製程熱交換節能技術

王啟川, PhD, 交通大學機械工程系特聘教授

Fellow ASME, Fellow ASHRAE

e-Mail ： ccwang@mail.nctu.edu.tw

Tel: 03-5712121 ext. 55105

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課前說明

- 主要教材來源：『熱交換設計』，王啟川，五南出版社，2007。
- 隨時提問
  - “There are no foolish questions and no man becomes a fool until he has stopped asking questions”
    - Charles P. Steinmetz quotes (Prussian Engineer and Inventor, 1865-1923)

課程大綱：

- 各類熱交換器選擇
- 熱傳增強技術與熱交換性能評估
- 狹點技術
- 熱交換器網路
- 案例分析

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。

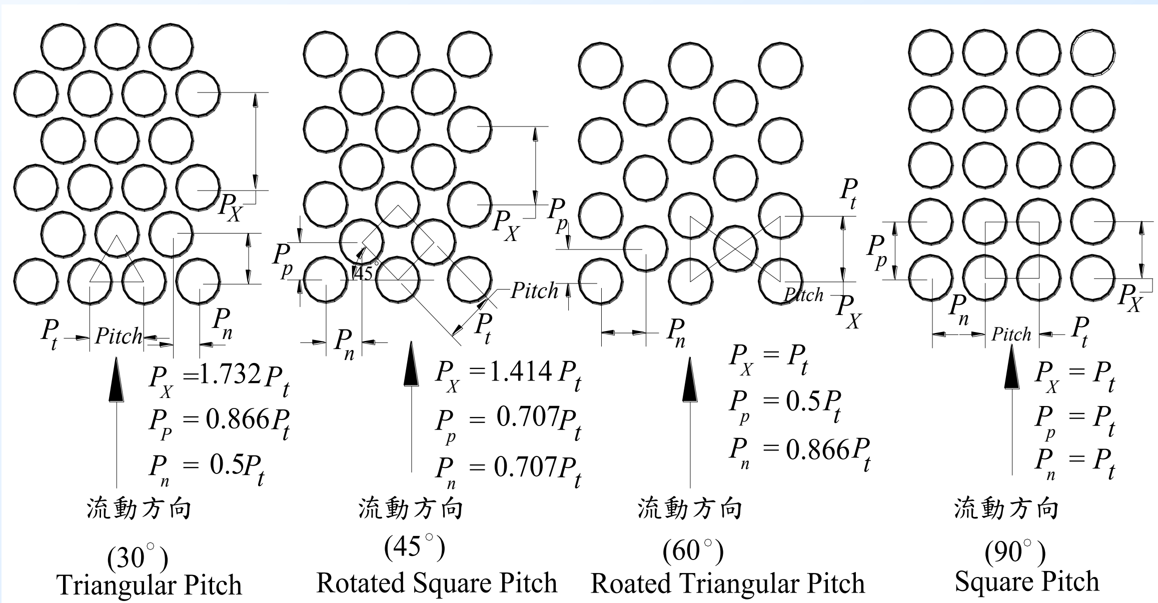
熱交換器選則操作範圍

熱交換器形式	最大壓力 (absolute)	溫度範圍	流體限制	典型熱交換器尺寸範圍	特性
氣冷式Air-cooled	500 bar (製程側)	600 °C (製程側)	僅受限於材料製作	5 ~350 m^2 (裸管，每一單元)，可使用多單元。	通常搭配風扇使用，並大量使用鰭片
可拆卸板式熱交換器	16~25 bar (某些應用設計可達 40 bar)	-25~ 200°C.	通常不適用氣體與兩相流動，使用上襯墊是決定因素	1 ~to 1200 m2	模組化設計，不易清理
固定床式再生式熱交換器	1 bar	~ 600°C.	常用於燃燒廢氣的熱回收兵用來愈熱空氣	–	製作上常使用磚或陶瓷材料
雙套管式熱交換器	300 bar (殼側) 1400 bar (管側).	–100 to 600°C (使用特殊材料時可更高)	僅受限於材料製作	0.25 ~ 200 m^2 per unit – multiple units are often used.	High thermal efficiency, standard modular construction.
熱管熱交換器Heat-pipe	~ 1 bar	通常低於200°C ，但可依需要選取管內工作流體，工作於高溫中	Low pressure gases.	100 ~ 1000 m2.	可設計成逆向像流動，冷熱側均可使用鰭片增加面積
板鰭式 Plate-fin	100 bar (鋁合金) 200 bar (不銹鋼)	–273~150°C (鋁合金) ~ 600°C (不銹鋼)	Low fouling.	熱交器體積通常小於 9 m3	Very small possible. Incorporation of multiple streams. Very large surface area per unit volume. DT
印刷式Printed-circuit	1000 bar	800°C (不銹鋼)	Low fouling	1 to 1000 m2	Very large surface area per unit volume. Stainless steel or higher alloys normal construction material.
轉輪再生式Rotary regenerators	~ 1 bar	980°C.	Low pressured gases.		Inter-stream leakage must be tolerated
殼管式Shell-and-tube	300 bar (殼側). 1400 bar (管側).	–25 ~ 600°C (使用特殊材料可操作於更低或更高的溫度)	僅受限於材料製作	10 to 1000 m2 (per shell – multiple shells can be used).	Very adaptable and can be used for nearly all applications.
螺旋式 Spiral	18 bar	~ 400°C	Subject only to materials of construction. Often used for fouling duties.	~ 200 m2.	High heat transfer efficiency. Cylindrical geometry useful as integral part of distillation tower.

