than TT, at similar conditions. Among PTTs, thermal performance factor increased with increasing wing depth ratio. Thermal performances varied between 0.71 and 1.01, 0.77 and 1.16, and 0.91 and 1.32 for the PTTs with w/W = 0.11, 0.22 and 0.33, respectively. It noteworthy that thermal performance was higher at lower Reynolds number, this implies that twisted tapes are more suitable for practical application at lower Reynolds number.

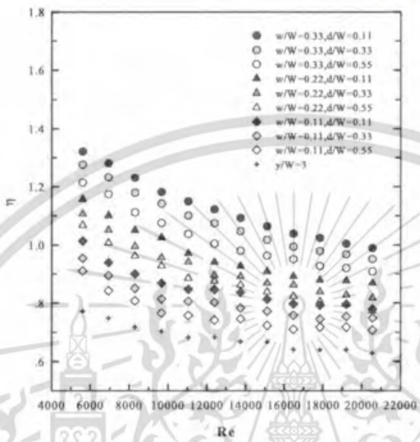
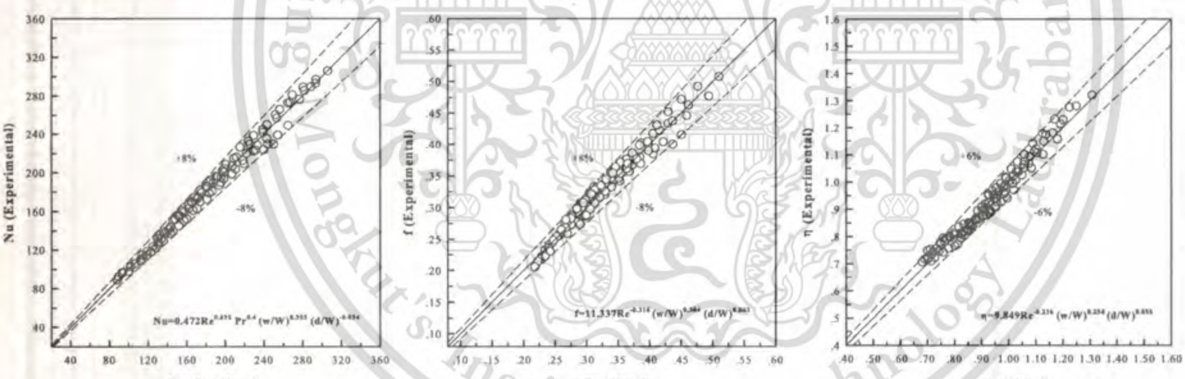


Fig. 5. Variation of thermal performance factor with Reynolds number



(a) (b) (c)

Fig. 6. Comparison between the predicted and experimental data of (a) Nu, (b) friction factor and (c) thermal performance factor

3.3. Effect of perforated diameter ratio (d/W)

The effect of perforated diameter ratio (d/W = 0.11, 0.33 and 0.55) on the heat transfer rate is shown in Fig. 2(a-b). Apparently, Nusselt number increased with the decrease of diameter ratio. This supports the finding by the dye visualization (Fig.4c) that the stream directed through a hole behaved between an axial flow and a swirl flow, with the larger hole diameter, the flow is assumed to be more similar to the axial flow leading to the loss of swirl intensity imparted to the flow between tape and surface wall. Heat transfer rate decreased up to 2.8% and 6% by the uses of the tapes with d/W = 0.33 for d/W = 0.55 compared to that of the tape with d/W = 0.11. By the same reason, friction generated by PTTs decreased with the increase diameter ratio (d/W) as presented in Fig. 3(a-b). The tapes with d/W = 0.33 for d/W = 0.55 yielded 6% and 11.6% lower mean friction factor than the one with d/W = 0.11. Effect of the perforated diameter ratio (d/W) on the thermal performance factor in a heat exchanger tube fitted with PTT is presented in Fig. 5. The thermal performances from using PTT with smaller hole diameter ratio (d/W) were observed to be higher than that those achieved from the tape with larger d/W. This signifies the dominant effect of increased heat transfer over that of increased friction factor as hole diameter decreases. It was found that the tape with the smallest diameter ratio (d/W = 0.11) provided higher thermal performance factor than the ones with d/W = 0.33 and 0.55 by around 4.9% and 10.4%, respectively. Note that for the present range, thermal performances achieved by using all tape inserts are above unity (low Re), indicating the economic benefit due to the heat transfer enhancement. The experimental results of Nusselt number, friction factor and thermal performance were fitted, usii g least square regression analysis, in which a wing-cut ratio (w/W) and a hole diameter ratio (d/W) were taken into account. The predicted data from the correlations of the Nu_{pred}, f_{pred} and η_{pred} are plotted against experimental data of the Nu_{exp}, f_{exp} and η_{exp} in Fig. 6(a-c). As shown from these figures the maximum deviations between the experimental data and correlations are ±8%, ±8%, and ±6%, respectively.

