

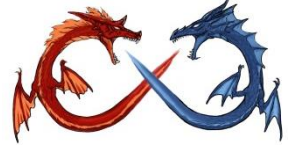
单相、两相传热强化技术及其进展 (Focus on tube side)

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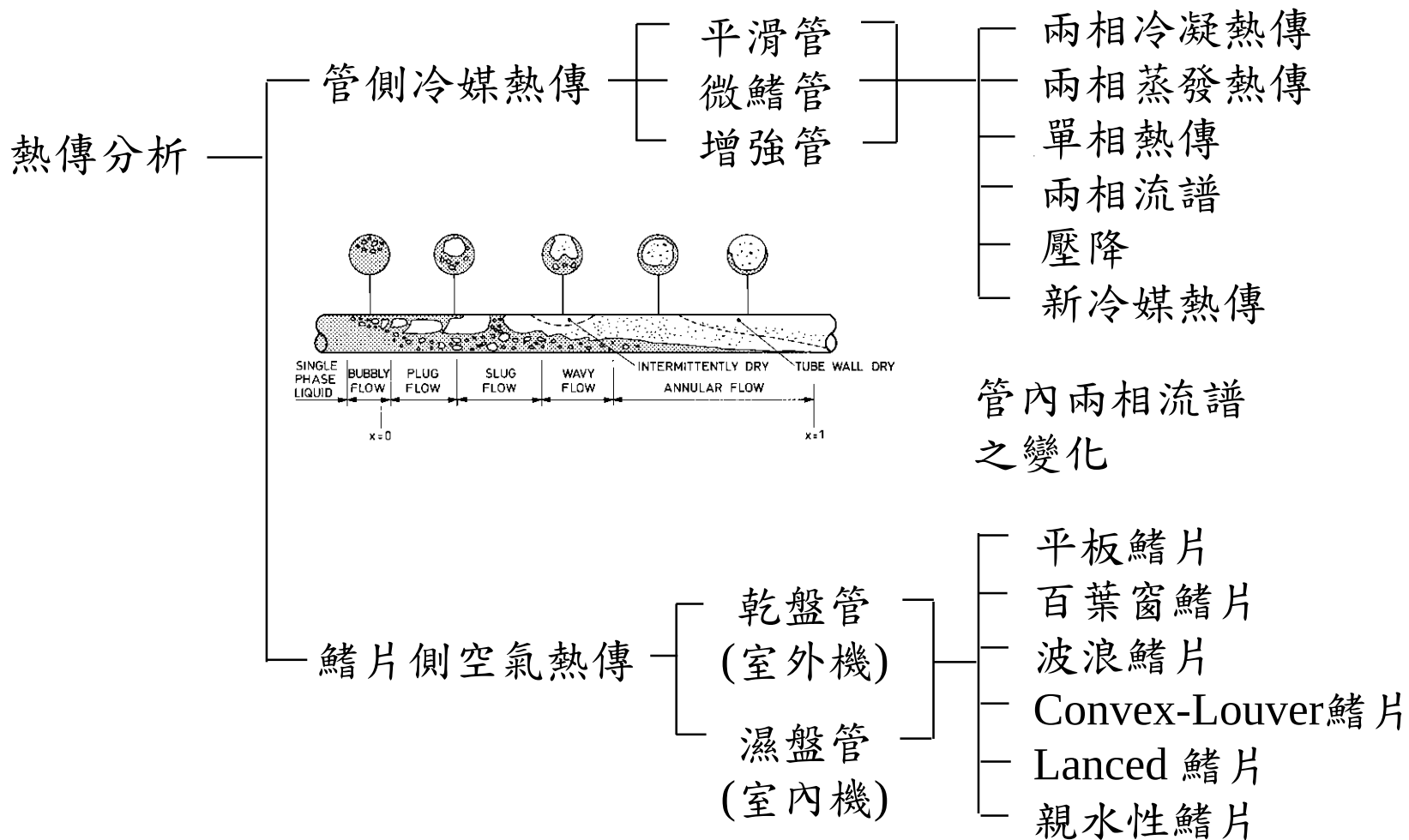


Outline

- 基本背景
- 定性選擇方法
- 一些觀念之澄清
- 定量性能評估
- 熱傳增強與熱傳管之選用
 - 基本概念
 - Boundary Layer Restarting
 - Intrusion design
 - Grooved design
 - Swirled Flow
 - Highly enhanced surface used for phase change
- Conclusions



典型氣冷式熱交換器的熱流現象





熱交換原理

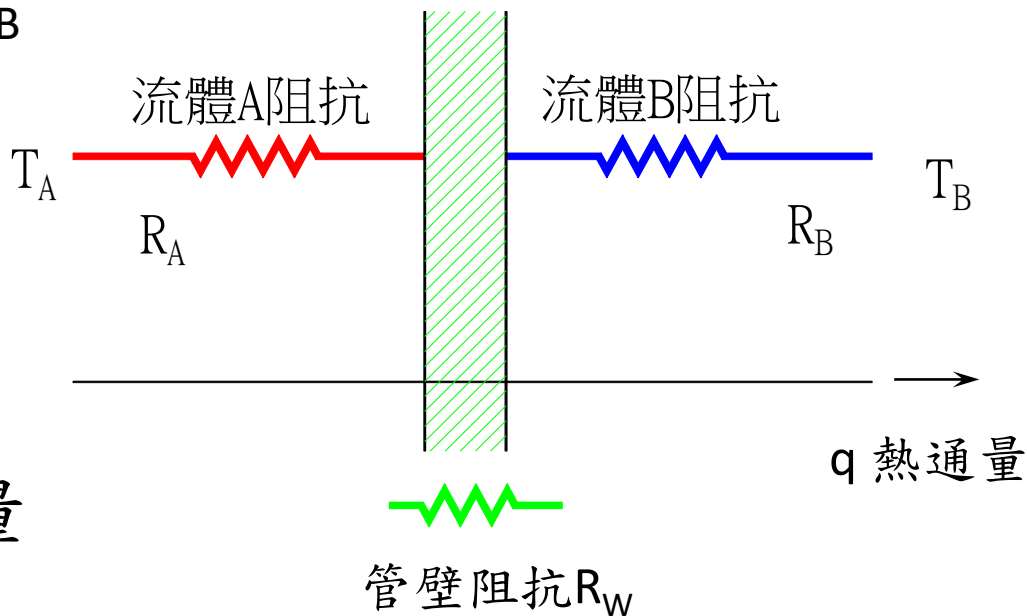
總阻抗： $R=R_A+R_W+R_B$

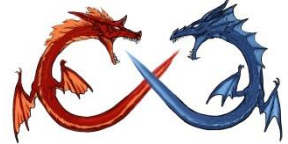
熱通量 q

$$q=(T_A-T_B)/R$$

$Q=q \cdot A \Rightarrow$ 總熱傳量

因此若能確切掌握流體阻抗，
即能精確設計熱交換器。





Fundamentals of Heat Transfer

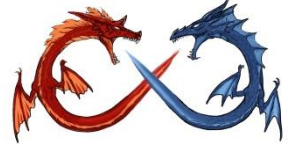
$$Q = UA\Delta T, \quad \frac{1}{UA} = \frac{1}{\eta_o h_o A_o} + \frac{\delta_w}{k_w A_w} + \frac{1}{h_i A_i}$$

or $Q = \eta_o h_o A_o \Delta T$ for electronic cooling

- U: overall heat transfer coefficient
 - h_i, h_o : inside & outside heat transfer coefficient
 - η_o : surface efficiency ($=1 - A_f/A_o(1 - \eta_f)$)
 - η_f : fin efficiency
- A: surface area
- ΔT : effective Temperature Difference

For Heat Transfer Augmentation. One needs to..

- (a) Increase A? (Most effective but limited by ΔP)
- (b) Increase U? (Effective but also prone to increasing ΔP)
- (c) What else can one do? (Manipulating ΔT ?)



總熱傳係數

所以若原熱交換器無鰭片的話，熱傳的阻抗方程式如下：

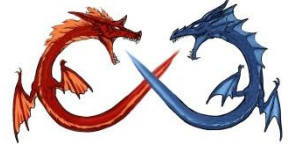
$$\frac{1}{UA} = \frac{1}{h_1 A_{w1}} + \frac{\Delta X}{kA} + \frac{1}{h_2 A_{w2}}$$

總阻抗 熱側阻抗 管壁阻抗 冷側阻抗

若有鰭片時，則阻抗方程式改寫如下：

$$\frac{1}{UA} = \frac{1}{\underbrace{\eta_1 h_1 A_{w1}}_{\text{熱側阻抗}}} + \frac{\Delta X}{kA} + \frac{1}{\underbrace{\eta_2 h_2 A_{w2}}_{\text{冷側阻抗}}}$$

總阻抗 熱側阻抗 管壁阻抗 冷側阻抗



例：鰭管式熱交換器阻抗分布

- 以空調常用的氣冷式冷凝器的相關應用而言， $h_i \approx 2000 \text{ W/m}^2 \cdot \text{K}$ ， $h_o \approx 50 \text{ W/m}^2 \cdot \text{K}$ ， $\eta_s \approx 0.7$ （請注意上述值只是在一種操作條件下的近似值，不可隨意將此值拿來運用於各種不同的應用場合上）。

假設 A_i 為一單位面積（ $= 1 \text{ m}^2$ ），而設 $A_o/A_i = 10$ ，故 $A_o = 10 \text{ m}^2$ ，總熱傳係數 U 的參考面積為 A_o ，則

$$\therefore \frac{1}{U_o A_o} = \frac{1}{2000 A_i} + \frac{1}{0.7 \times 50 A_o} = \frac{1}{2000 A_i} + \frac{1}{35 A_o} = \frac{0.0005}{A_i} + \frac{0.0286}{A_o}$$

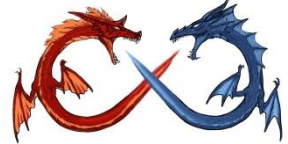
$$\Rightarrow \frac{1}{U_o} = 0.005(\text{管內阻抗}) + 0.0286(\text{管外阻抗}) = 0.0336 \text{ K} \cdot \text{m}^2 / \text{W}$$

$$\Rightarrow U_o = 29.77 \text{ W/m}^2 \cdot \text{K}$$

⇒管內的阻抗約佔 15% (0.005/0.0336)，

⇒而管外阻抗約佔85%。氣冷式熱交換器的熱傳阻抗幾乎都是落在空氣側

- 殼管式熱交換器的殼側與管側流體之熱傳係數均為幾千大小，其熱傳阻抗約佔有同等重要地位。



例：鰭管式熱交換器阻抗分布 (Conti.)

- ◆ Case 1：上述例題如將空氣側熱傳係數提高一倍，即 $h_i \approx 2000$ $\text{W/m}^2 \cdot \text{K}$ ， $h_o \approx 100 \text{ W/m}^2 \cdot \text{K}$ ， $\eta_s \approx 0.65$ (請注意空氣側熱傳係數增加，則鰭片效率會跟著降低，為什麼(想一想原因)?)。

假設 A_i 為一單位面積 ($= 1 \text{ m}^2$)，而設 $A_o/A_i = 10$ ，故 $A_o = 10 \text{ m}^2$ ，總熱傳係數 U 的參考面積為 A_o ，則

$$\therefore R_t = \frac{1}{U_o A_o} = \underbrace{\frac{1}{2000 A_i}}_{\text{冷媒側阻抗}} + \underbrace{\frac{1}{0.65 \times 100 A_o}}_{\text{空氣側阻抗}} = \frac{1}{2000 A_i} + \frac{1}{65 A_o} = \frac{0.0005}{A_i} + \frac{0.0154}{A_o}$$

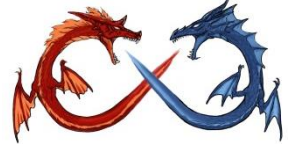
總阻抗

$$\Rightarrow R_t = \frac{1}{U_o A_o} = 0.0005(\text{管內阻抗}) + 0.00154(\text{空氣側阻抗}) = 0.00204 \text{ K} \cdot \text{m}^2 / \text{W}$$
$$\Rightarrow \frac{1}{U_o} = 0.0204 \text{ K} \cdot \text{m}^2 / \text{W} \Rightarrow U_o = 49.02 \text{ W} / \text{m}^2 \cdot \text{K}$$

➡ 管內的阻抗約佔25% (0.0005/0.00204)，

➡ 而空氣側阻抗約佔75%。

- ◆ 總熱傳係數比值 $49.02/29.77=1.65$ ，意味著如果兩側流體的溫差不變，則空氣側熱傳係數增加一倍，可提升熱傳量約65%。



例:鰭管式熱交換器阻抗分布 (Conti.)