殼管式 (熱傳管)

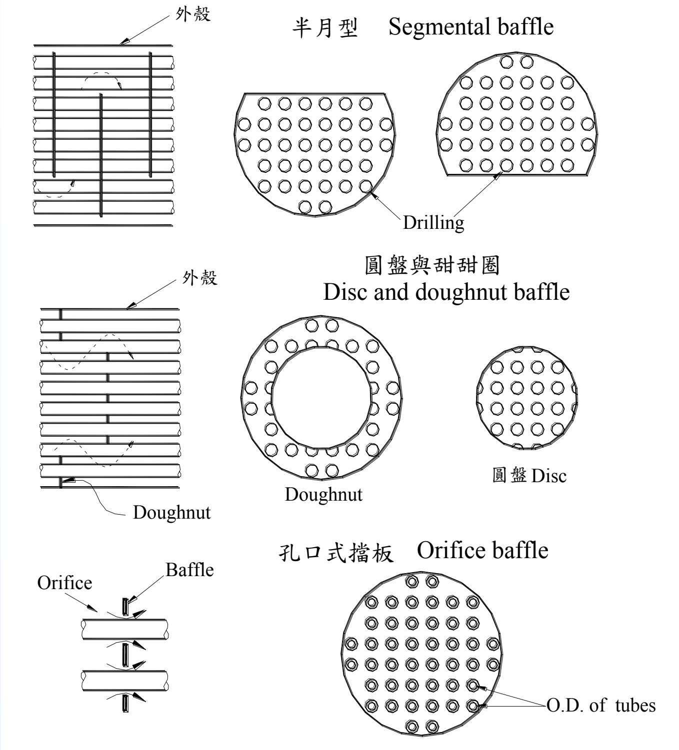
- 熱傳管多使用平滑管，最常見的熱傳管尺寸為15.88, 19.05 與 25.4 mm，通常管徑越大其表面就越粗糙且越容易清理 。
- 使用機械式的清理方式，熱傳管的尺寸應在19.05 mm以上
- 如果需要降低殼側的熱傳阻抗，則可使用低鰭管，常見的低鰭管鰭高在片6.35 mm以下，鰭片間距每米在250~1200片間
- 管內流體的流速應在0.9~2.4 m/s間(3 ~ 8 ft/s, 水之類的工作流體)而殼側的流速約在0.6~1.5 m/s 間(2~5 ft/s)
- 管間距的選取，密集度與保持殼側的可請理度兩者間的平衡，比較密集的比值(P_t/d_o，pitch ratio)熱傳性質較佳但較大的管間距比較容易清理，典型的設計範圍在1.25~2.0間，1.25為設計上的下限值，不過，傳熱管的最小距離($P_t - d_o$)至少要1/8 英吋(3.175 mm) ，如果要經常使用機械式清理，建議至少要有1/4 英吋(6.35 mm)以上

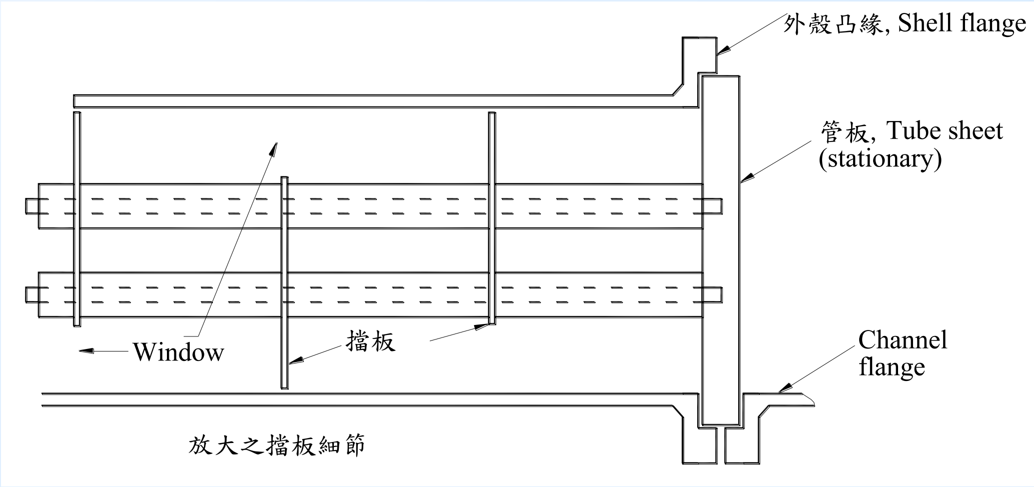
- 傳熱管的排列方式如圖，共有30°、45°、60°及90°四種型式，其中以30°型式最為常見，其熱傳與壓降的比值最好，60°則較30°略差，45°則平平，在紊流流動狀態下且有壓降考量的限制時，90°有最佳的表現，但是在層流流動狀態下時，反而是最差；如果要考慮使用機械式清理，則可能必須使用45º或90º 的型式。



擋板的設計

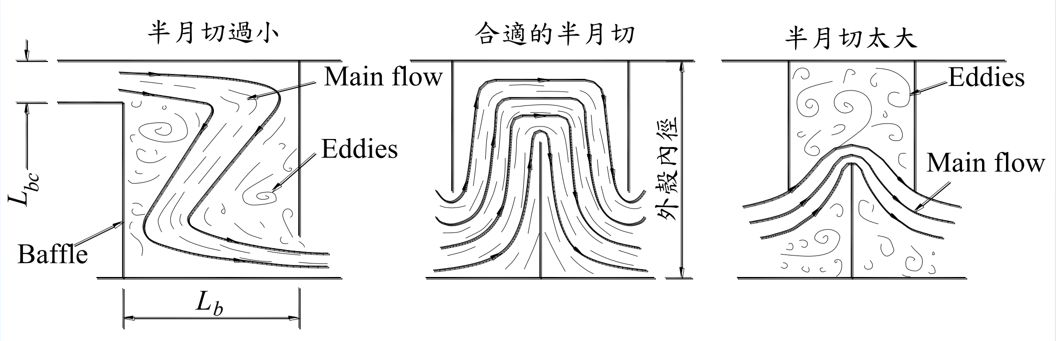
- 常見的管長/外殼直徑的比值多在5~10間。為了固定熱傳管，避免振動，通常需要使用如圖所示的擋板來固定；擋板除了固定的功能外，還兼具導流的功用