4. Conclusions

Augmentation of heat transfer rate in heat exchanger tubes by means of perforated twisted tapes (PTT) inserts is investigated experimentally. The results showed those heat transfer and friction factors were significantly influenced by the presences of wings and holes on PTTs. Both heat transfer and friction increased with the increase of wing depth ratio (w/W) and the decrease of perforation hole diameter ratio (d/W). Due to the dominant effect of increased heat transfer over that of increased friction factor, the thermal performance factor was found to be increased as wing depth ratio (w/W) increased and hole diameter ratio (d/W) decreased.

Acknowledgements

The authors would like to acknowledge with appreciation, the Energy Policy and Planning Office, Ministry of Energy, Thailand (EPPO) for financial support of this research.

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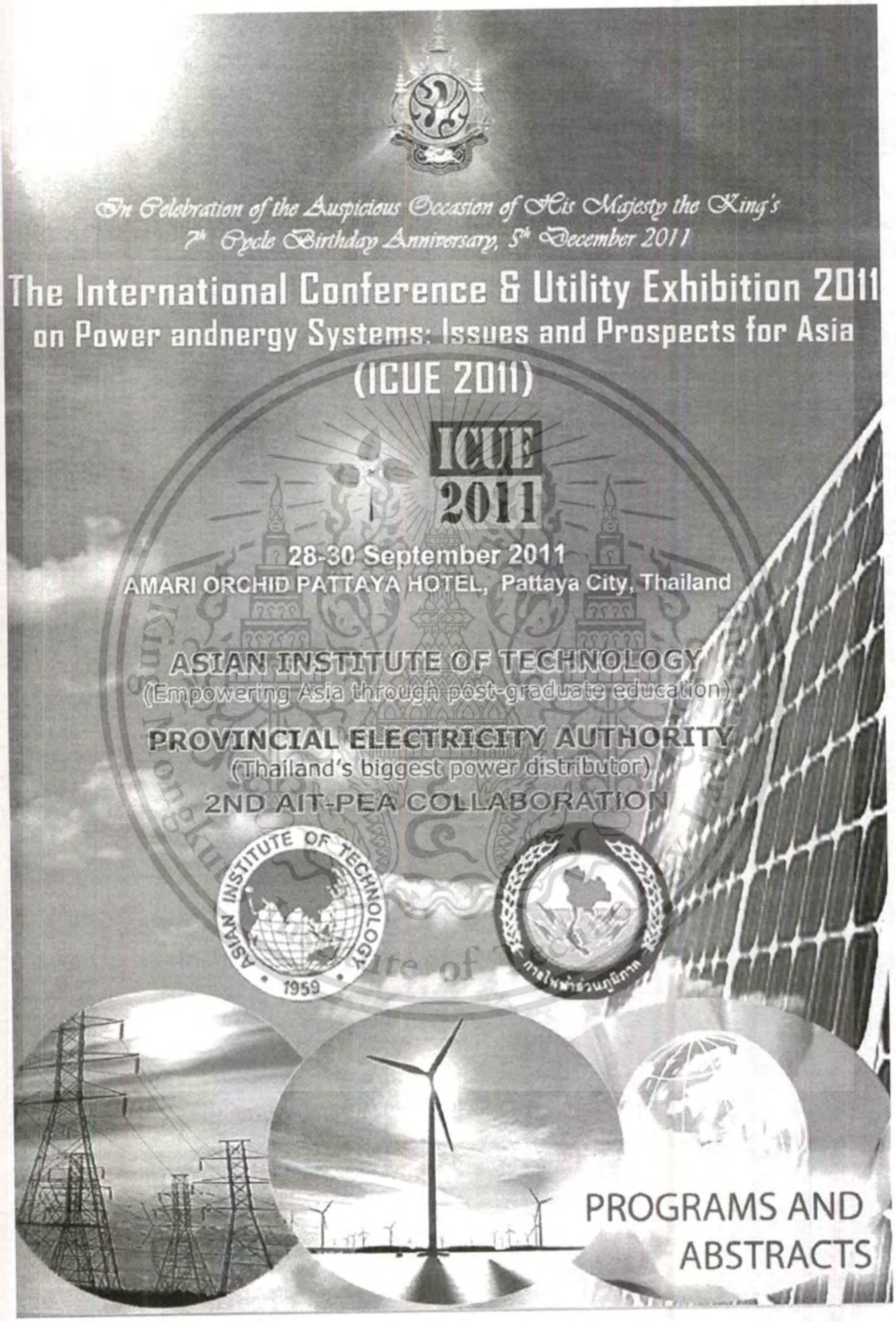
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International Conference & Utility Exhibition On Power and Energy Systems: Issues and Prospects for Asia (ICUE 2011) 1
28-30 September 2011, Pattaya City, Thailand