- ◆ Case 2：上述例題如將冷媒側熱傳係數提高一倍，即 $h_i \approx 4000$ $\text{W/m}^2 \cdot \text{K}$ ， $h_o \approx 50 \text{ W/m}^2 \cdot \text{K}$ ， $\eta_s \approx 0.70$ (請注意冷媒側熱傳係數增加，原來的鰭片效率並不會跟著改變，為什麼(想一想原因)?)。

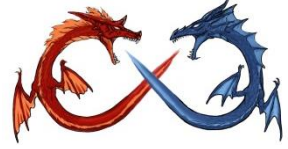
假設 A_i 為一單位面積 ($= 1 \text{ m}^2$)，而設 $A_o/A_i = 10$ ，故 $A_o = 10 \text{ m}^2$ ，總熱傳係數 U 的參考面積為 A_o ，則

$$\begin{aligned} \therefore R_t &= \frac{1}{U_o A_o} = \underbrace{\frac{1}{4000 A_i}}_{\text{冷媒側阻抗}} + \underbrace{\frac{1}{0.70 \times 50 A_o}}_{\text{空氣側阻抗}} = \frac{1}{4000 A_i} + \frac{1}{35 A_o} = \frac{0.00025}{A_i} + \frac{0.0286}{A_o} \\ \Rightarrow R_t &= \frac{1}{U_o A_o} = 0.00025 (\text{管內阻抗}) + 0.0286 (\text{空氣側阻抗}) = 0.00311 \text{ K} \cdot \text{m}^2 / \text{W} \\ \Rightarrow \frac{1}{U_o} &= 0.0311 \text{ K} \cdot \text{m}^2 / \text{W} \Rightarrow U_o = 32.15 \text{ W} / \text{m}^2 \cdot \text{K} \end{aligned}$$

➔ 管內的阻抗約佔8.0% ($0.00025/0.00311$)，

➔ 而空氣側阻抗約佔92%。

- ◆ 總熱傳係數比值 $32.15/29.77=1.08$ ，意味著如果兩側流體的溫差不變，則冷媒側熱傳係數增加一倍，可提升熱傳量約8%。

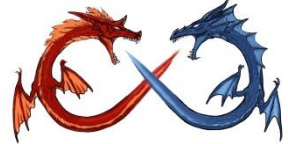


Stop & Think

- Question

- 兩熱交換器，型式完全一樣，幾何尺寸相同，一使用銅，一使用鈦，如果工作流體相同，流量相同，則

- 兩熱交換器的熱傳量，何者較大？
 - 兩熱交換器的熱傳性能 (以熱傳係數而言)，何者較佳？
 - 兩熱交換器的壓損，何者較大？



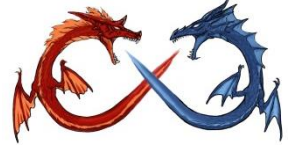
熱交換器選擇考量因素

- 熱流需求：包括熱交換量、工作流體的溫度、壓力與可允許的壓損，選擇的熱交換器當然必須滿足這些熱流的基本需求；且熱交換器能在工作溫度壓力下長期運作，能忍受因溫差所產生的熱應力影響，應力主要由入口壓力與溫度差所引起，常見各種熱交換器所能忍受的最大壓力與溫度範圍大致如下
- 熱交換器與流體的匹配性：熱交換器的材料必須與工作流體能長期搭配，無腐蝕的問題；其中必須特別注意結垢的影響，設計上應同時考慮正常操作設計點下與非設計點上運作時，結垢在不同溫度壓力變化下的影響，一般而言，典型氣對氣熱交換器的結垢影響較小，這是因為許多製程的應用上，氣體多半比液體來的乾淨，而且如果使用轉輪式的再生式熱交換器，氣體在不同時間時的相反方向的流動也有助於熱交換器本身的自清，因此結垢的影響相對比液體側小；不過此類再生式熱交換器也因氣體交互流過熱交換器而可能產生污染問題。



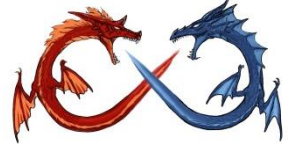
熱交換器選擇考量因素 (Conti.)

- 流體型式：由於氣體體的熱傳係數遠低於液體，因此氣對氣熱交換器通常需非常大的熱交換面積，一般的作法乃藉由增加鰭片、縮小水力直徑與使用小管徑熱傳管來增加面積的密集度，密集度增加同時增加流動壓損，氣側壓損的影響相對於液體測重要很多，設計上必須特別注意。對液對液的熱交換而言，為了避免交叉感染的影響，通常不應該考慮使用再生式熱交換器，相較於氣對氣熱交換器，液對液熱交換器的壓損影響較小。對氣對液的熱交換器而言，由於氣側的熱傳係數遠低液體側，因此設計上的初步原則為盡量平衡兩側的熱傳性能(即 $\eta_o h_o A_o \sim \eta_i h_i A_i$)。
- 維護性：設計時必須考量停機清理與置換的問題；同時應留意製程應用條件改變時所帶來的影響。
- 造價：造價為選擇設非常重要的因素，例如板式熱交換器的造價會比殼管式熱交換器大，但是如果同時考量裝置、操作、維護等成本的影響，板式熱交換器成本可能反而比較便宜。設計上如果比較在意長期操作的成本，則在設計上就必須特別留意流動的壓損而非純粹的熱傳考慮。
- 空間與重量：許多應用上必須考慮到裝置時空間與重量的問題，例如熱交換器裝置於高樓層的重量負荷或是都會區維護空間缺乏的現實問題。

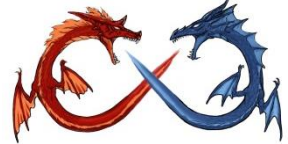


熱交換器的選擇原則（初步定性）

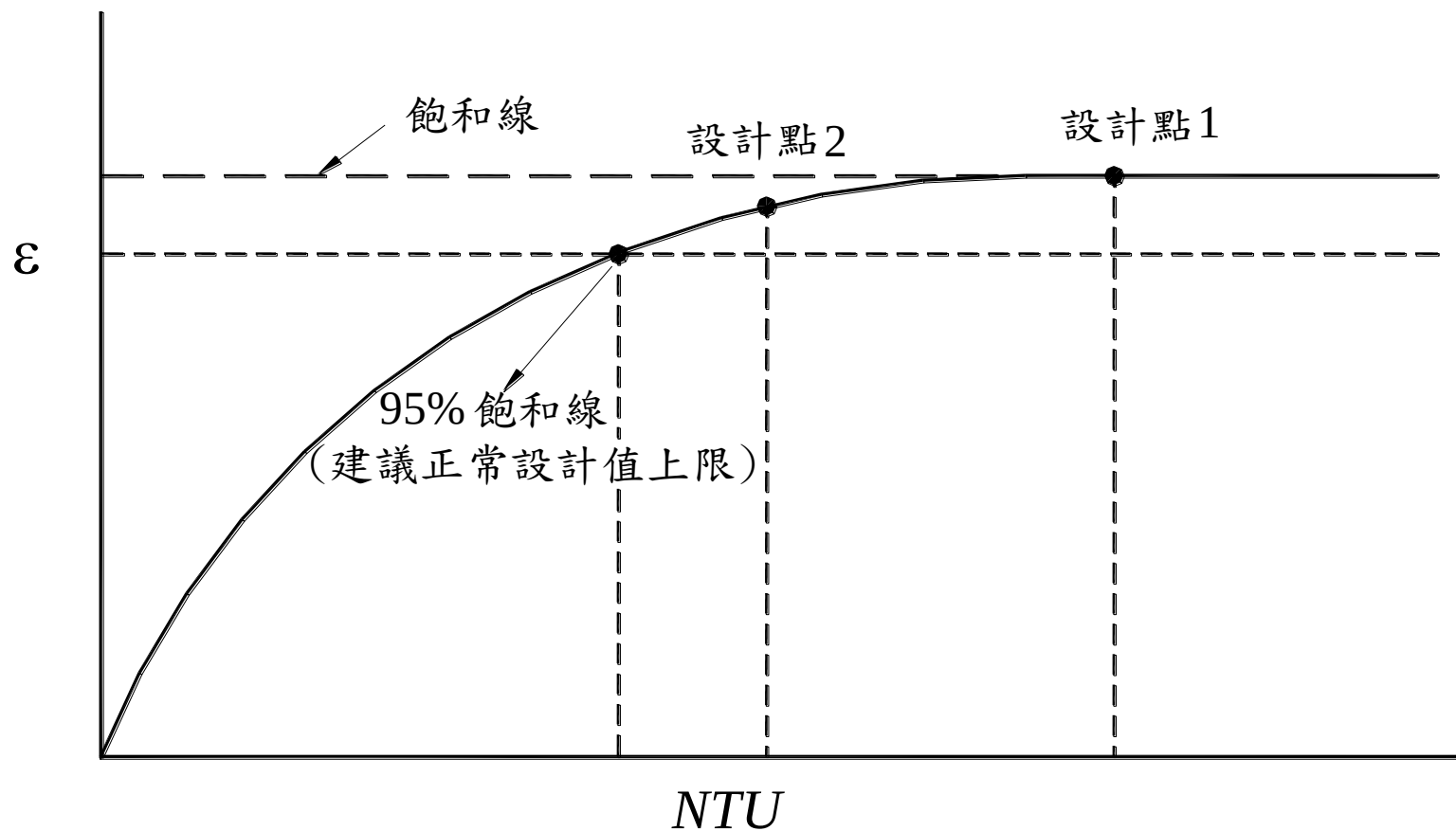
- Maximum Pressure
- Temperature range
- Fluids limitation
- Size range available
- Fouling & cleanability
- Plot area available
- Design life
- Location (maintenance)
- Is there a “temperature cross?”. If so, HX approaches counter flow is more appropriate



一些值得商榷的觀念

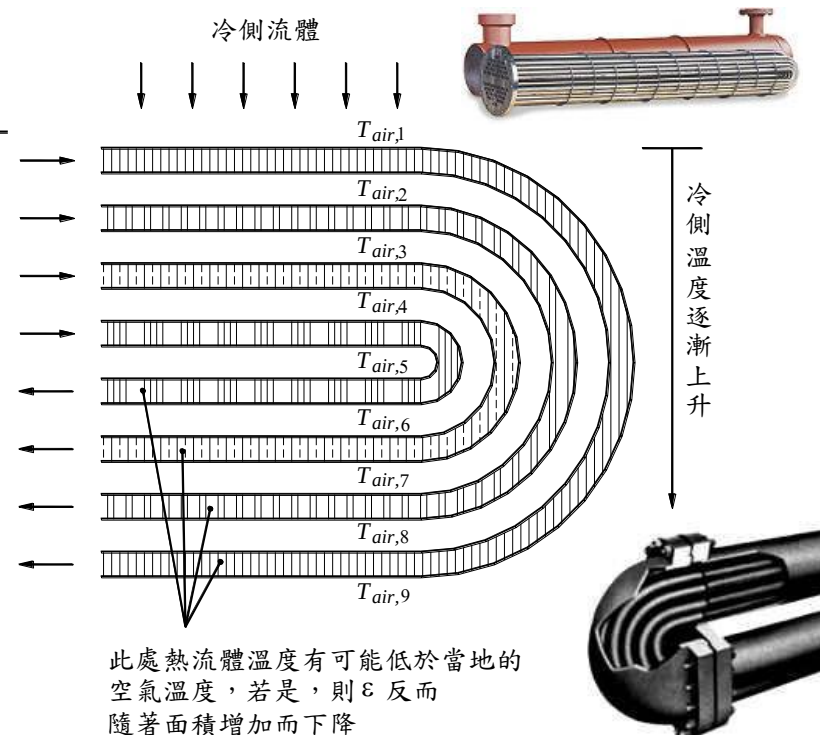
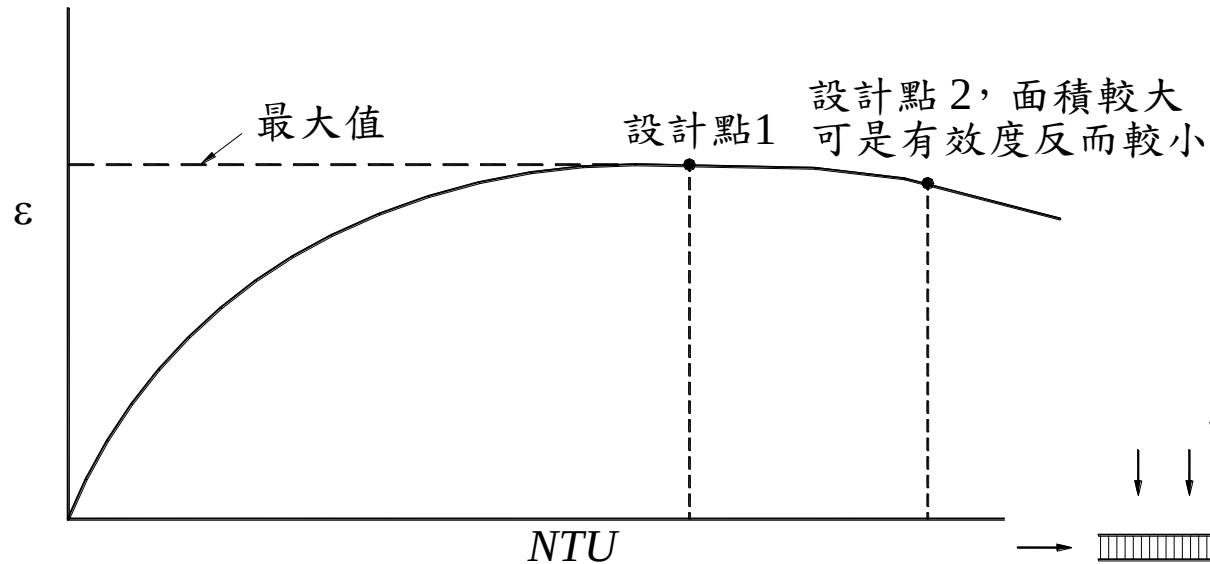