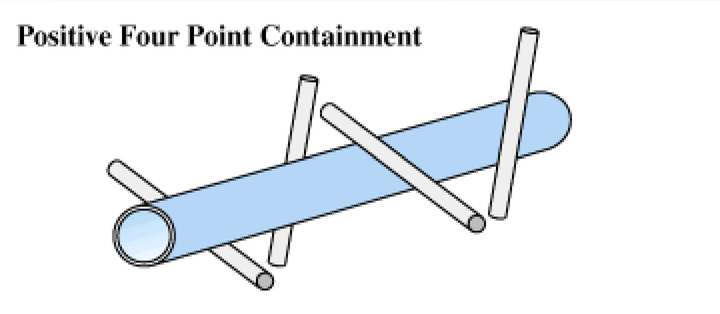
- 擋板間距多在1/5~1 shell dia.
- 最佳擋板間距在2/5~1/2 shell dia.
- 最小可清理的擋板間距為 2 in. or 1/5 shell dia. (兩者較大者)

- 半月切一般在20~49%
- 20~25%最常見
- 如果當板間距甚大，建議半月切為45~50%
- 如果結垢為主要設計的良因素，則半月切應小於25%
- 單相流體建議水平式半月切，兩相流體建議垂直半月切
- 如果振動問題嚴重，可考慮將window部分的熱傳管移除(並降低壓降)



Rod baffle

- 降低流動振動
- 降低流阻，降低½以上之阻力
- 通常熱傳效率較差，但可安排成逆向流提升效率
- Q/ΔP值小於傳統的擋板



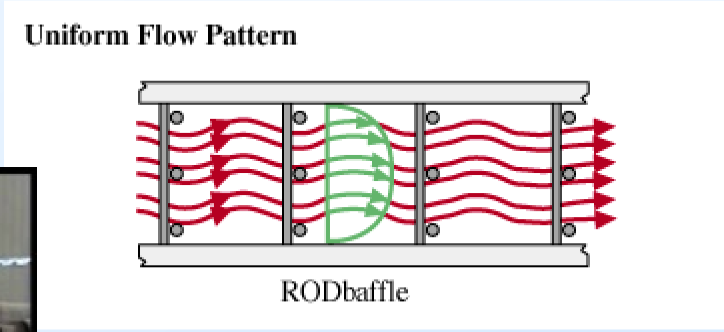
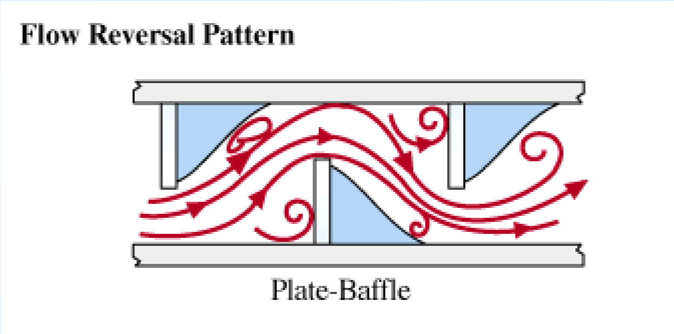
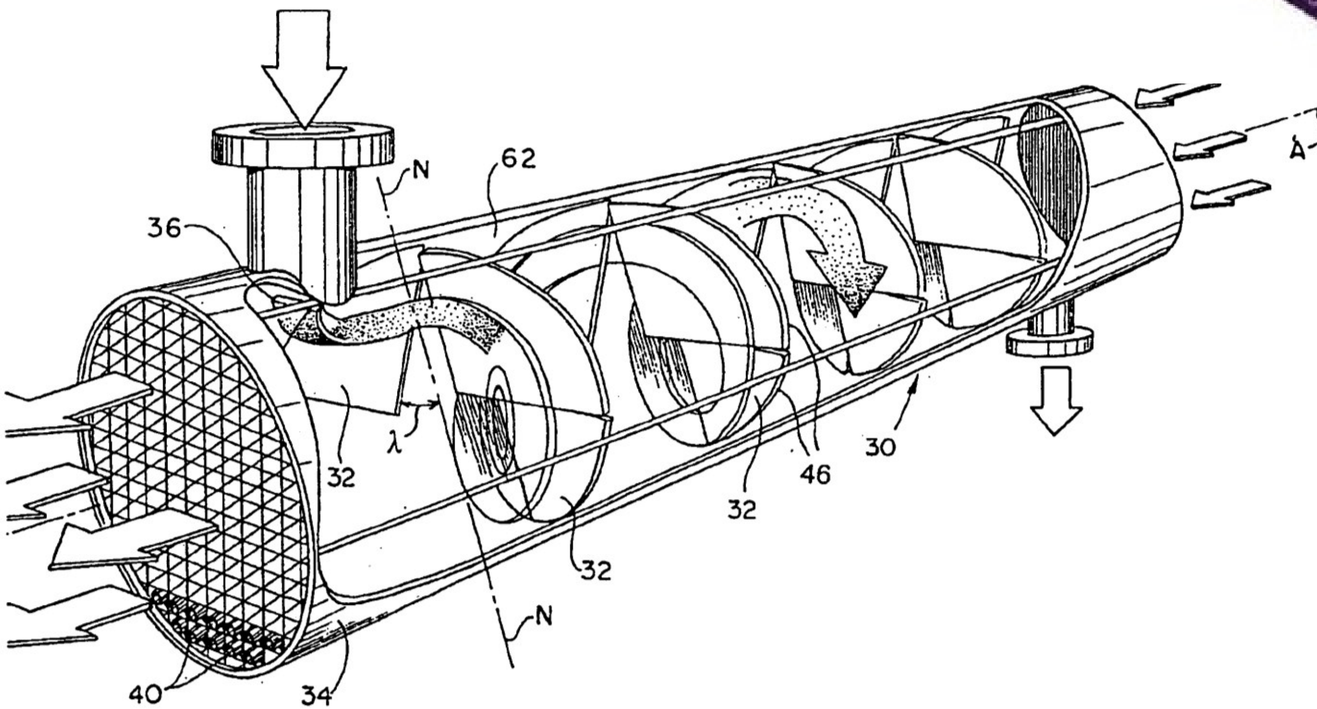