Influences of Twisted-Tape with Parallel Rectangular-Wing on Thermal Performance of a Heat Exchanger

P. Eiamsa-ard, C. Thianpong and S. Eiamsa-ard

Abstract**– The present paper reports heat transfer enhancement and friction factor characteristics in the tubes inserted with rectangular-winged twisted-tapes (TT-RWs). The wing-depth ratio (*d/W*) was varied from 0.1 to 0.3 while the tape-twist ratio was kept constant at *y/W* = 4.0. The experiments were conducted under uniform heat flux condition for Reynolds number between 5500 and 20,200 (based on the inner diameter, D), using water as the working fluid. The tubes with typical twisted tape (TT) and twisted tape with alternate-axis (TA) as well as a plain tube were also tested for assessment. The obtained results demonstrate that the utilization of the tubes with TT-RWs leads to the increases of both Nusselt number and friction compared to those from the uses of the tubes with TT, TA and the plain tube. In addition, Nusselt number increases as a wing-depth ratio of the twisted tape increases. The improvement of Nusselt number is directly related to the superior destruction of the boundary layer which is responsible by the stronger swirl flow and turbulence intensity. According to the results, the TT-RW with *d/W* = 0.3 yields the highest Nusselt number which is around 100% higher than that of the plain tube, corresponding to the thermal performance factor of 1.36 at constant pumping power.**

Index Terms**– Friction factor, heat exchanger, heat transfer enhancement, heat exchanger, swirl flow, twisted-tape, turbulence flow.**

I. Introduction

PASSIVE heat transfer enhancement techniques are manners to increase heat transfer coefficient without external power. Tube inserts form an important group of the enhancement techniques. Among the inserts, twisted tapes have gained great attention due to their low cost and acceptable performance. Twisted tapes are used as swirl generators, in order to improve fluid mixing reduce and induce interruption of thermal boundary layers. Heat transfer augmentation associated with twisted tape is always accompanied by an increase of pressure drop penalty. The proper design of the twisted tapes provides an increase of heat transfer rate with a reasonable pressure drop, resulting in effective energy savings.

Numerous research works have aimed to trade off the effects of a benefit (a heat transfer improvement) and a drawback (a pressure drop increase) of utilization of twisted tape shapes [1]-[15] in order to obtain attractive conditions. The attempts were made by the modifications of twisted tape geometries. Some modifications of the twisted tapes were conducted principally to suppress an increase of pressure drop in tubes including the loose-fit twisted tape at varying width twisted tape [1]-[2], regularly spaced tape elements [3]-[4], short-length twisted tape [5], spacer at the trailing edge of left-right twisted tapes [6]-[7]. However, heat transfer rates and thermal performances associated with the modified twisted tapes mentioned above, were significantly decreased compared to those given by a typical twisted tape. On the other hand, some twisted tapes were modified with major purpose to enhance heat transfer rates by improving fluid mixing between the tube core and inner tube wall such as twin twisted tapes [8], dual twisted tape elements in tandem [9], alternate clockwise and counter-clockwise twisted-tape [10]-[12], twisted tape consisting of centre wings and alternate-axes [13], peripherally-cut twisted tape [14] and serrated twisted-tape [15]. Most of the modified tapes in the latter group [8]-[15] offered superior heat transfer rate and thermal performance over the typical twisted tape. As mentioned above, the focus on the approach to enhance fluid mixing between a tube core and a tube wall is more promising for a better over performance of a heat exchanger than focusing on suppression of pressure drop. This inspires the design of twisted tape in the present work.

The modified twisted tapes used in the present work are the ones with rectangular-wings and alternate-axes (TT-RWs). The wings and alternate-axes are prospected to augment fluid mixing and turbulence intensity within a heat exchanger tubes especially around a tape edge. Effects of the wing depth ratio

The authors would like to acknowledge with appreciation, the Energy Policy and Planning Office, Ministry of Energy, Thailand (EPPO) for financial support of this research.