ε - NTU 關係是否已飽和？

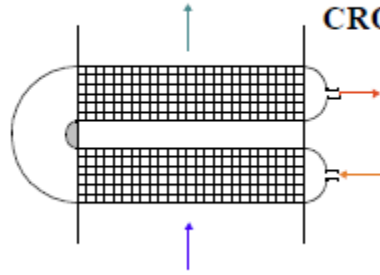




熱交換器越大熱傳效果越好？





Crossflow 2 Passes Infinite channels/tube rows in both streams $(Mc_p)_{\text{smaller}}$ in either stream — Unmixed/unmixed Mixed/mixed	 CRO-UNMIX-MP-CO-1
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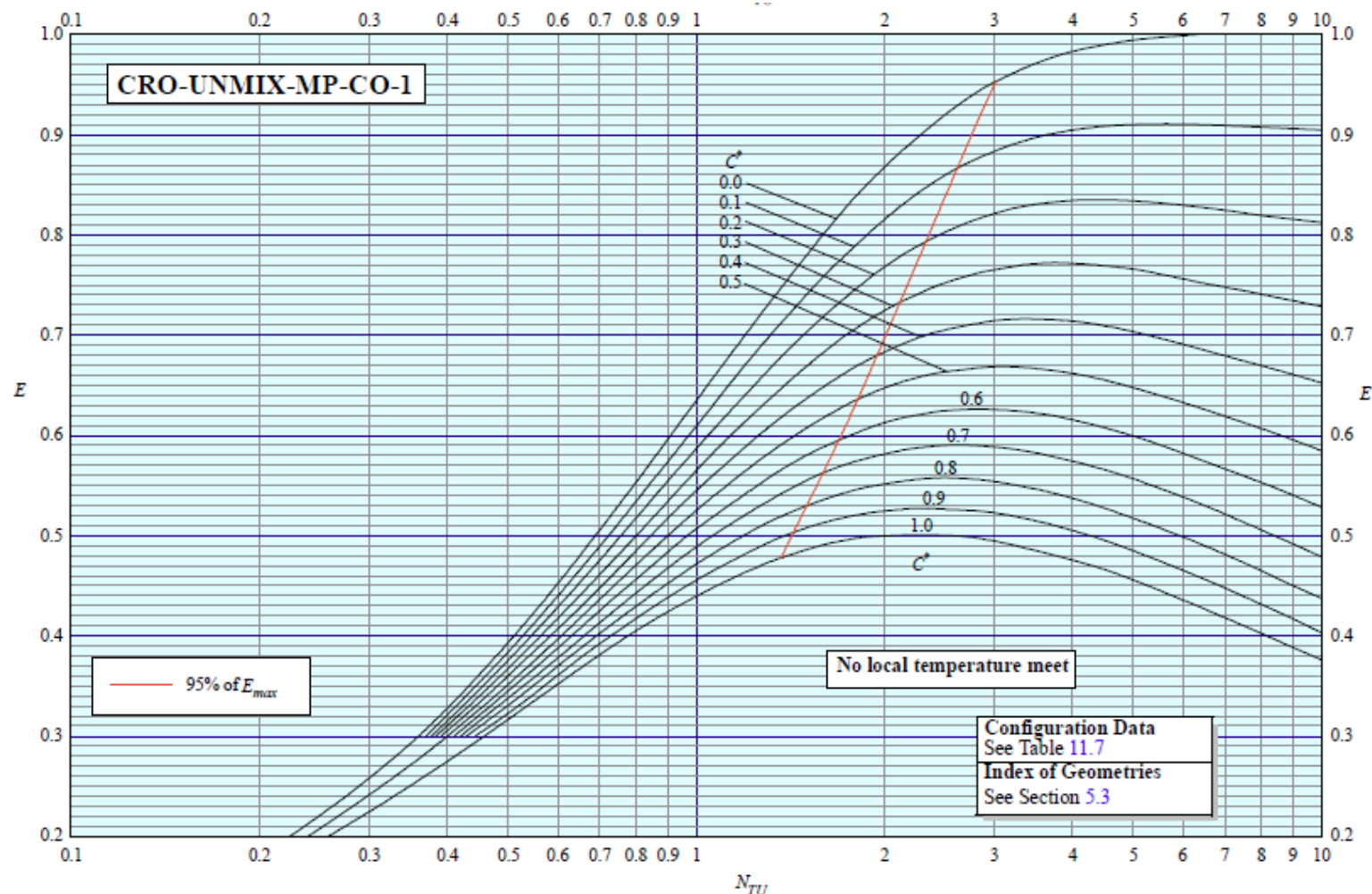


FIGURE 9 TWO-PASS CROSSFLOW WITH CO-CURRENT FLOW, BOTH STREAMS MIXED BETWEEN PASSES



A Quick Overview of Compact Air-Cooled Heat Sinks

Applicable for Electronic Cooling—Recent Progress, Inventions 2017, 2, 5; doi:10.3390/inventions2010005

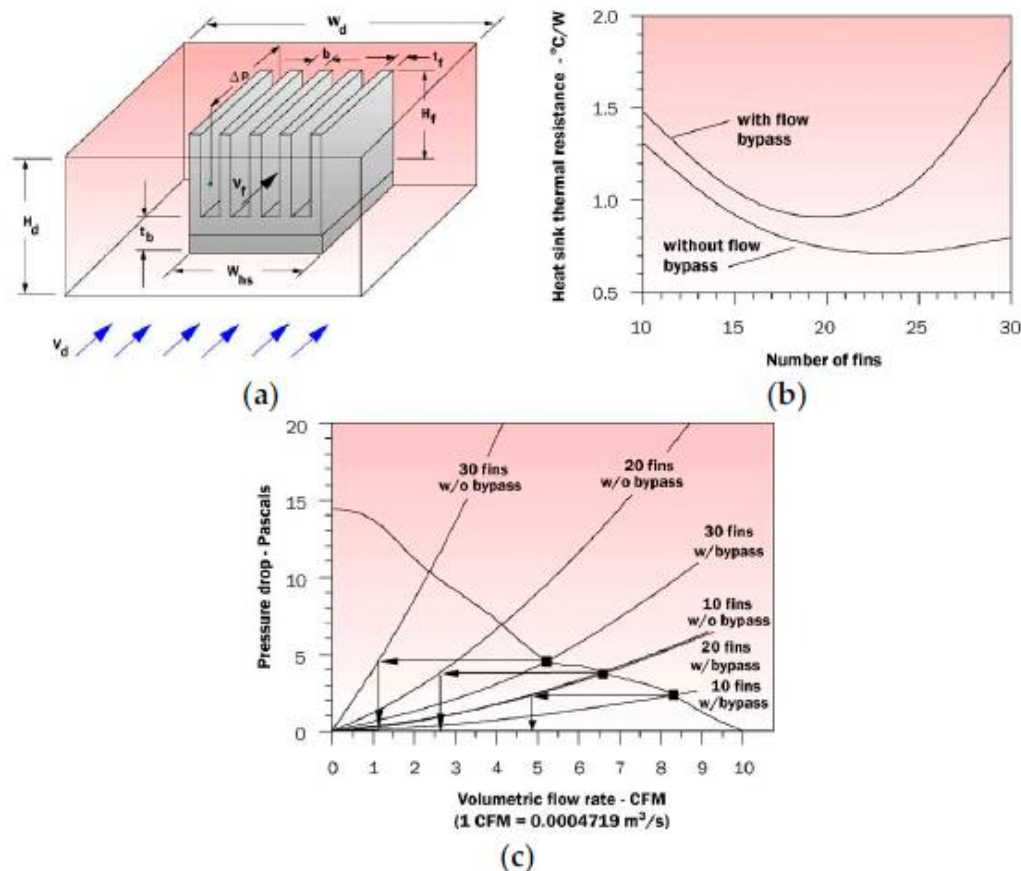
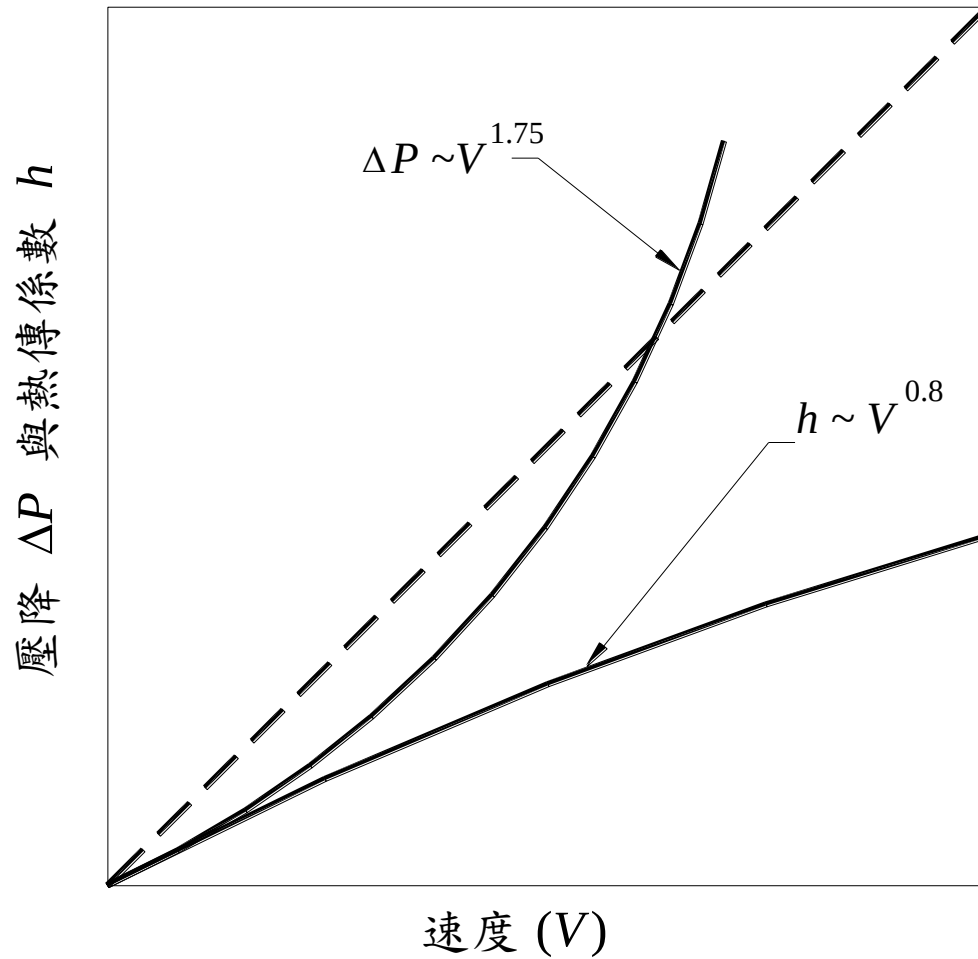


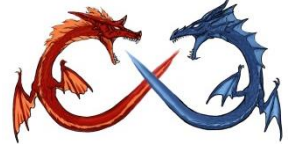
Figure 2. Effect of bypass and fin number on the thermal performance of a heat sink [16] with permission from Electronics Cooling / ITEM Media. (a) Schematic of airflow into a parallel plate fin heat sink with flow bypass; (b) Effect of number of fins on heat sink thermal resistance with and without flow bypass; (c) Heat sink pressure drop curves (with and without flow bypass) and fan curve with flow operating points. CFM is flow rate in English unit (cubic feet per minute).



熱交換器設計是否僅考慮熱傳量？



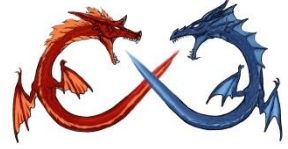
熱傳係數、壓降與速度的關係示意圖



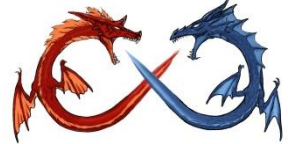
熱傳增強管與熱傳增強鰭片使用的迷思

熱交換器性能評價方法 (Performance evaluation criteria)，使用熱傳增強管與熱傳增強鰭片來取代原有熱交換器的目的有四：

- 在維持相同的熱傳量且不增加壓降的條件下，如何減少熱交換器的面積。
- 在維持相同的熱傳量與原有熱交換器的面積下，如何降低流體間的有效溫差(或 $LMTD$)。
- 在維持原有熱交換器的面積下，如何增加熱傳量。
- 在維持相同的熱傳量與原有熱交換器的面積下，如何減少流體通過熱交換器的壓降(即減少pumping power)。



- *FG*法則 (fixed geometry criteria)：適用於截面積與管長固定條件。*FG*法則可用於1對1直接置換式的應用，因為新的熱交換器與舊有的熱交換器一樣大。
- *FN*法則 (fixed number of tubes geometry criteria)：適用於截面積固定，但管長可允許變化。
- *VG*法則 (variable geometry criteria)：適用於熱傳量固定，但要適度的減少熱交換器的面積。

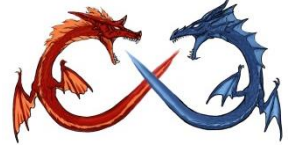


PEC的應用法則與目的

法則	幾何參數	固定參數				目的
		$\dot{m}$	W	Q	ΔT	
FG-1a	N, L	☆			☆	$\uparrow Q$
FG-1b	N, L	☆		☆		$\downarrow \Delta T$
FG-2a	N, L		☆		☆	$\uparrow Q$
FG-2b	N, L		☆	☆		$\downarrow \Delta T$
FG-3	N, L			☆	☆	$\downarrow W$
FN-1	N		☆	☆	☆	$\downarrow L$
FN-2	N	☆		☆	☆	$\downarrow L$
FN-3	N	☆		☆	☆	$\downarrow W$
VG-1		☆	☆	☆	☆	$\downarrow NL$
VG-2a	N, L	☆	☆		☆	$\uparrow Q$
VG-2b	N, L	☆	☆	☆		$\downarrow \Delta T$
VG-3	N, L	☆		☆	☆	$\downarrow W$

在 FG 法中，傳熱管支數 N 與管長 L 均為固定

在 VG-2與VG-3法則中， $N \times L$ 值為固定



$$h = c_{p,a} \text{Pr}^{-2/3} j G \quad (7-3)$$

由於PEC的比對是以熱傳增強表面(enhanced surface)替換原有的參考表面(referenced surface)，因此吾人有興趣的熱傳特性與原有的參考表面比值如下：

$$\frac{hA}{h_{ref} A_{ref}} = \frac{j}{j_{ref}} \frac{A}{A_{ref}} \frac{G}{G_{ref}} \quad (7-4)$$

同時，推動流體所需要的消耗功率可表示如下：

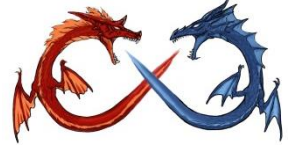
$$W = \Delta P \times \dot{V} = \left(\frac{fA}{A_c} \frac{G^2}{2\rho} \right) \left(\frac{GA_c}{\rho} \right) \quad (7-5)$$

因此，新的熱傳增強表面與參考表面的耗功比值為：

$$\frac{W}{W_{ref}} = \frac{\left(\frac{fA}{A_c} \frac{G^2}{2\rho} \right) \left(\frac{GA_c}{\rho} \right)}{\left(\frac{f_{ref} A_{ref}}{A_{c,ref}} \frac{G_{ref}^2}{2\rho} \right) \left(\frac{G_{ref} A_{c,ref}}{\rho} \right)} = \frac{f}{f_{ref}} \frac{A}{A_{ref}} \left(\frac{G}{G_{ref}} \right)^3 \quad (7-6)$$