Plate Baffle Example

Rod Baffle Example

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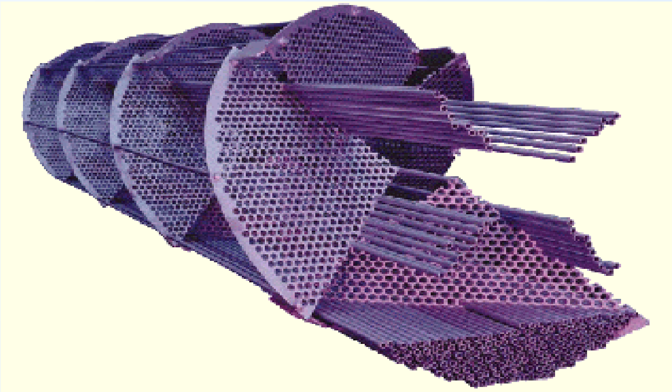
Intensified heat transfer techniques

Shell-side:

- **Helical Baffles®,** which reduce the number of dead spots created by segmented baffle design, where no heat transfer occurs between the tube-side and shell-side fluids



- **EM Baffles®,** which employs expanded metal baffles (tube supports) made of plate material that has been slit and expanded. The open structure allows a longitudinal flow pattern and results in lower hydraulic resistance, so that flow induced tube vibration will not occur.

Impingement baffle

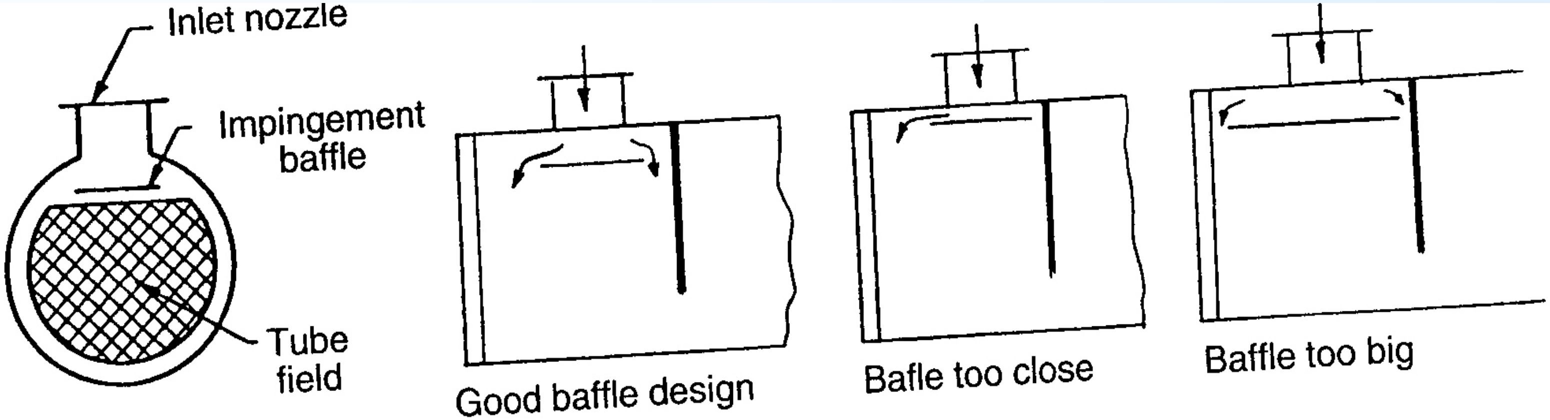


FIGURE 10.2 Impingement baffles at the shell-side inlet nozzle. (From Bell, 1998.)

殼側設計

- E shell, 最普遍造價也最低
- 回數增加，可提升熱傳(壓降需再允許範圍)，回數增加會降低熱傳有效率(effectiveness, why?)
- F shell (longitudinal baffle)，逆向流設計，熱效率較佳，但鮮少使用(why?)
- G & H shell : 降低ΔP的需求；鮮少使用，比較適合有phase change 的應用
- J shell 的壓損僅為E shell 的1/8，適合冷凝器的應用
- X shell 壓損最小(適合氣體的加熱冷卻應用)，但分布較差

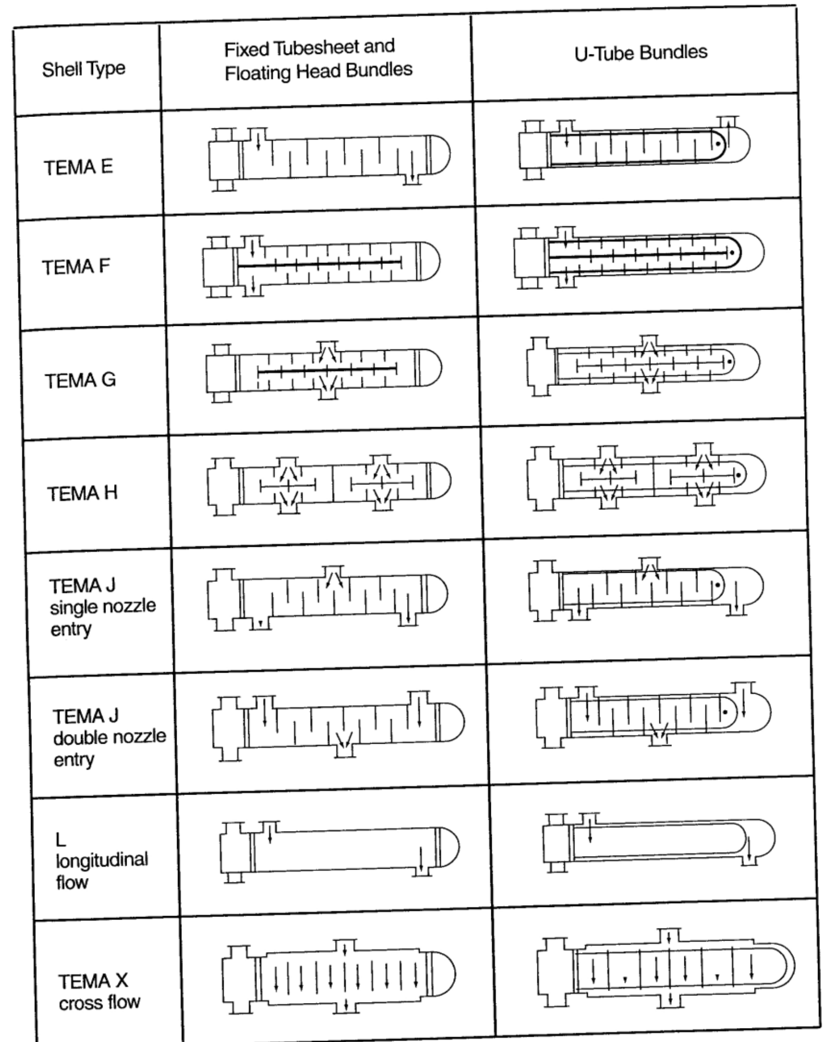


FIGURE 10.3 Shell-side flow arrangement for various shell types (Courtesy of Heat Transfer Research, Inc., College Station, Texas).