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(d/W = 0.1, 0.2 and 0.3) on the heat transfer, pressure drop and thermal performance factor characteristics are also examined. The experimental results of the tubes equipped with TT-RWs are compared to those of the ones with TT and TA as well as the plain tube for evaluation. The experiments were conducted under uniform wall heat flux condition for Reynolds number ranges from 5500 to 20,200. In addition, the empirical correlations for heat transfer, friction factor and thermal performance factor are also developed.

II. Twisted Tape With Rectangular-Wing

In the experiment, the twisted-tapes with rectangular-wing (TT-RWs) were made from the aluminum strip sheet and twisted at constant twist length of y = 80 mm or y/W = 4.0, where the tape has 20 mm in width (W) and 0.8 mm in thick (δ) as depicted in Fig. 1. The twisted tapes were cut at the edge of the tape in straight line at every pitch length (180°/twist length). Then each cut was bended in 45° to the main flow to form a parallel-wing. The wings were generated in three different depths of cut of W = 2, 4 and 6 mm, corresponding to depth ratios DR = d/W = 0.1, 0.2 and 0.3, respectively. To form the alternate axis, the tape was cut on both sides at every twist length (360 degree). Subsequently, both sides of the tape were twisted simultaneously to angle difference of 90 degree. In the experiments, each tape was located at the tube core as a part of a test section.

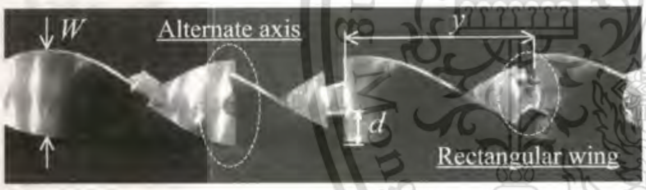

Fig. 1. Photograph of twisted-tape with rectangular-wing and alternate-axis (TT-RW).

III. Experimental Set-up

The experiments were carried out using water as the testing fluid and details of the experimental facility are demonstrated in Fig. 2. The system consisted of a heat exchanger tube, a centrifugal water pump, rotameter for measuring the volumetric water flow rate, a water tank, a manometer for measuring a static pressure, K-type thermocouples for measuring local temperatures, RTDs for measuring the temperatures of inlet/outlet of water in test tube, data logger, personal computer, and variac transformer for controlling the voltage supply, amp/volt meter. The heat exchanger tube was made from the copper. The tube had a test length of L = 1000 mm, inner diameter (D) of 20.5 mm, outer diameter of 23.5 mm, and thickness (t) of 1.5 mm. In the experimented, the tube was heated by continually winding flexible electrical wire until a uniform wall heat flux boundary condition was achieved. The electrical output power was controlled by a variac transformer with keeping the current of less than 9 amps to obtain a constant heat flux along the entire length of the test section. The outer surface of the test tube was well insulated and necessary precautions were also taken to minimize convective heat loss to surroundings. The inner and outer temperatures of the bulk water were measured at certain points with RTDs in conjunction with the data logger. Fifteen K-type thermocouples were tapped on the local wall of the tube and the thermocouples were placed round the tube to measure the circumferential temperature variation, which was found to be negligible. The mean local wall temperature was determined by means of calculations based on the reading of Chromel-constantan thermocouples.

In the apparatus setting above, the water at 27°C from an overhead water tank was directed through the rotameter and passed to the heat exchanger tube. The volumetric water flow rate was measured by a rometer. The pressure drop of the heat transfer test tube was measured with manometers under isothermal flow condition. The volumetric water flow rates from the water tank were adjusted corresponding to Reynolds number between 5500 and 20,200, through the control value, situated prior to entering the test tube. During the experiments, the bulk water was heated by an adjustable electrical heater wrapping along the heating test section. Both the inlet and outlet temperatures of the bulk water from the tube were measured by RTDs. It was necessary to measure the temperature at 15 stations on the outer surface of the heating test section for evaluation of an average Nusselt number. The all data of temperature, volumetric flow rate and pressure drop were taken at steady state conditions. The average Nusselt numbers were calculated based on all fluid properties at the overall bulk mean temperature