從式7-4與式7-6，可將 G/G_{ref} 消去，因而得到如下的方程式：

$$\frac{\frac{hA}{h_{ref} A_{ref}}}{\left(\frac{W}{W_{ref}} \right)^{1/3} \left(\frac{A}{A_{ref}} \right)^{2/3}} = \frac{\frac{j}{j_{ref}}}{\left(\frac{f}{f_{ref}} \right)^{1/3}} \quad (7-7)$$



VG-1法則

Q/Q_{ref} 與 W/W_{ref} 是固定的，我們使用新型熱傳增強熱交換器的目的是希望可以適度的減少熱交換器的面積。由 $Q/Q_{ref} = 1$ ， $hA/h_{ref}A_{ref} = 1$ 與 $W/W_{ref} = 1$ ，則式7-7可簡化如下：

$$\frac{\frac{hA}{h_{ref}A_{ref}}}{\left(\frac{W}{W_{ref}}\right)^{1/3} \left(\frac{A}{A_{ref}}\right)^{2/3}} = \frac{\frac{j}{j_{ref}}}{\left(\frac{f}{f_{ref}}\right)^{1/3}} \Rightarrow \frac{A}{A_{ref}} = \left(\frac{f}{f_{ref}}\right)^{1/2} \left(\frac{j_{ref}}{j}\right)^{3/2} \quad (7-8)$$

另外將式7-8代入式7-6可得到：

$$1 = \frac{f}{f_{ref}} \frac{A}{A_{ref}} \left(\frac{G}{G_{ref}}\right)^3 = \frac{f}{f_{ref}} \left(\frac{f}{f_{ref}}\right)^{1/2} \left(\frac{j_{ref}}{j}\right)^{3/2} \left(\frac{G}{G_{ref}}\right)^3 \quad (7-9)$$

$$\Rightarrow \frac{G}{G_{ref}} = \left(\frac{j}{j_{ref}} \frac{f_{ref}}{f}\right)^{1/2} \quad (7-10)$$



Fundamentals of heat exchanger design, chapter 10

TABLE 10.6 Continued

Case	Determine:	Information or Constraints Given and Comments	Resultant Formulas to Compute the Objective of the PEC
FG-2b	$\frac{\Delta T_{\max}}{\Delta T_{\max,p}}$	$\frac{n_t}{n_{t,p}} = \frac{A_{fr}}{A_{fr,p}} = 1$ $\frac{L}{L_p} = \frac{A}{A_p} = 1$ $\frac{\mathcal{P}}{\mathcal{P}_p} = 1$ $\frac{q}{q_p} = 1$	Compute $UA/(UA)_p$ as outlined for the FG-2a case. Compute NTU from Eq. (10.29) and subsequently, P, for the given flow arrangement. Then from Eq. (10.28), $\frac{\Delta T_{\max}}{\Delta T_{\max,p}} = \frac{P_p}{P} \frac{\dot{m}_p}{\dot{m}}$
FG-3	$\frac{\mathcal{P}}{\mathcal{P}_p}$	$\frac{n_t}{n_{t,p}} = \frac{A_{fr}}{A_{fr,p}} = 1$ $\frac{L}{L_p} = \frac{A}{A_p} = 1$ $\frac{q}{q_p} = \frac{\Delta T_{\max}}{\Delta T_{\max,p}} = 1$ $\frac{\dot{m}}{\dot{m}_p} = \frac{G}{G_p}$	From Eqs. (10.26), (10.24), and (10.28) $\frac{UA}{(UA)_p} = \frac{1 + \mathbf{R}_p^*}{(j_p/j)[(f/f_p)(\mathcal{P}_p/\mathcal{P})]^{1/3} + \mathbf{R}^*} \quad (1)$ $\frac{\mathcal{P}}{\mathcal{P}_p} = \frac{f}{f_p} \left(\frac{G}{G_p} \right)^3 \quad (2)$ $\frac{q}{q_p} = 1 = \frac{G}{G_p} \frac{P}{P_p} \quad (3)$ From these equations, compute G/G_p as follows: (1) Assume G/G_p. (2) Solve for $\mathcal{P}/\mathcal{P}_p$ from Eq. (2). (3) Compute $UA/(UA)_p$ from Eq. (1). (4) Calculate NTU from Eq. (10.28). (5) Determine P and P_p for the exchanger flow arrangement given. (6) compute q/q_p from Eq. (3) which would be different from unity until convergence. Iterate on G/G_p to obtain $q/q_p = 1$. Using converged value of G/G_p, calculate $\mathcal{P}/\mathcal{P}_p$ from Eq. (2).
FN-1	$\frac{L}{L_p}$	$\frac{n_t}{n_{t,p}} = \frac{A_{fr}}{A_{fr,p}} = 1$ $\frac{A}{A_p} = \frac{L}{L_p}$ $\frac{\dot{m}}{\dot{m}_p} = \frac{G}{G_p}$ $\frac{\mathcal{P}}{\mathcal{P}_p} = \frac{q}{q_p} = \frac{\Delta T_{\max}}{\Delta T_{\max,p}} = 1$	From Eqs. (10.26), (10.24), and (10.28) $\frac{UA}{(UA)_p} = \frac{1 + \mathbf{R}_p^*}{(j_p/j)(f/f_p)^{1/3}(L_p/L)^{2/3} + \mathbf{R}^*(L_p/L)}$ $\frac{\mathcal{P}}{\mathcal{P}_p} = 1 = \frac{f}{f_p} \frac{L}{L_p} \left(\frac{G}{G_p} \right)^3, \quad \frac{q}{q_p} = 1 = \frac{G}{G_p} \frac{P}{P_p}$

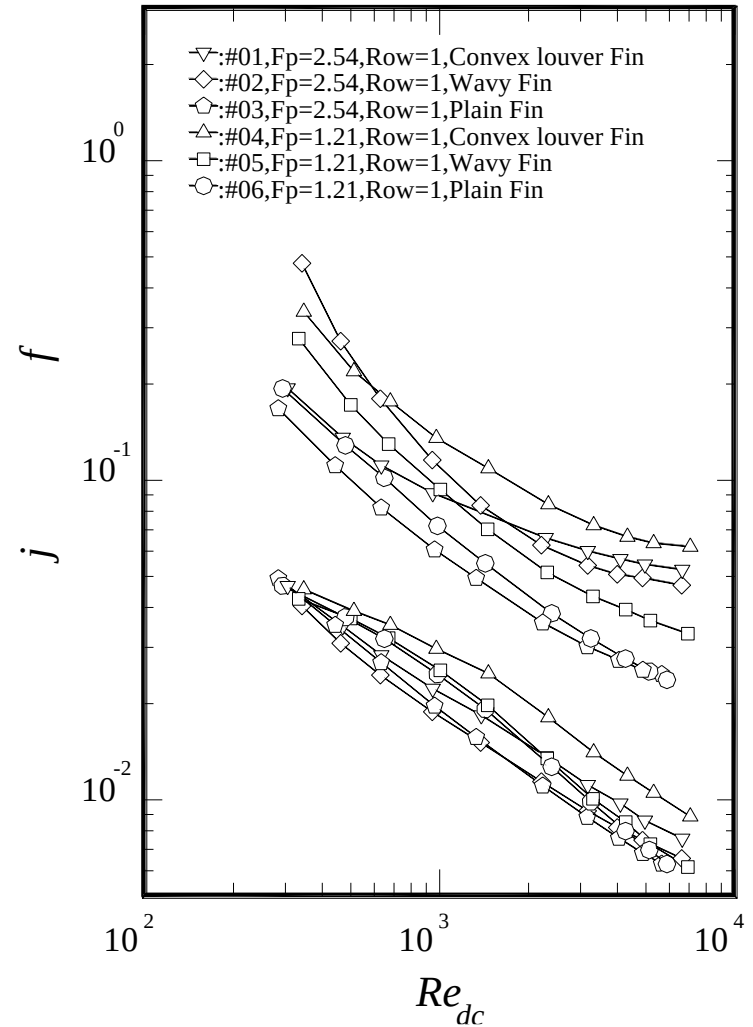
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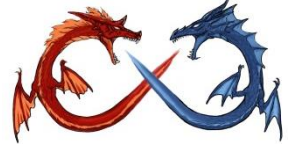
其他常見的熱交換器評價方法

熱交換器性能的評價方法相當的多，這些方法可大致歸納如下：

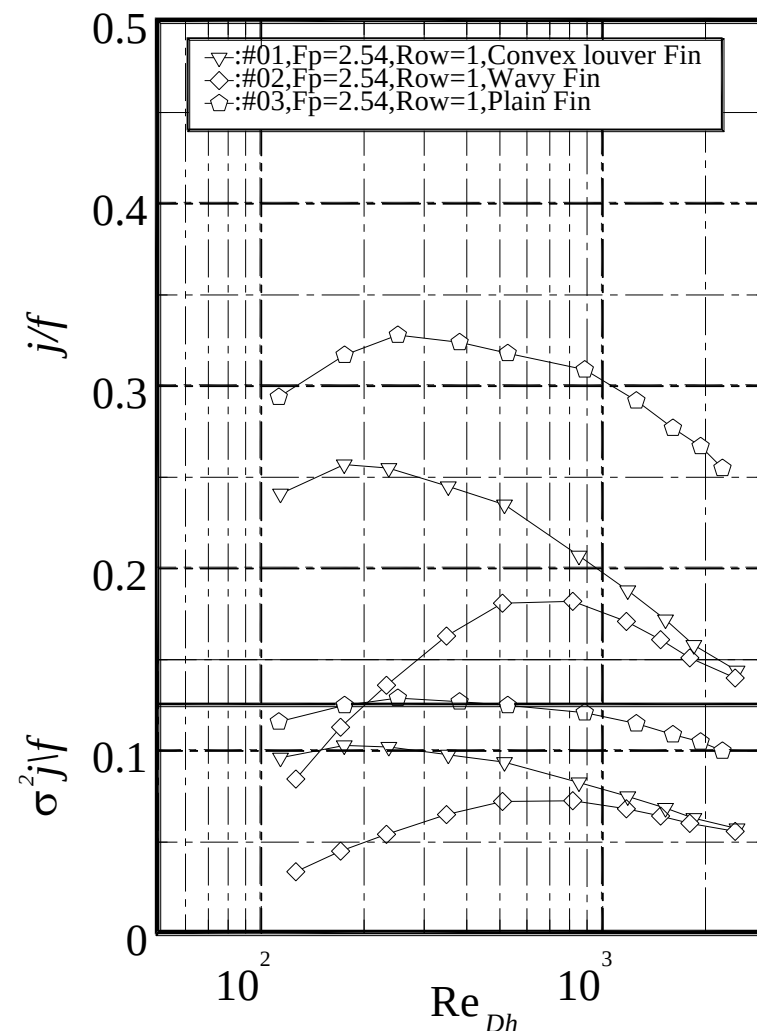
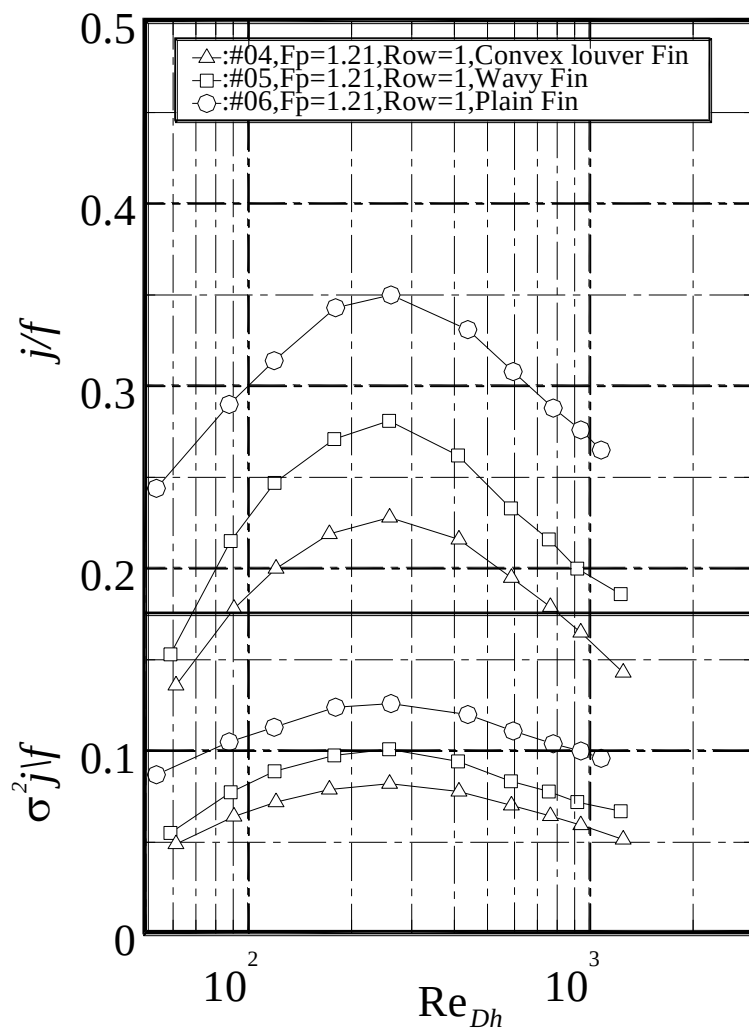
- 直接評價 j 與 f ，作為判別。
- 評價熱傳量與推動流體所需的動力的相互關聯，作為評價基礎。
- 與一參考表面或熱傳管做相對性的評價(如前面介紹的Webb的方法)。
- 其他評價方法(例如Cowell, 1990的方法)。