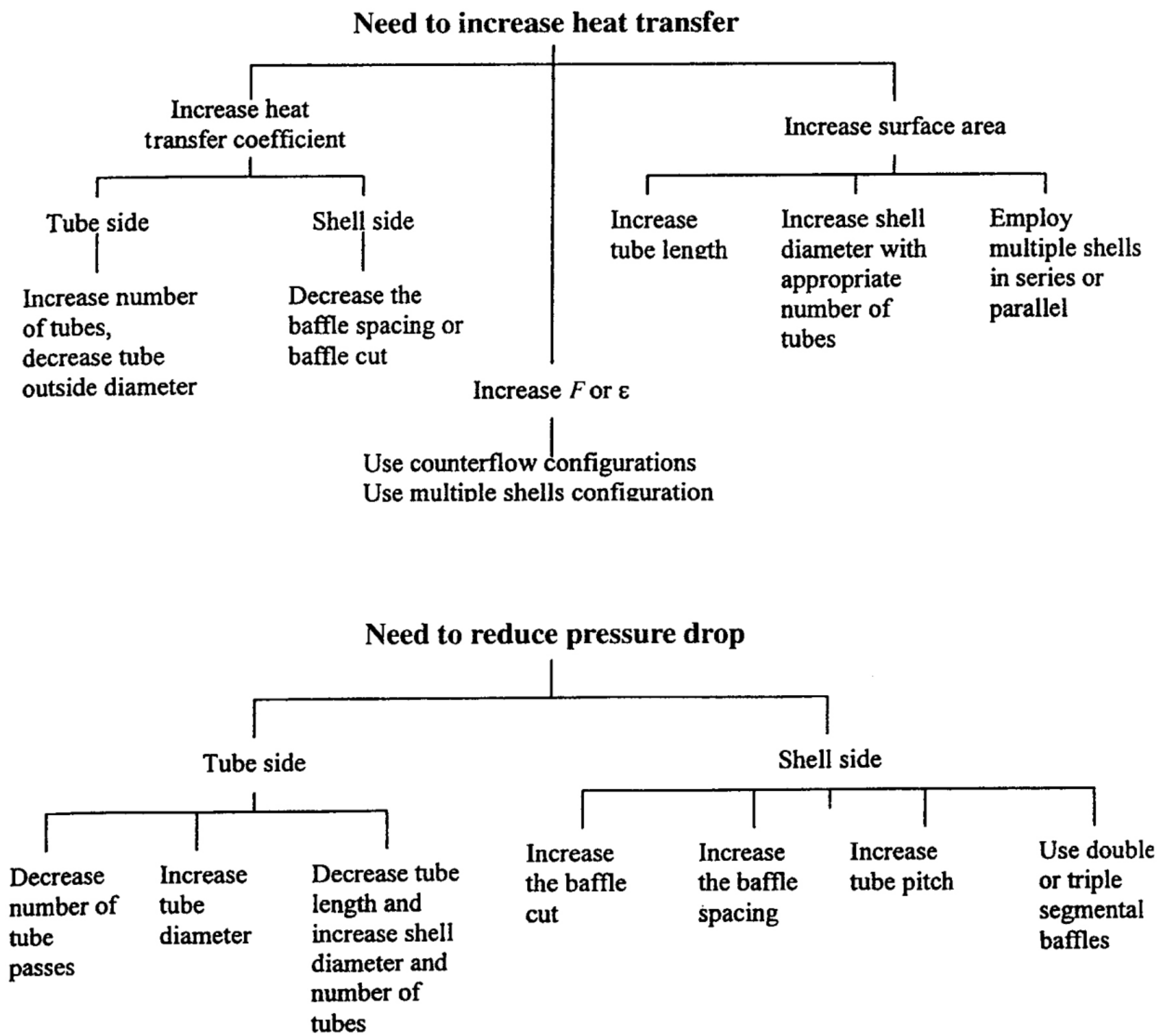


FIGURE 10.10 Influence of various geometrical parameters of a shell-and-tube exchanger on heat transfer and pressure drop.

套管式熱交換器

- Shell < 100 mm
- 構造簡單，不需擋板(降低Leakage & bypass loss)
- 簡單較佳設計Guideline (ESDU 92013):
  - 殼側熱傳係數約小於管內熱傳係數25%

板式熱交換器

- Highly flexible (可拆卸式), sealing may be a problem.
- Maximum temperature/pressure constraints.
- 單相流體應用較適合
- 較佳的熱傳性能(尤其考慮到空間上的限制)

氣冷式熱交換器

- 絕大部分為Cross flow，effectiveness 較低
- 對環境影響遠比Cooling tower 小
- Induced draft 可提供較佳的流量分佈，但fan操作於較高溫度，比較耗功
- A-frame 常用於Steam condensation
- 風扇噪音為使用上的問題
- 最典型的管徑尺寸為1 in. with 2 in. wound fins. Fin pitch ~ 11 fins/in.
- 典型操作面速 2~ 5 m/s.
- Row number: 3~8.