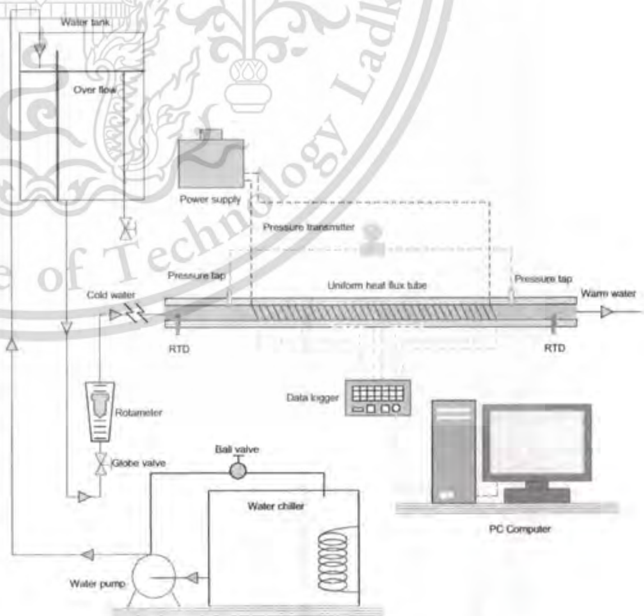

Fig. 2. Schematic heat transfer experimental setup.

In order to assess the reliability of the experimental facility and operating, the uncertainties of the data obtained experimentally were examined. The uncertainty in the data calculation was based on Ref. [16]. The maximum uncertainties of non-dimensional parameters were ±5% for Reynolds number, ±7% for Nusselt number and ±9% for friction. The uncertainty in the volumetric flow rate measurement was estimated to be less than ±4%, and pressure has a corresponding estimated uncertainty of ±5%, whereas the uncertainty in temperature measurement at the tube wall was about ±0.5%. The experimental results were reproducible within these uncertainty ranges.

IV. Data Reduction

In the present work, the water is used as working fluid and flowed through a uniform heat flux. The steady state of the heat transfer rate is assumed to be equal to the convective heat transfer from the test section which can be expressed as:

$$Q_{water} = Q_{in} - Q_{loss} = Q_{conv} \tag{1}$$

in which

$$Q_{water} = MC_p(T_o - T_i) \tag{2}$$

The convective heat transfer from the test section can be written as:

$$Q_{conv} = hA(\widetilde{T}_w - T_b) \tag{3}$$

The tube cross sectional flow area of the tube, A is calculated by using the tube diameter ($A = \rho D^2/4$).
Water bulk temperature (T_b) is determined from,

$$T_b = (T_o + T_i)/2 \tag{4}$$

The averaged wall temperatures are calculated from 15 points, lined between the inlet and the exit of the test pipe.

$$\widetilde{T}_w = \Sigma T_w / 15 \tag{5}$$

where T_w is the local wall temperature and evaluated at the outer wall surface of the inner tube. The averaged heat transfer coefficient, h and the mean Nusselt number, Nu are estimated as follows:

$$h = MC_p(T_o - T_i)/A(\widetilde{T}_w - T_b) \tag{6}$$

The average Nusselt number is calculated as:

$$Nu = hD/k \tag{7}$$

The Reynolds number is given by

$$Re = UD/\nu \tag{8}$$

The mean flow velocity inside tube is given as:

$$U = M/(\rho\pi D^2/4)$$

Friction factor, f can be written as:

$$f = 2\Delta P(D/L\rho U^2) \tag{9}$$

in which U is mean velocity of the tube. All of thermo-physical properties of the air are determined at the overall bulk water temperature from equation (4).

A useful comparison between twisted tape swirl generator (tube with enhanced devices, t) and axial flow (plain tube, p) can be made by comparing heat transfer coefficients at equivalent pumping power, since this is relevant to the operation cost. For equivalent pumping power

$$(\dot{V}\Delta P)_p = (\dot{V}\Delta P)_t \tag{10}$$

and the relationship between friction and Reynolds number for both the plain tube and those with swirl generator can be expressed as:

$$(fRe^3)_p = (fRe^3)_t \tag{11}$$

and then

$$\eta = (Nu_t/Nu_p)/(f_t/f_p)^{1/3} \tag{12}$$

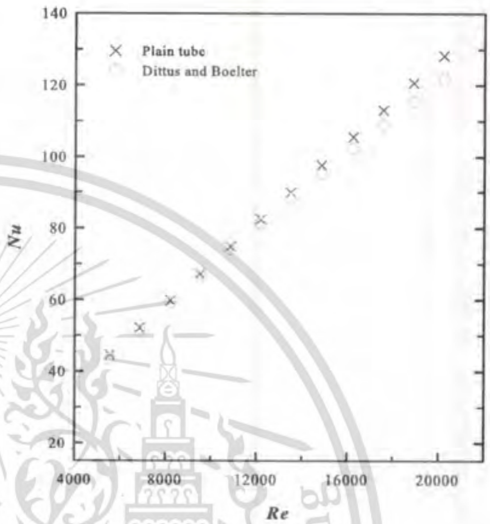

Fig. 3. Confirmation of Nusselt number for plain tube.