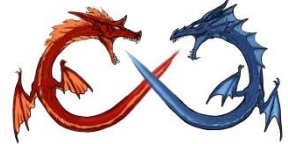


直接熱流特性來評價



複合式百葉窗片(convex-louver)、波浪鰭片(wavy)與平板型鰭片(plain)以 j/f 性能指標來評價



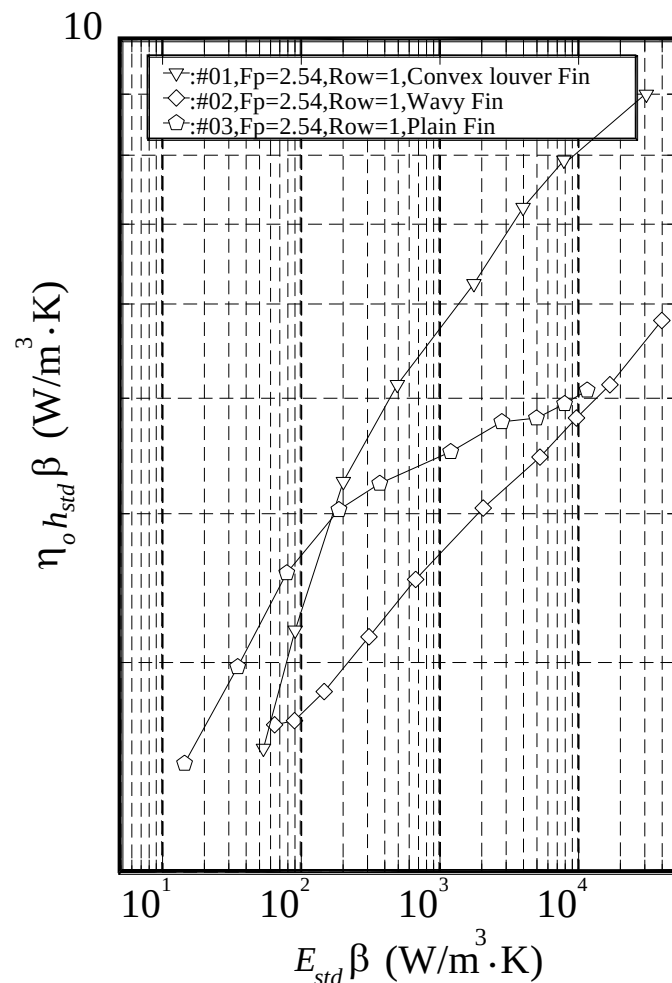
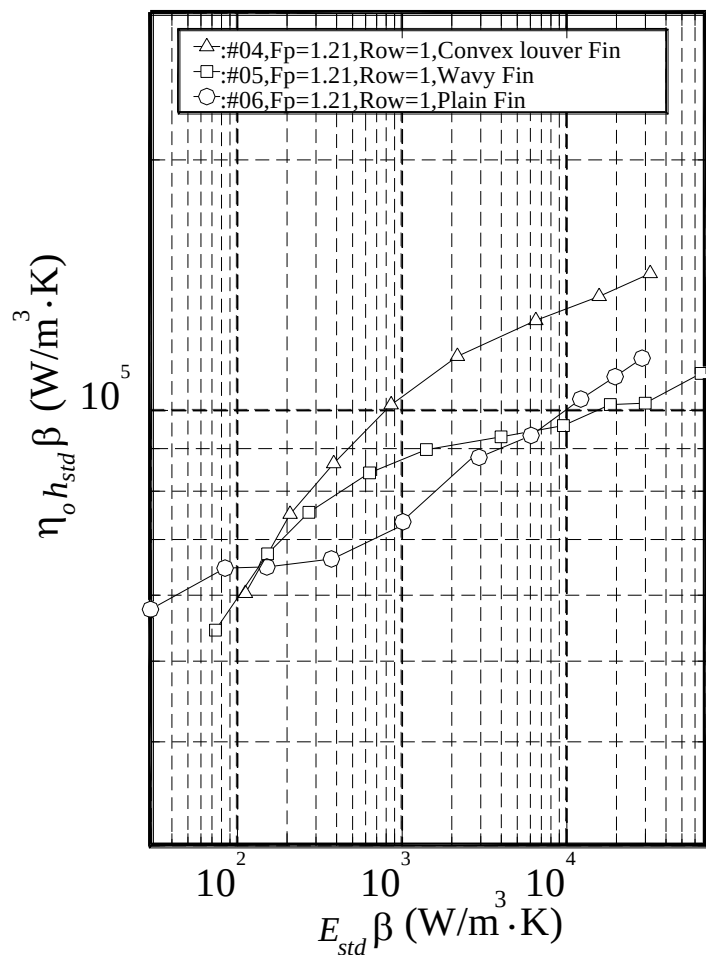


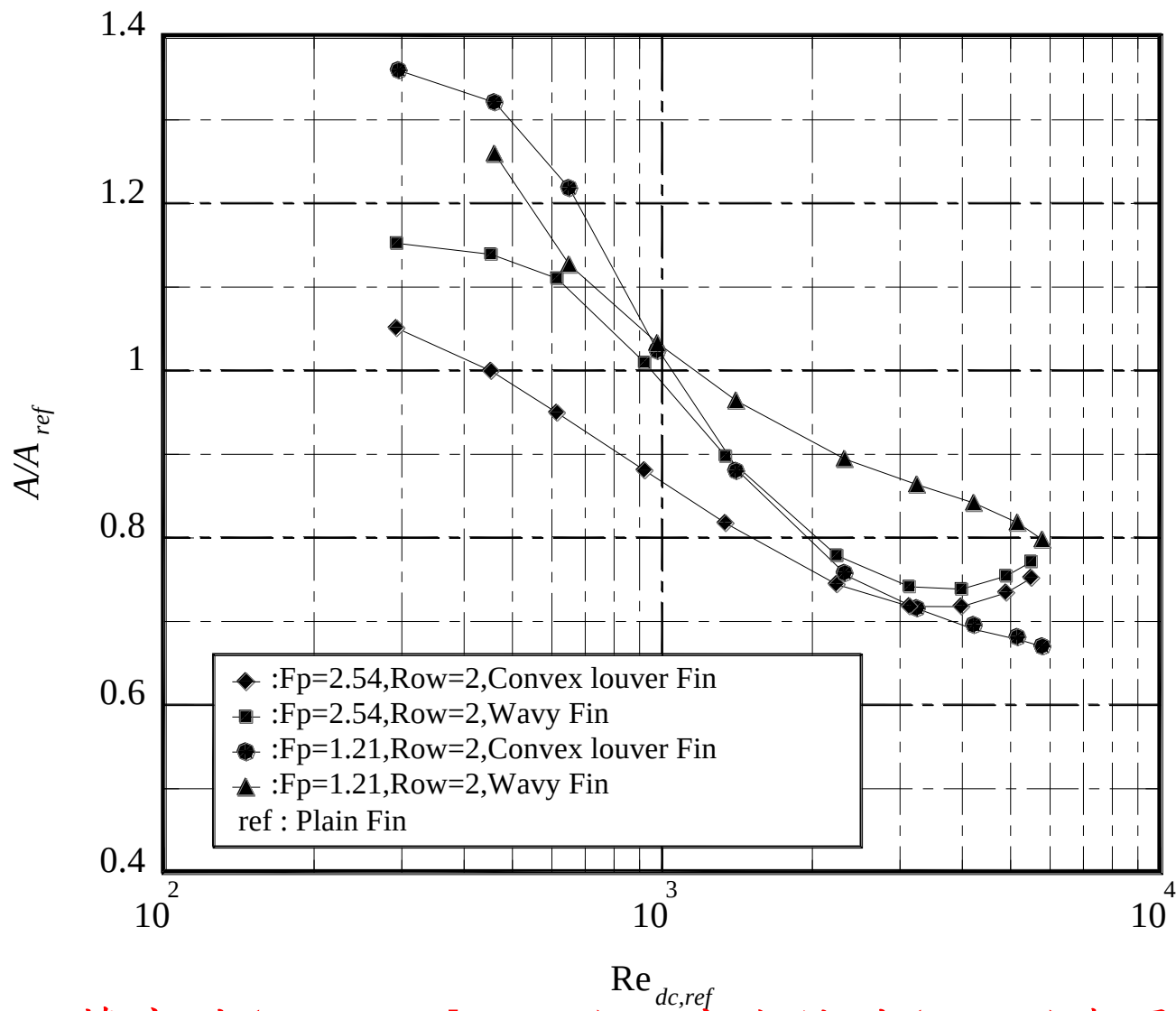
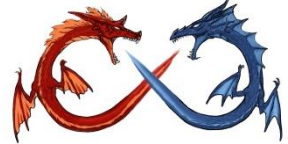
另外一種常用的評價方法為「體積優先」評價法(volume goodness factor comparison, Kays and London ,1950)，採用的評價參數為：

$$\eta_o h_{std} \beta = \frac{c_p \mu}{Pr^{2/3}} \eta_o \frac{4\sigma}{D_h^2} j Re \quad (7-17)$$

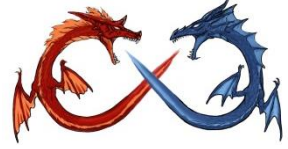
$$E_{std} \beta = \frac{\mu^3}{2g_c \rho^2} \frac{4\sigma}{D_h^4} f Re^3 \quad (7-18)$$

複合式百葉窗片(convex-louver)、波浪鰭片(wavy)與平板型鰭片(plain)單位體積、溫差下的熱傳能力與單位體積的摩擦消耗的關係圖



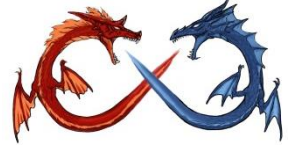


複合式百葉窗片(convex-louver)、波浪鰭片(wavy)與平板型鰭片(plain)以 VG-1法則的性能評價

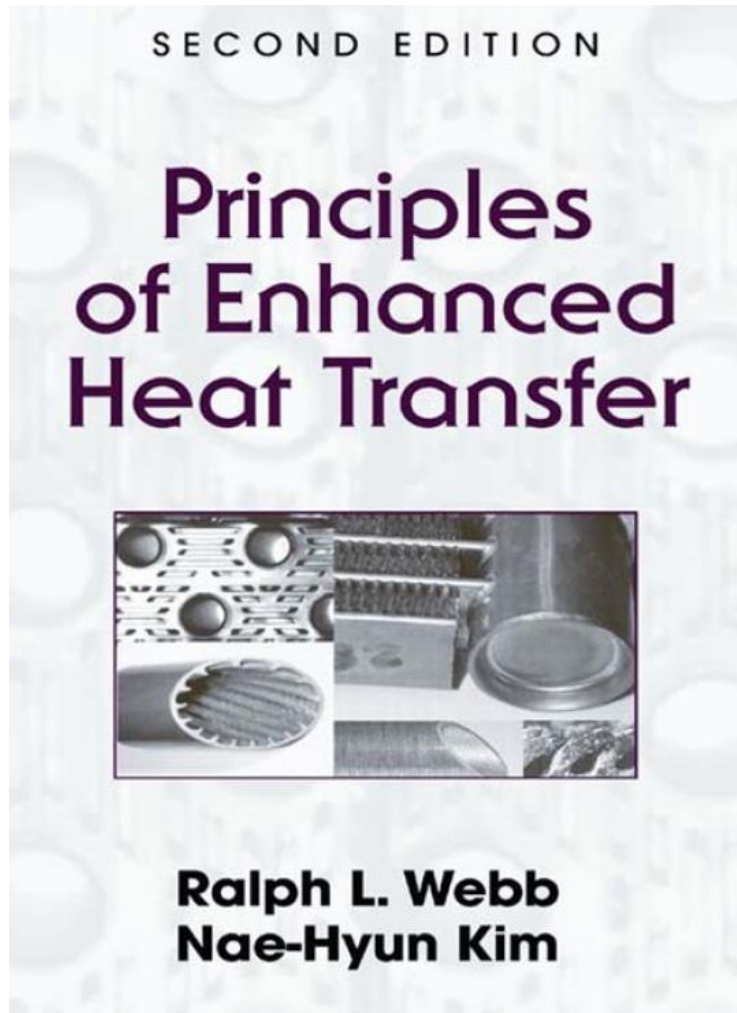


Classification of enhancement techniques (active methods)

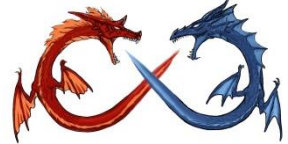
- Mechanical Aids: Examples of the mechanical aids include rotating tube exchangers and scrapped surface heat and mass exchangers.
- Surface vibration.
- Fluid vibration.
- Electrostatic fields.
- Injection.
- Suction.