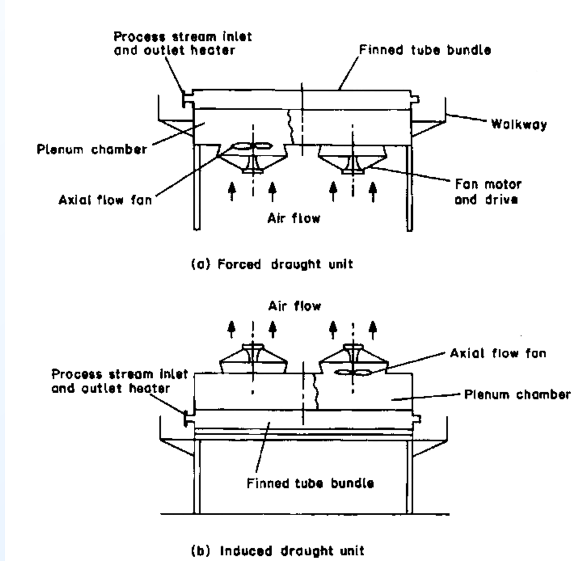
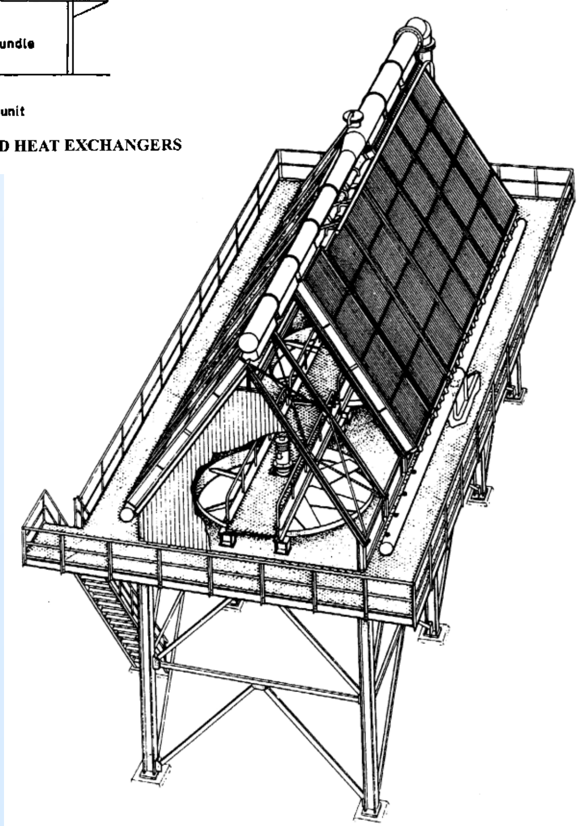


FIGURE 13 AIR-COOLED HEAT EXCHANGERS



熱交換器的選擇

- F_T Method
- Effectiveness Method

Get $Q/\Delta T_m$

TABLE 9.2 *U* & *C* VALUES FOR SHELL-AND-TUBE HEAT EXCHANGERS (courtesy of Johnson Hunt Ltd)

$\dot{Q}/\Delta T$ (W/K)	Cold Side Fluid	Parameter	Hot Side Fluid								
			Low Pressure Gas (<1 bar)	Medium Pressure Gas (20 bar)	High Pressure Gas (150 bar)	Process Water	Low Viscosity Organic Liquid	High Viscosity Liquid	Condensing Steam	Condensing Hydrocarbon	Condensing Hydrocarbon with Inert Gas
1 000	Low Pressure Gas (<1 bar)	U (W/m^2 K) C (£/(W/K))	55 5.70	93 5.02	120 5.51	102 4.93	99 4.96	63 5.50	107 4.87	100 4.95	86 5.11
	Medium Pressure Gas (20 bar)	U (W/m^2 K) C (£/(W/K))	93 5.02	300 4.18	350 4.81	429 4.03	375 4.09	120 4.76	530 3.95	388 4.07	240 4.28
	High Pressure Gas (150 bar)	U (W/m^2 K) C (£/(W/K))	120 5.51	350 4.81	400 6.25	600 4.56	450 4.38	200 5.50	600 4.56	400 4.82	300 4.81
	Treated Cooling Water	U (W/m^2 K) C (£/(W/K))	105 4.89	484 3.98	600 4.56	938 3.77	714 3.85	142 4.59	1607 3.61	764 3.83	345 4.12
	Low Viscosity Organic Liquid	U (W/m^2 K) C (£/(W/K))	99 4.96	375 4.09	450 4.38	600 3.91	500 3.97	130 4.67	818 3.81	524 3.95	286 4.20
	High Viscosity Liquid	U (W/m^2 K) C (£/(W/K))	68 5.39	138 4.61	200 5.50	161 4.46	153 4.51	82 5.16	173 4.42	155 4.50	214 4.33
	Boiling Water	U (W/m^2 K) C (£/(W/K))	105 4.89	467 3.99	550 4.91	875 3.79	677 3.87	140 4.60	1432 3.64	722 3.85	336 4.13
	Boiling Organic Liquid	U (W/m^2 K) C (£/(W/K))	99 4.96	375 4.09	450 4.38	600 3.91	500 3.97	130 4.67	818 3.81	524 3.95	286 4.20
5 000	Low Pressure Gas (<1 bar)	U (W/m^2 K) C (£/(W/K))	55 2.11	93 1.63	120 2.26	102 1.58	99 1.59	63 1.95	107 1.55	100 1.59	86 1.68
	Medium Pressure Gas (20 bar)	U (W/m^2 K) C (£/(W/K))	93 1.63	300 1.11	350 1.89	429 1.02	375 1.05	120 1.49	530 0.98	388 1.05	240 1.18
	High Pressure Gas (150 bar)	U (W/m^2 K) C (£/(W/K))	120 2.26	350 1.89	400 2.25	600 1.10	450 1.46	200 1.93	600 1.10	400 1.45	300 1.45
	Treated Cooling Water	U (W/m^2 K) C (£/(W/K))	105 1.56	484 1.00	600 1.10	938 0.88	720 0.91	142 1.41	1607 0.83	764 0.90	345 1.07
	Low Viscosity Organic Liquid	U (W/m^2 K) C (£/(W/K))	99 1.59	375 1.05	450 1.46	600 0.95	500 0.99	130 1.46	818 0.89	524 0.98	286 1.13
	High Viscosity Liquid	U (W/m^2 K) C (£/(W/K))	68 1.86	138 1.43	200 1.93	161 1.36	153 1.38	82 1.71	173 1.32	155 1.37	124 1.48
	Boiling Water	U (W/m^2 K) C (£/(W/K))	105 1.56	467 1.00	550 1.20	875 0.88	677 0.93	140 1.42	1432 0.84	722 0.91	336 1.08
	Boiling Organic Liquid	U (W/m^2 K) C (£/(W/K))	99 1.59	375 1.05	450 1.46	600 0.95	500 0.99	130 1.46	818 0.89	524 0.98	286 1.13