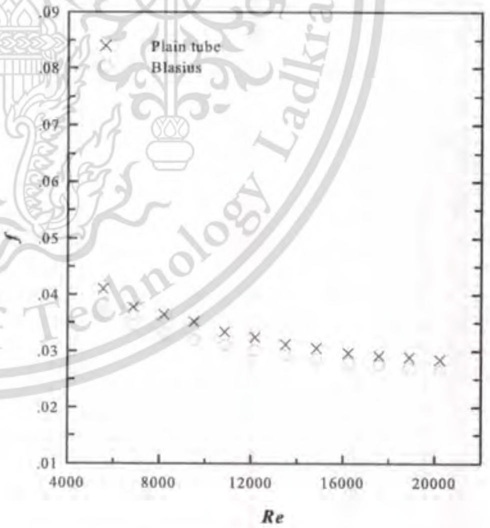

Fig. 4. Confirmation of friction factor for plain tube.

V. Results and Discussion

The results of heat transfer, friction factor and thermal performance behaviors in a heat exchanger tube fitted with the twisted-tapes with rectangular-wing (TT-RWs) are described and discussed along with those of in the tubes with TT and TA as well as the plain tube. Prior to the main results, the comparisons between the data obtained from the present plain tube and those calculated by the empirical correlations (proposed in open literature [17]), are presented to ensure the reliability of the experimental facility and operating. The correlations are stated as:

Correlation from Dittus-Boelter [17] is of the form:

$$Nu = 0.023Re^{0.8}Pr^{0.4} \tag{13}$$

Correlation from Blasius [17] is of the form:

$$f = 0.316Re^{-0.25} \tag{14}$$

The comparisons for Nusselt number and friction factor are shown in Figs. 3 and 4, respectively. In the figures, the present friction factor data agree well with those from Blasius equation within ±7.9% while the present Nuselt number data are within ±2.5% of those given by Dittus-Boelter equation.

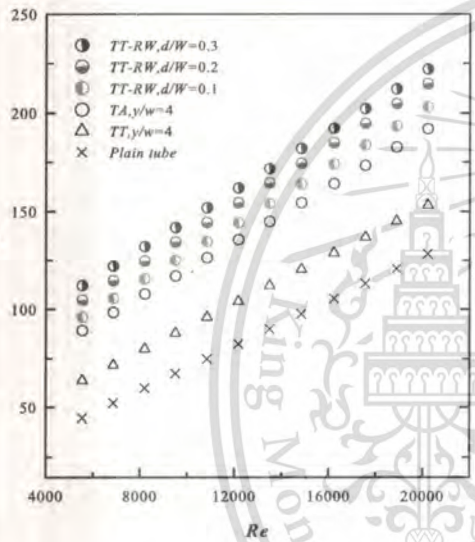

Fig. 5. Variation of Nusselt number with Reynolds number.

The experimental heat transfer result in term of Nusselt number obtained in the present investigation is illustrated in Fig. 5. In general, Nusselt number increases with increasing Reynolds number due to the rise of fluid turbulence intensity. Evidently, all tubes fitted with TT-RWs give higher Nusselt numbers than the ones with TT, TA and the plain tube. The possible actions for heat transfer enhancement by the TT-RWs can be summarized as follows: (1) a swirl flow due to their helical geometry (2) an extra fluctuation of flow at each alternate-point (3) an extra force caused by wings located near the tube wall. All actions cause the disruption of thermal/hydrodynamic boundary layer along the tube wall and thus heat transfer auxiliary. Therefore, the combined effects associated with TT-RW lead to superior heat transfer over the effect of swirl flow alone given by TT or swirl flow coupled with extra flow fluctuation provided by TA. It also can be observed that the heat transfer enhancing effect by alternate-axis is more significant than that by wing. In addition, Nusselt number increases as wing-depth ratio (d/W) increases. Mean Nusselt numbers of the tubes equipped with TT-RWs at wing-depth ratios of 0.1, 0.2 and 0.3 are enhanced respectively by around 39%, 49% and 57% over those in the tube with TT and 78%, 91% and 100% over those in the plain tube.