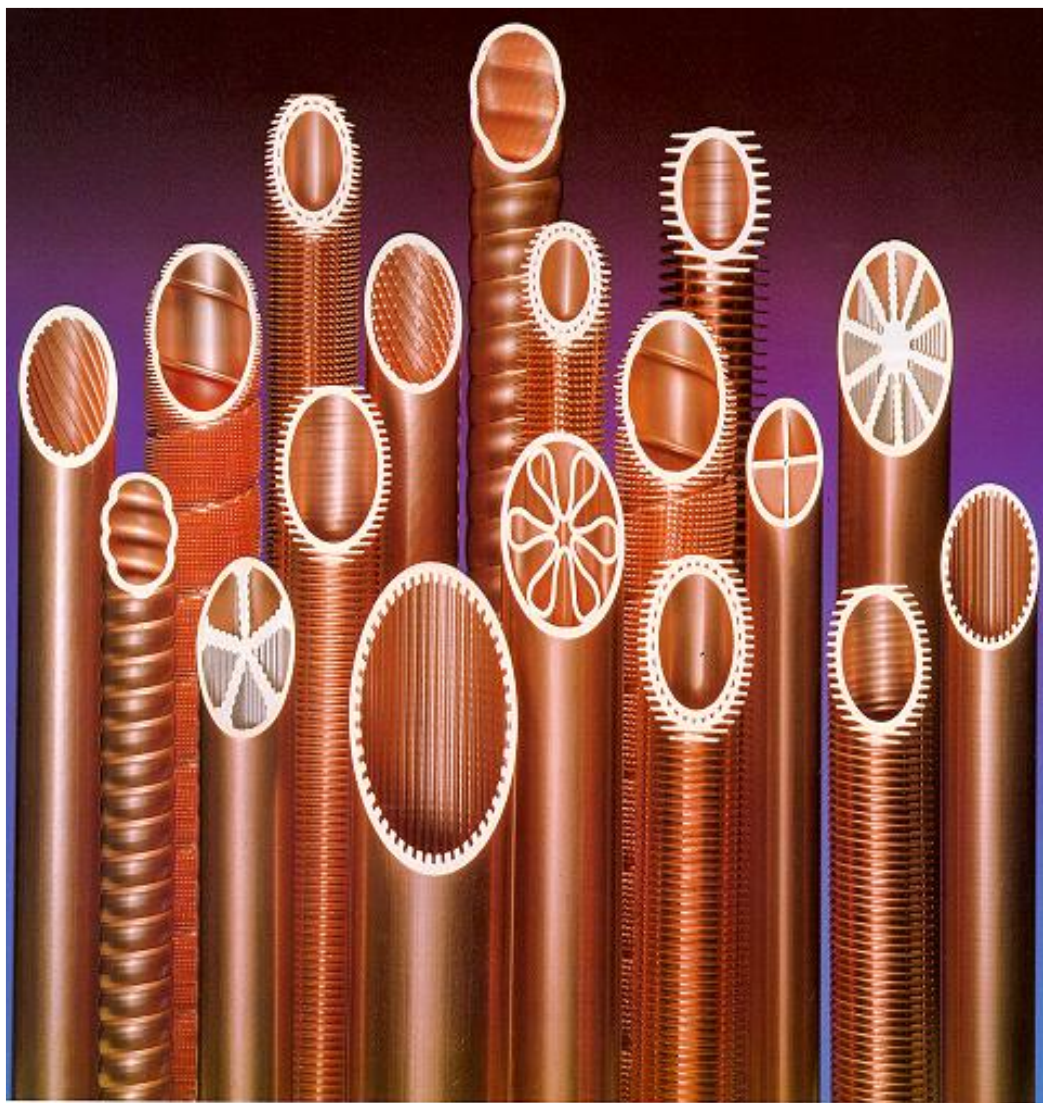
Passive enhancement methods..

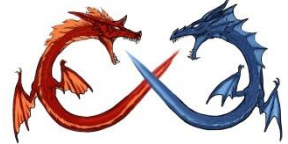


- **Treated Surfaces:** They are heat transfer surfaces that have a fine-scale alteration to their finish or coating. The alteration could be continuous or discontinuous, where the roughness is much smaller than what affects single-phase heat transfer, and they are used primarily for boiling and condensing duties.
- **Rough surfaces:** They are generally surface modifications that promote turbulence in the flow field, primarily in single-phase flows, and do not increase the heat transfer surface area. Their geometric features range from random sand-grain roughness to discrete three-dimensional surface protuberances.
- **Extended surfaces:** They provide effective heat transfer enlargement. The newer developments have led to modified fin surfaces that also tend to improve the heat transfer coefficients by disturbing the flow field in addition to increasing the surface area.
- **Displaced enhancement devices:** These are the insert techniques that are used primarily in confined force convection. These devices improve the energy transfer indirectly at the heat exchange surface by displacing the fluid from the heated or cooled surface of the duct/pipe with bulk fluid to the core flow.
- **Swirl flow devices:** They produce and superimpose swirl flow or secondary recirculation on the axial flow in a channel. These devices include helical strip or cored screw type tube inserts, twisted tapes. They can be used for single phase or two-phase flows heat exchanger.
- **Coiled tubes:** These techniques are suitable for relatively more compact heat exchangers. Coiled tubes produce secondary flows and vortices which promote higher heat transfer coefficient in single phase flow as well as in most boiling regions.
- **Surface tension devices:** These consist of wicking or grooved surfaces, which directly improve the boiling and condensing surface. These devices are most used for heat exchanger occurring phase transformation.
- **Additives for liquids:** These include the addition of solid particles, soluble trace additives and gas bubbles into single phase flows and trace additives which usually depress the surface tension of the liquid for boiling systems.
- **Additives for gases:** These include liquid droplets or solid particles, which are introduced in single-phase gas flows either as dilute phase (gas–solid suspensions) or as dense phase (fluidized beds).



熱傳增強管 – 如何使用？

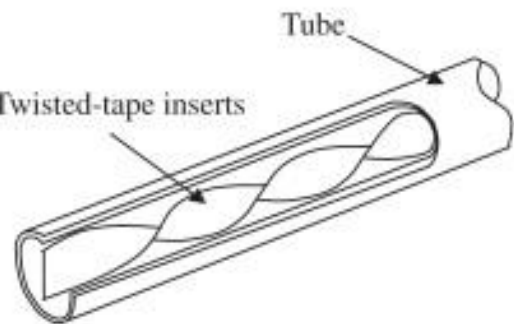
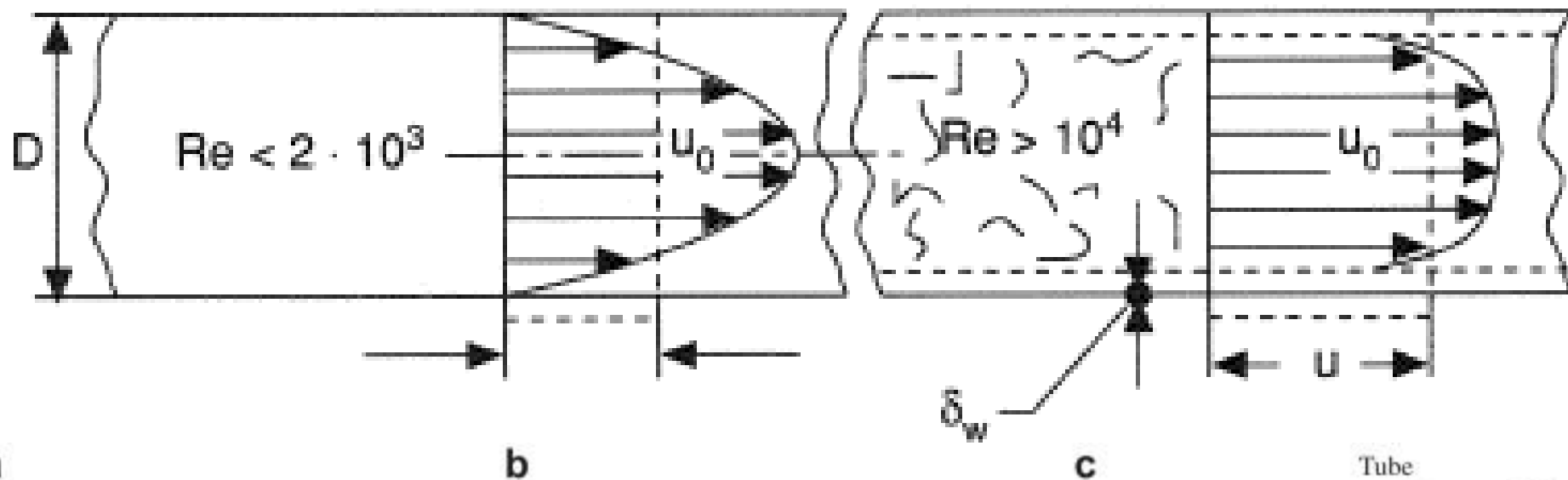




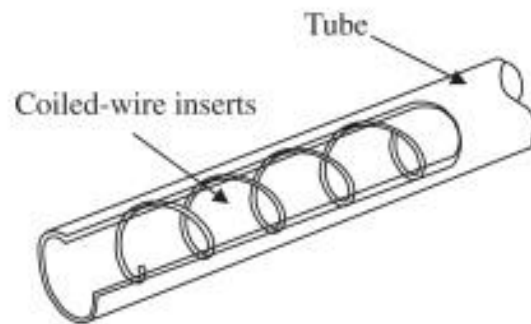
層流與紊流之熱傳增強概念

層流

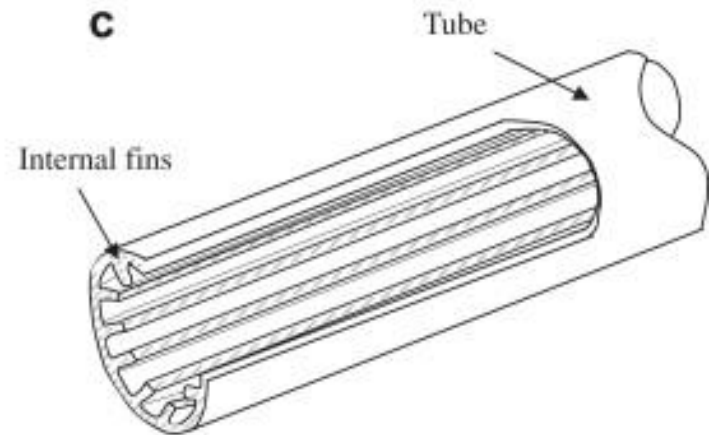
紊流



Twisted-tape inserts



Coiled-wire inserts



Internal fins



Review of heat transfer enhancement methods: Focus on passive methods using swirl flow devices, Renewable and Sustainable Energy Reviews 49 (2015) 444–469



Fig. 17. Photo of the novel enhanced heat transfer tube [24].

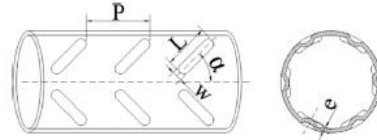


Fig. 19. Structure of the DDIR tube [26].

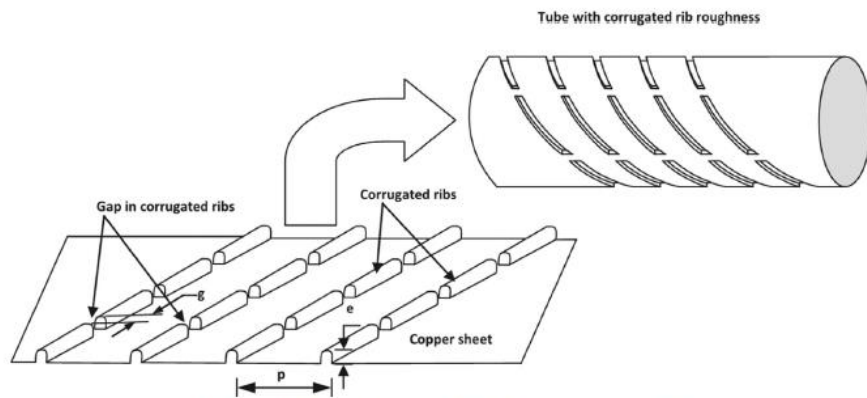


Fig. 18. Typical geometry of circular tube with discrete corrugated ribs [25].

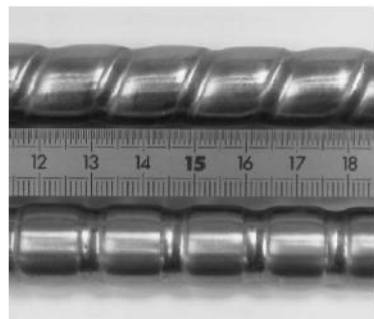


Fig. 15. Helical and transverse corrugated tubes tested [20].

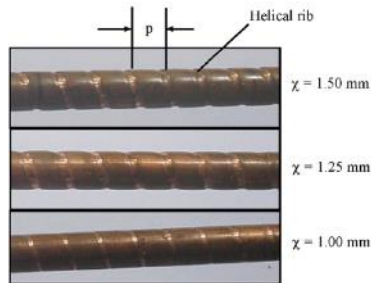


Fig. 16. Photograph of the tube with helical rib used in the present study [23].

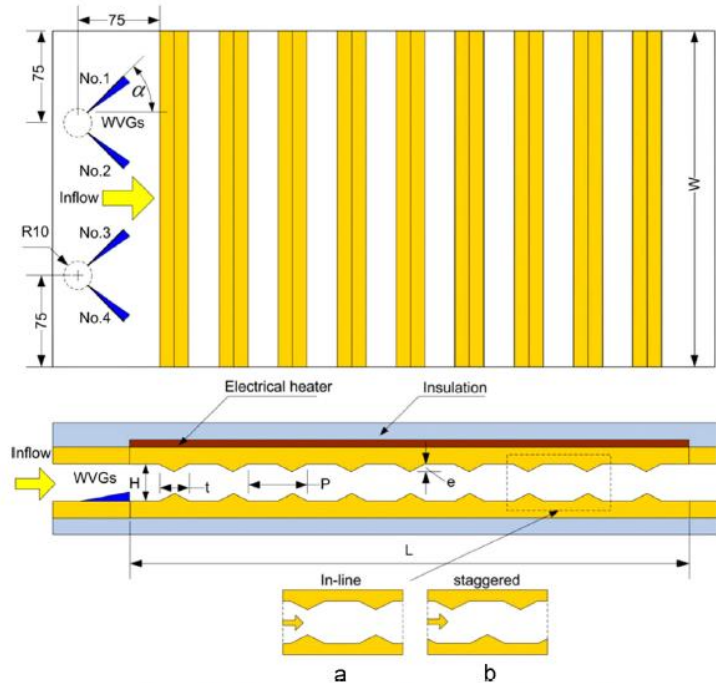


Fig. 20. Test section with WVGs: (a) in-line and (b) staggered rib arrangements [27].

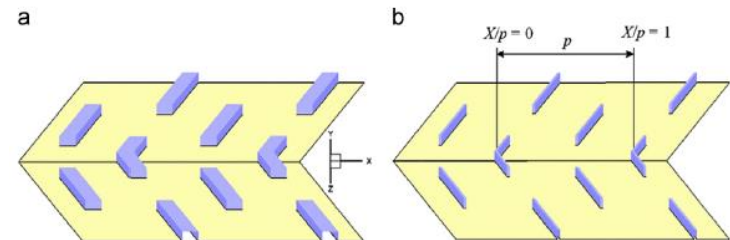


Fig. 21. Arrangement of V-discrete (a) square and (b) thin ribs on lower wall [28].