TABLE 9.3 *U* & *C* VALUES FOR DOUBLE-PIPE HEAT EXCHANGERS(Courtesy Brown Fintube Ltd)

$\dot{Q}/\Delta T$ (W/K)	Cold Side Fluid	Parameter	Hot Side Fluid								
			Low Pressure Gas (< 1 bar)	Medium Pressure Gas (20 bar)	High Pressure Gas (150 bar)	Process Water	Low Viscosity Hydrocarbon Liquid	High Viscosity Hydrocarbon Liquid	Condensing Steam	Condensing Hydrocarbon	Condensing Hydrocarbon with Inert Gas
1 000	Low Pressure Gas (< 1 bar)	*U* (W/m² K) *C* (£/(W/K))	55 4.8	95 3.8	125 2.9	105 3.8	100 3.8	65 4.7	110 2.8	100 3.8	85 3.9
	Medium Pressure Gas (20 bar)	*U* (W/m² K) *C* (£/(W/K))	95 3.4	300 2.5	350 2.9	430 2.5	375 5	120 2.6	530 2.5	390 2.5	240 2.5
	High Pressure Gas (150 bar)	*U* (W/m² K) *C* (£/(W/K))	120 2.9	350 2.9	400 2.9	500 2.9	400 2.9	150 2.9	600 2.9	420 2.9	350 2.9
	Treated Cooling Water	*U* (W/m² K) *C* (£/(W/K))	105 2.8	484 2.5	500 2.9	940 2.5	715 2.5	145 2.5	1610 2.5	765 3.9	345 2.5
	Low Viscosity Hydrocarbon Liquid	*U* (W/m² K) *C* (£/(W/K))	100 2.7	375 2.5	425 2.9	600 2.5	500 2.5	130 2.5	820 2.5	525 2.5	290 2.5
	High Viscosity Hydrocarbon Liquid	*U* (W/m² K) *C* (£/(W/K))	70 4.7	140 2.5	175 2.9	160 2.5	155 2.5	85 3.9	175 2.5	155 2.5	125 2.5
	Boiling Water	*U* (W/m² K) *C* (£/(W/K))	105 3.9	470 2.5	550 2.9	875 2.5	670 2.5	140 2.5	1435 2.5	725 2.5	340 2.5
	Boiling Organic Liquid	*U* (W/m² K) *C* (£/(W/K))	100 4	375 2.5	430 2.9	600 2.5	500 2.5	130 2.5	820 2.5	525 2.5	285 2.5
5 000	Low Pressure Gas (< 1 bar)	*U* (W/m² K) *C* (£/(W/K))	55 2.16	95 1.26	125 1.1	105 1.23	100 1.22	65 1.84	110 1.23	100 1.22	85 3.5
	Medium Pressure Gas (20 bar)	*U* (W/m² K) *C* (£/(W/K))	95 1.5	300 0.86	350 1	430 0.76	375 0.8	120 1.16	530 0.6	390 0.8	240 0.94
	High Pressure Gas (150 bar)	*U* (W/m² K) *C* (£/(W/K))	120 2.05	350 1.1	400 1.1	500 1	400 1.1	150 1.3	600 1	420 1.1	350 1.2
	Treated Cooling Water	*U* (W/m² K) *C* (£/(W/K))	105 1.4	484 0.75	500 1	940 0.5	715 0.72	145 1	1610 0.4	765 0.72	345 0.9
	Low Viscosity Hydrocarbon Liquid	*U* (W/m² K) *C* (£/(W/K))	100 1.45	375 0.8	425 1.05	600 0.8	500 0.9	130 1.1	820 0.5	525 0.9	290 1.2
	High Viscosity Hydrocarbon Liquid	*U* (W/m² K) *C* (£/(W/K))	70 2.4	140 1.66	175 1.2	160 0.95	155 1	85 2.7	175 0.82	155 1.6	125 1.95
	Boiling Water	*U* (W/m² K) *C* (£/(W/K))	105 1.4	470 0.9	550 1	875 0.5	670 0.6	140 1.1	1435 0.4	725 0.73	340 0.9
	Boiling Organic Liquid	*U* (W/m² K) *C* (£/(W/K))	100 1.45	375 0.8	430 1.05	600 0.8	500 0.9	130 1.28	820 0.7	525 1	285 1.1

TABLE 9.4 *U* AND *C* VALUES FOR PLATE HEAT EXCHANGERS (courtesy APV Baker Ltd)

$\dot{Q}/\Delta T$ (W/K)	Cold Side Liquid	Parameter	Hot Side Liquid: Water	Hot Side Liquid: Aqueous Inorganic Liquid	Hot Side Liquid: Organic Liquid
5 000	Water	U (W/m^2 K) C (£/(W/K))	5500 0.160	3500 0.130 to 0.170	1750 0.140 to 0.190
	Aqueous Inorganic Liquid	U (W/m^2 K) C (£/(W/K))	3500 0.130 to 0.170	1750 0.190	1250 0.160 to 0.210
	Organic Liquid	U (W/m^2 K) C (£/(W/K))	1750 0.140 to 0.190	1250 0.160 to 0.210	1150 0.220
10 000	Water	U (W/m^2 K) C (£/(W/K))	5500 0.1	3500 0.1	1850 0.11
	Aqueous Inorganic Liquid	U (W/m^2 K) C (£/(W/K))	3500 0.1	2200 0.1	1400 0.13
	Organic Liquid	U (W/m^2 K) C (£/(W/K))	1850 0.11	1400 0.13	1150 0.15
50 000	Water	U (W/m^2 K) C (£/(W/K))	5500 0.025	3500 0.025 to 0.035	1850 0.035
	Aqueous Inorganic Liquid	U (W/m^2 K) C (£/(W/K))	3500 0.025 to 0.035	3500 0.035	1400 0.055
	Organic Liquid	U (W/m^2 K) C (£/(W/K))	1850 0.035	1400 0.055	1150 0.065
100 000	Water	U (W/m^2 K) C (£/(W/K))	5500 0.025	3500 0.035	1850 0.045
	Aqueous Inorganic Liquid	U (W/m^2 K) C (£/(W/K))	3500 0.035	3500 0.035	1400 0.055
	Organic Liquid	U (W/m^2 K) C (£/(W/K))	1850 0.045	1400 0.055	1150 0.065