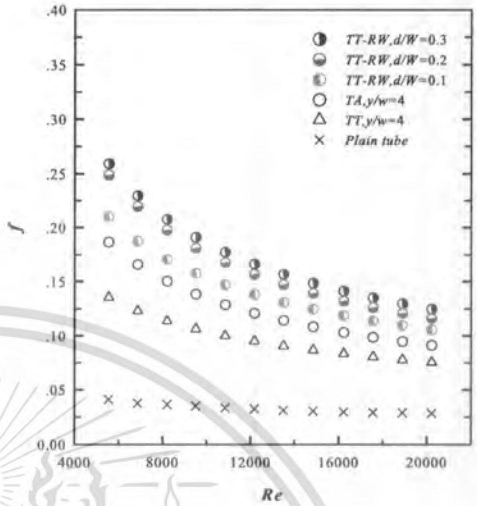

Fig. 6. Variation of friction factor with Reynolds number.

Friction factors are shown in Fig. 6 as a function of Reynolds number. For the tubes fitted with twisted tape inserts, friction factor noticeably decreases with increasing Reynolds number while that in the plain tube insignificantly changes over the studied Reynolds number range. Friction factors of the tubes with TT-RWs are consistently higher than those in the tubes with TT, TA and also the plain tube, under similar conditions. The higher friction factors in the tubes with TT-RWs are the results combined effects associated by both wing and alternate-axis actions leading to the high dissipation of dynamic pressure of the fluid over those in other tubes. The effect of wing-depth ratio on friction factor is similar to that on Nusselt number as friction factor increases with increasing wing-depth ratio. Mean friction factors of the tubes equipped with TT-RWs at wing-depth ratios of 0.1, 0.2 and 0.3 are increased respectively by around 46%, 66% and 76% over those in the tube with TT and 330%, 389% and 418% over those in the plain tube. It is noteworthy that the percentage increase in friction factor of a tube with TT-RW is significantly larger than that in Nusselt number at the identical volumetric flow rate.

Figure 7 shows the variation of thermal performance factor (η) with Reynolds number. All performance factors are evaluated at equivalent pumping power constraint. For the tubes equipped with TT-RWs and TA, thermal performance factor considerably decreases as the Reynolds number increases while that of the tube with TT slightly decreases for the present Reynolds number range. This signifies that the use of tape insert becomes less feasible in terms of energy saving at high Reynolds numbers. The thermal performance factors associated with TT-RWs are consistently higher than those given by TA and TT, at the given Reynolds number.

Therefore, it can be concluded that the use of TT-RW is more promising than the use of TA or TT, for energy saving purpose. Over the range examined, the TT-RWs provide highest thermal performance factor than the TT up to 42%. In addition, thermal performance factor increases as wing-depth ratio increases. For the present range, the maximum thermal performance factor is achieved with the use of TT-RW at wing-depth ratio of 0.3 and Reynolds number of 5500.

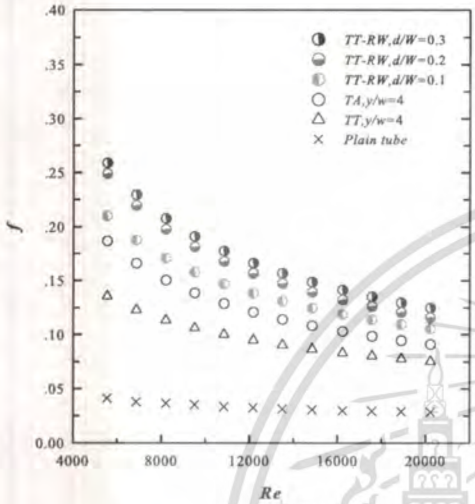

Fig. 6. Variation of friction factor with Reynolds number.

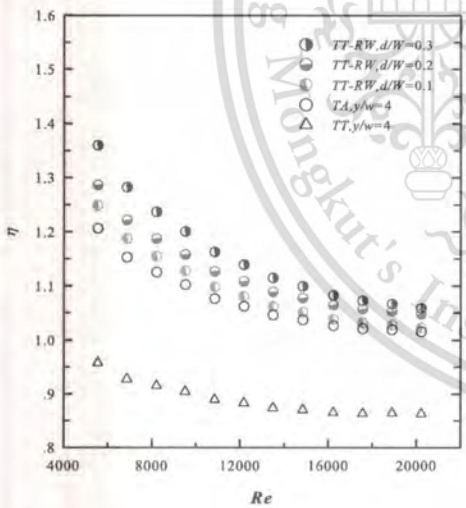

Fig. 7. Variation of thermal performance factor with Reynolds number.