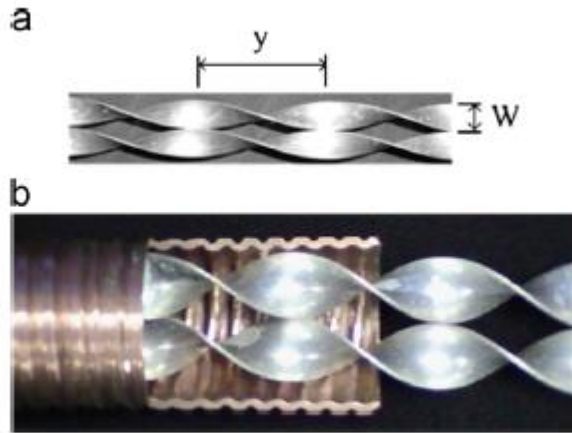
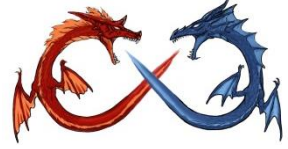


Fig. 30. Details of (a) double twisted tapes and (b) helical-ribbed tube with double twisted tape insert [40].

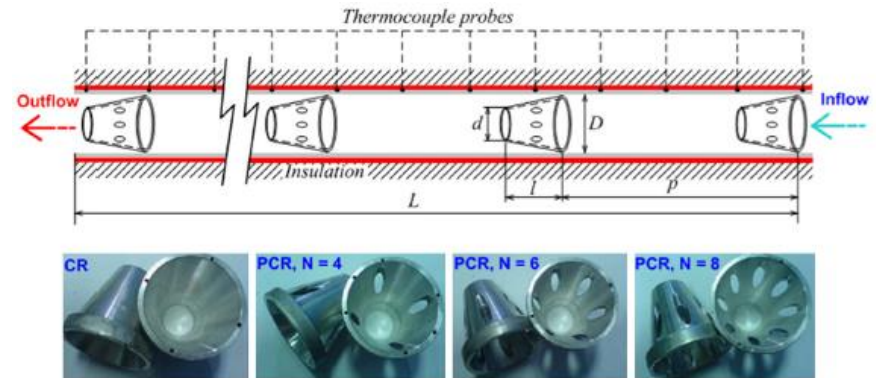


Fig. 40. The arrangement of CR/PCR in a round tube [60].

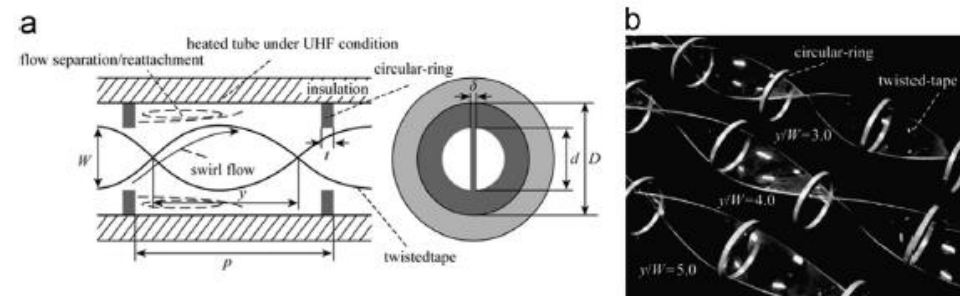


Fig. 45. Diagram and photograph of circular-rings combined with a twisted tape [67].

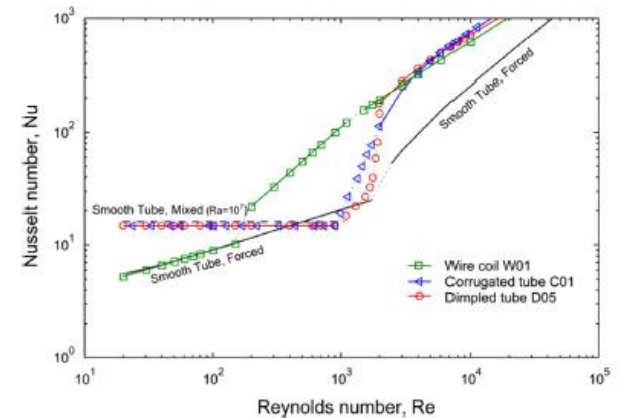


Fig. 48. Nusselt number vs. Reynolds number. Experimental correlations for the wire coil, the corrugated tube and the dimpled tube [68].

Full length twisted tape increases the pressure drop comparing to an empty tube. Most of the researches need to reduce the extra pressure drop by using short length twisted tape. Twisted tape in turbulent flow insert is not very effective.



Heat transfer augmentation in a circular tube with perforated double counter twisted tape inserts☆

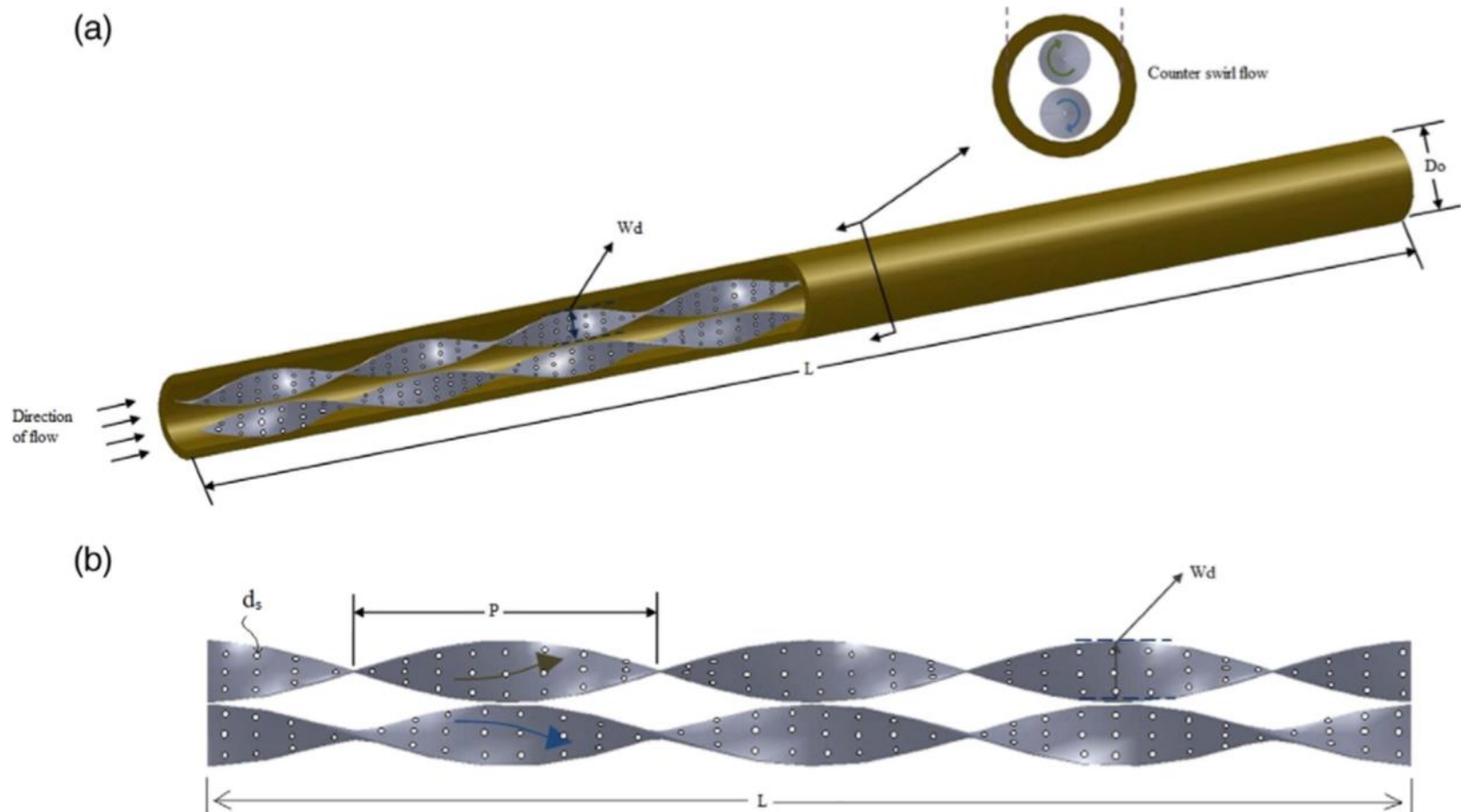


Fig. 2. 2(a) Geometry of test section fitted with perforated double counter twisted tape insert; 2(b) Geometric parameters of the perforated double counter twisted tape insert.

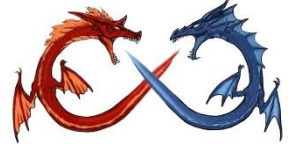
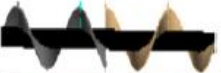
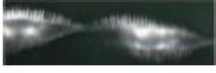
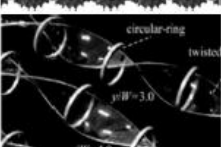
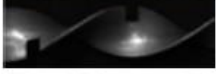
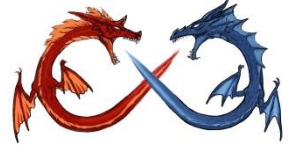


Table 2
Configuration sketches of the stationary twisted tapes.

Configuration	Name	Reference	Configuration	Name	Reference
	Typical twisted tape	[34–42]		Twisted-tape with V-winglets	[43]
	Perforated twisted tape	[44,45]		Left-right twisted tape	[46]
	Notched twisted tape	[47]		Right and left helical screw-tape	[48]
	Jagged twisted tape	[47]		Helically twisted tape	[49]
	Twisted tape consisting wire nails	[50]		Perforated helical twisted-tape	[51]
	Center-deared twisted tape	[52–56]		Twisted tape with geometrical progression ratio	[57]
	V-cut twisted tape	[58]		Double twisted tape	[59–63]
	Twisted-tape with alternate axes	[64,65]		Triple twisted tape	[66]
	Serrated twisted tape	[67]		Ribbed spiky twist tape	[68,69]
	Twisted tape consisting of centre wings and alternate-axes	[70]		Twisted tape with circular-rings	[71]
	Peripherally-cut twisted tape with an alternate axis	[72]		Staggered twisted tape with central holes	[73]
	Square-cut twisted tape	[74–76]		Elliptic-cut twisted tape	[77]
	Helical screw tape	[78–80]		Regularly-spaced twisted tape	[81]

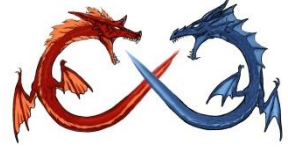
Twisted tape with different geometries can offer different performances and mechanisms of heat transfer enhancement. With the purpose of creating stronger turbulence intensity in the vicinity of tube wall or reducing the pressure drop to a required level, slots, holes, cuts, gaps, etc. are manufactured based on the structure of typical twisted tape.



Twisted tape (types)

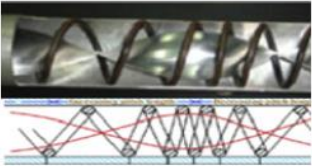
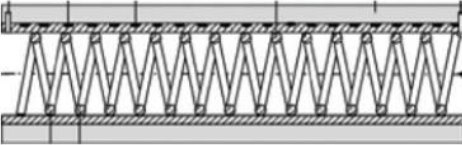
(Usually for single phase applications)

- Typical twisted tape
- Varying length, alternate-axes and pitches twisted tape
- Multiple twisted tapes
- Twisted tape with rod and varying spacer
- Twisted tape with attached fins and baffles
- Twisted tapes with slots, holes, cuts (create more turbulence)
- Helical left–right twisted tape with screw
- Tapes with different surface modifications

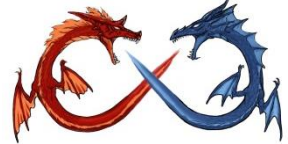


A comprehensive review on passive heat transfer enhancements in pipe exchangers, Renewable and Sustainable Energy Reviews 19 (2013) 64–81

Table 3
Configuration sketches of various wire coils various.

Configuration	Name	Reference	Configuration	Name	Reference
	Coiled square wires	[40]		Twisted tape and wire coil	[42]
	Non-uniform wire coil combined with twisted tape	[43]		Triangle cross sectioned coiled wire	[44]
	Coiled wire turbulators	[39]		Wire coil in pipe	[46]

Wire coils is especially for air side (in tube) augmentation



Heat transfer augmentation using twisted tape inserts: A review, Renewable and Sustainable Energy Reviews 63 (2016) 193–225

typical 1-phase enhancement is less than 2

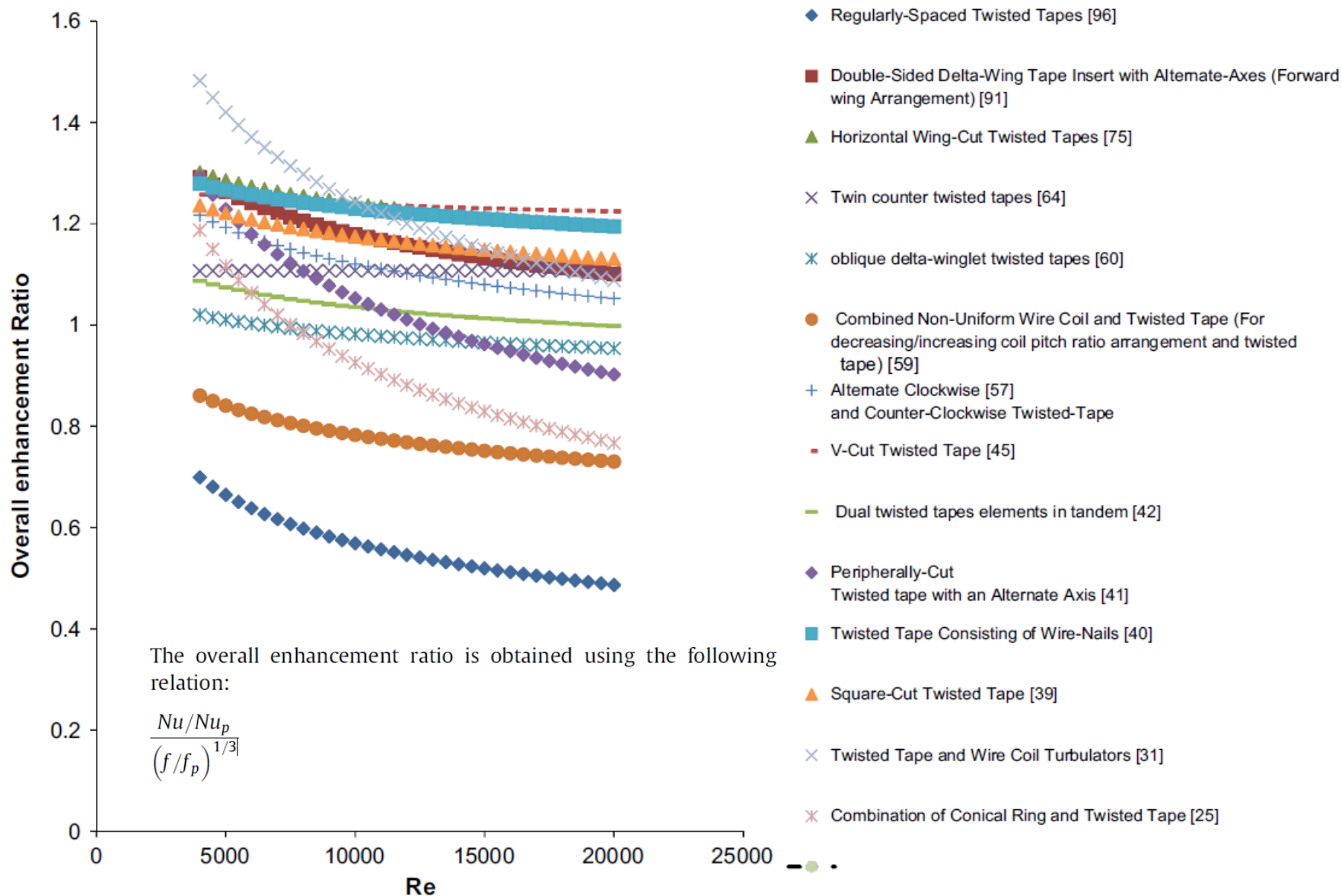


Fig. 34. Overall enhancement ratio vs. Reynolds number for different twisted tape geometries.

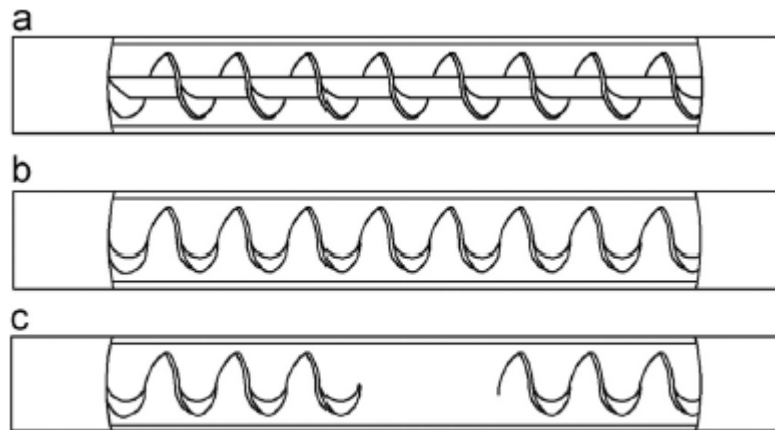


Fig. 4. (a) Full length helical tape with a centred rod, (b) full length helical tape without centred rod, (c) regularly spaced helical tapes without rod [17].

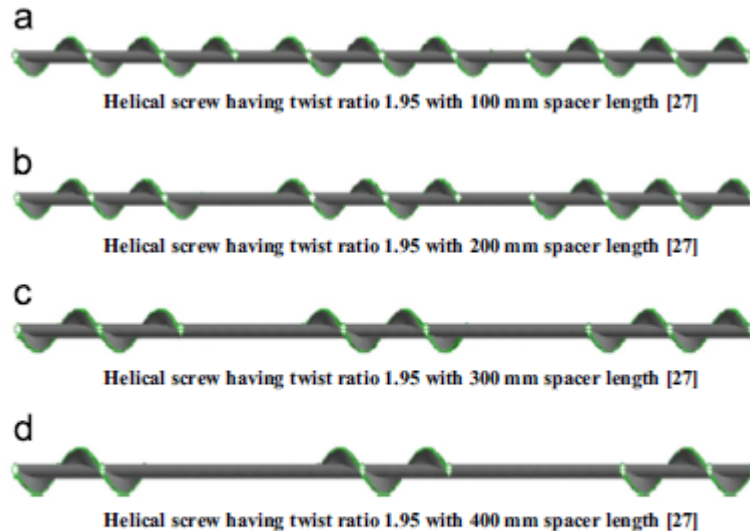


Fig. 9. (a) Helical screw having twist ratio 1.95 with 100 mm spacer length [27]. (b) helical screw having twist ratio 1.95 with 200 mm spacer length [27]. (c) helical screw having twist ratio 1.95 with 300 mm spacer length [27]. (d) helical screw having twist ratio 1.95 with 400 mm spacer length [27].

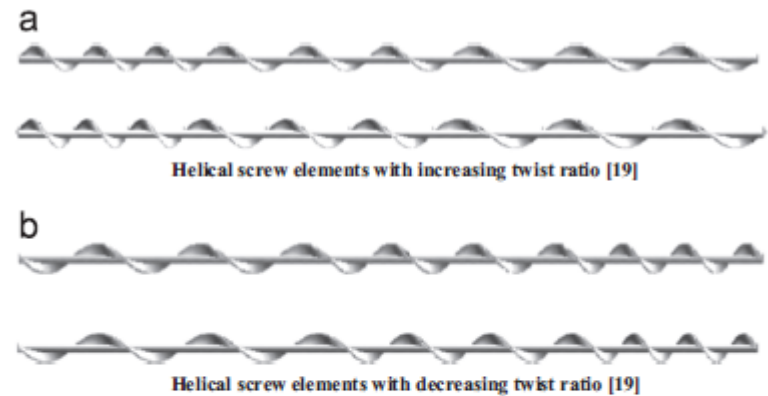


Fig. 5. (a) Helical screw elements with increasing twist ratio [19] (b) helical screw elements with decreasing twist ratio [19].

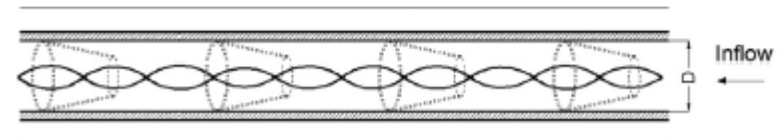


Fig. 6. Conical ring turbulator and twisted tape [23].



Fig. 10. Tube having twisted tape inserts and wire coil [28].

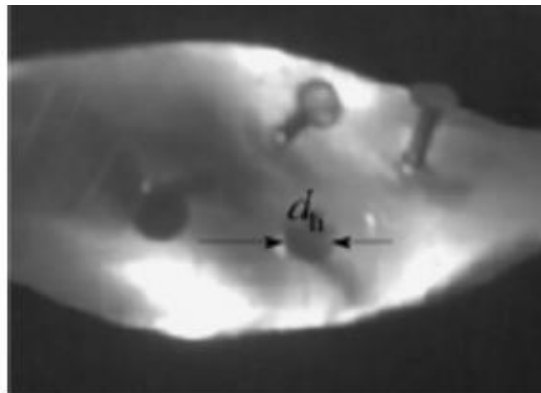
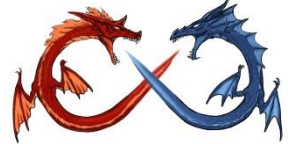
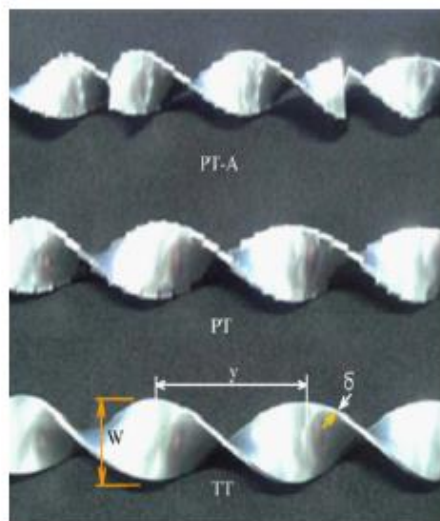


Fig. 15. Twisted tape having wire nails studied by Murugesan et al. [36].



PT-A: Peripherally-cut twisted tapes with alternate axis
 PT: Peripherally-cut twisted tapes
 TT: Typical twisted tapes
 y: Twist length
 W: Width of twisted tape
 δ : Thickness of twisted tape

Fig. 16. Twisted tape studied by Seemawute and Eiamsa-ard [37].



Fig. 22. rectangular cut twisted tape tested [48].

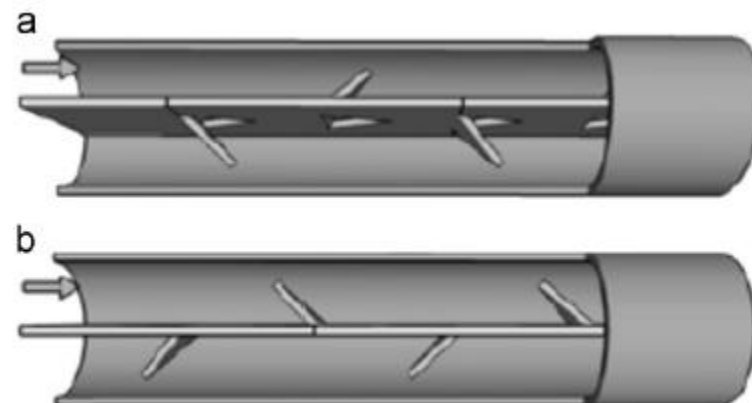


Fig. 31. (a) Backward wing arrangement (b) Forward wing arrangement (Eiamsa-ard and Promvong [87]).



Fig. 24. Decreasing/increasing coil pitch ratio arrangement and twisted tape [54].



Fig. 18. Perforated twisted tapes with parallel wings [40].

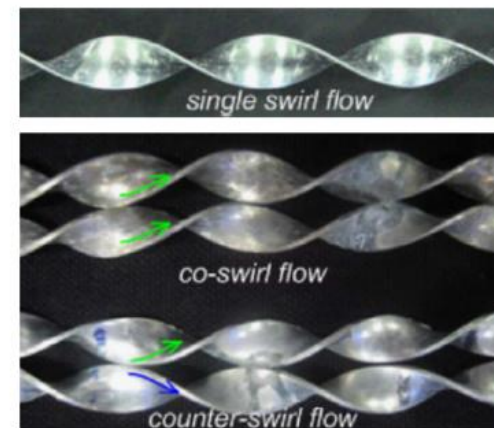
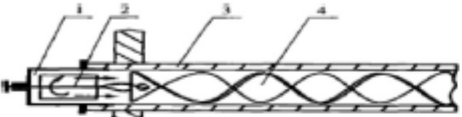
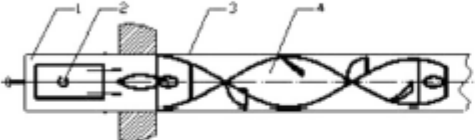
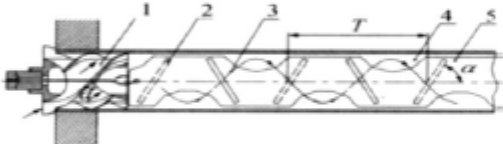


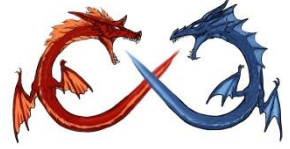
Fig. 25. Single twisted tape (single swirl flow), twin co-twisted tape (co swirl flow) and twin counter twisted tapes (counter-swirl flow) [59].



$$\eta = \frac{Nu_T / Nu_0}{(f_T / f_0)^{1/3}}$$

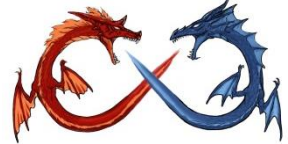
Table 1
Configuration sketches of the self-rotating twisted tapes.

Configuration	Illustration	Name	Reference
	1 – Tube bearing, 2 – spindle, 3 – tube, 4 – twisted tape	Typical twisted tape	[14–21]
	1 – Tube bearing, 2 – spindle, 3 – tube, 4 – twisted tape with oblique teeth	Oblique teeth twisted tape	[22]
	1 – Tube bearing, 2 – straight strip with elliptical teeth and broken edge, 3 – spiral line, 4 – elliptical teeth, 5 – straight strip	Straight strip with elliptical teeth and broken edge	[23]
	–	Miniature hydraulic turbine	[24–28]
	–	Triangular groove twisted tape	[29,30]
	–	Semicircle groove twisted tape	[31]
	–	Regularly spaced twisted tape	[32]
	–	Helically twisted tape	[33]



Stationary vs. self rotating twisted tape

- The stationary twisted tapes can generate the swirl flow in the fluid flow, which can enhance the flow turbulence intensity, and lead to better convection heat transfer compared to the plain tube with an axial flow.
- The self-rotating twisted tapes can strengthen the heat transfer efficiency and achieve on-line automatic anti-scaling and descaling effect.
- The self-rotating twisted tapes change the direction of fluid flow, form a rotational flow, get the bulk flow and fluid at the tube surface mixed and disrupt the development of boundary layer. Compared with the stationary twisted tapes, the self-rotating twisted tapes have the dual effects (swirled and mixing).



典型沸騰(蒸發)管