Correlations of the present experimental Nusselt number, friction factor and thermal performance for the tubes fitted with for TT-RWs with three different depth ratios (d/W = 0.1, 0.2, and 0.3) are developed as follows:

$$Nu = 0.48Re^{0.562}Pr^{0.4}(d/W)^{0.104} \tag{15}$$
$$f = 40.59Re^{-0.56}(d/W)^{0.17} \tag{16}$$
$$\eta = 5.89Re^{-0.17}(d/W)^{0.048} \tag{17}$$

The predicted data show the maximum deviations of ±6% for Nusselt number, ±5% for friction factor and ±6% for thermal performance, as seen in Figs. 8-10.

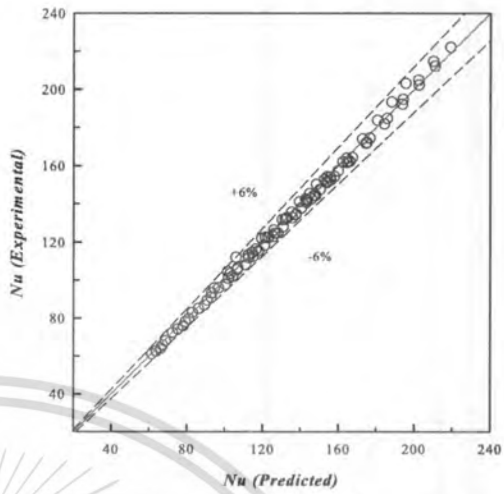

Fig. 8. Validation test of Nusselt number correlation.

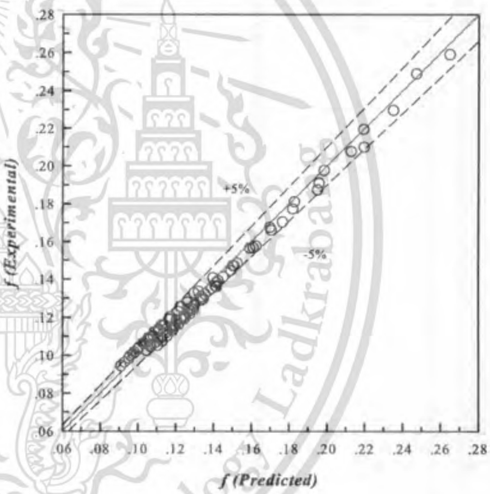

Fig. 9. Validation test of friction factor correlation.

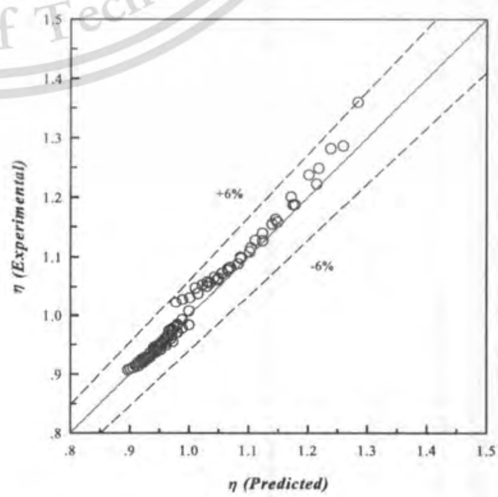

Fig. 10. Validation test of thermal performance factor correlation.

VI. Conclusions

Experimental investigations have been carried out to study the effects of the TT-RWs on the characteristics of the heat transfer, friction and thermal performance, in a heat exchanger tube. Major findings can be summarized as follows:

- The heat transfer rate in the heat exchanger tube can be enhanced by insertion of TT-RWs while it brings about an energy loss of the fluid flow. The mean heat transfer rates of the tubes with TT-RWs at d/W = 0.1, 0.2, and 0.3 are respectively 78%, 91% and 100% over that of the plain tube. However, the percentage increase in friction factor is much higher than that in Nusselt number at the same Reynolds number.
- The thermal performance factors of the tubes equipped with TT-RWs considerably increase with decreasing Reynolds number and increasing wing-depth ratio. At the given Reynolds number, TT-RWs consistently provide higher performance factor than TA and TT. For the present range, the maximum thermal performance factor of 1.36 is achieved with the use of TT-RW at wing-depth ratio of 0.3 and Reynolds number of 5500.

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VIII. Biographies

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