TABLE 9.5 U & C VALUES FOR PRINTED-CIRCUIT HEAT EXCHANGERS (courtesy Heatric Ltd)

$\dot{Q}/\Delta T$ (W/K)	Cold Side Fluid	Parameter	Hot Side Fluid								
			Low Pressure Gas (< 1 bar)	Medium Pressure Gas (20 bar)	High Pressure Gas (150 bar)	30 per cent Triethylene Glycol/Water	Condensing Hydrocarbon	Low Viscosity Liquid (< 1 cP)	Medium Viscosity Liquid (1 to 5 cP)	High Viscosity Fluid (20 cP)	Condensing Steam
1 000	Low Pressure Gas (< 1 bar)	U (W/m^2 K)	198	331	357	369	338	356	310	236	378
		C (£/(W/K))	12	12	12	12	12	12	12	12	12
	Medium Pressure Gas (20 bar)	U (W/m^2 K)	331	1029	1324	1506	1090	1304	845	458	1666
		C (£/(W/K))	12	12	12	12	12	12	12	12	12
	High Pressure Gas (150 bar)	U (W/m^2 K)	357	1324	1856	2234	1427	1816	1034	508	2605
		C (£/(W/K))	12	12	12	12	12	12	12	12	12
	30 per cent Triethylene Glycol/ Water	U (W/m^2 K)	364	1506	2234	2804	1641	2176	1141	533	3414
		C (£/(W/K))	12	12	12	12	12	12	12	12	12
	Boiling Organic Liquid	U (W/m^2 K)	321	932	1168	1307	982	1152	778	438	1426
		C (£/(W/K))	12	12	12	12	12	12	12	12	12
	Low Viscosity Liquid (< 1 cP)	U (W/m^2 K)	356	1304	1816	2176	1404	1778	1021	505	2526
		C (£/(W/K))	12	12	12	12	12	12	12	12	12
	Medium Viscosity Liquid (1 to 5 cP)	U (W/m^2 K)	310	845	1034	1141	886	1021	716	417	1231
		C (£/(W/K))	12	12	12	12	12	12	12	12	12
	High Viscosity Liquid (20 cP)	U (W/m^2 K)	236	458	508	533	470	505	417	294	552
		C (£/(W/K))	12	12	12	12	12	12	12	12	12
	Treated Cooling Water	U (W/m^2 K)	376	1621	2496	3230	1778	2424	1206	547	4068
		C (£/(W/K))	12	12	12	12	12	12	12	12	12
5 000	Low Pressure Gas (< 1 bar)	U (W/m^2 K)	198	331	357	369	338	356	310	236	378
		C (£/(W/K))	5.2	3.6	4.0	3.6	3.6	3.6	3.6	4.8	3.6
	Medium Pressure Gas (20 bar)	U (W/m^2 K)	331	1029	1324	1506	1090	1304	845	458	1666
		C (£/(W/K))	3.6	2.4	2.8	2.4	2.4	2.4	2.4	2.5	2.4
	High Pressure Gas (150 bar)	U (W/m^2 K)	357	1324	1856	2234	1427	1816	1034	508	2605
		C (£/(W/K))	4.0	2.8	3.2	2.8	2.8	2.8	2.8	2.8	2.8
	30 per cent Triethylene Glycol/Water	U (W/m^2 K)	369	1506	2234	2804	1641	2176	1141	533	3414
		C (£/(W/K))	3.6	2.4	2.8	2.4	2.4	2.4	2.4	2.4	2.4
	Boiling Organic Liquid	U (W/m^2 K)	321	932	1168	1307	982	1152	778	438	1426
		C (£/(W/K))	3.6	2.4	2.8	2.4	2.4	2.4	2.4	2.7	2.4
	Low Viscosity Liquid (< 1 cP)	U (W/m^2 K)	356	1304	1816	2176	1404	1778	1021	505	2526
		C (£/(W/K))	3.6	2.4	2.8	2.4	2.4	2.4	2.4	2.4	2.4
	Medium Viscosity Liquid (1 to 5 cP)	U (W/m^2 K)	310	845	1034	1141	886	1021	716	417	1231
		C (£/(W/K))	3.6	2.4	2.8	2.4	2.4	2.4	2.4	2.7	2.4
	High Viscosity Liquid (20 cP)	U (W/m^2 K)	236	458	508	533	470	505	417	294	552
		C (£/(W/K))	4.8	2.6	2.8	2.4	2.6	2.4	2.7	2.8	2.4
	Treated Cooling Water	U (W/m^2 K)	376	1621	2496	3230	1778	2424	1206	547	4068
		C (£/(W/K))	3.6	2.4	2.8	2.4	2.4	2.4	2.4	2.4	2.4

TABLE 9.6 *U* & *C* VALUES FOR PLATE-FIN HEAT EXCHANGERS (courtesy Marston Palmer Ltd)

(Key: NA - Not Applicable; NUS - Not Common)

$\dot{Q}/\Delta T$ (W/K)	Cold Side Fluid	Parameter	Hot Side Fluid								
			Low Pressure Gas (< 1 bar)	Medium Pressure Gas (20 bar)	High Pressure Gas (150 bar)	Process Water	Low Viscosity Hydrocarbon Liquid	High Viscosity Hydrocarbon Liquid	Condensing Steam	Condensing Hydrocarbon	Condensing Hydrocarbon with Inert Gas
5 000	Low Pressure Gas (< 1 bar)	U (W/m^2 K) C (£/(W/K))	163 3.10	217 3.10	NA	NA	264 3.10	NUS	NUS	270 3.10	NUS
	Medium Pressure Gas (20 bar)	U (W/m^2 K) C (£/(W/K))	217 3.10	325 3.10	NA	NA	377 3.10	NUS	NUS	402 3.10	NUS
	High Pressure Gas (150 bar)	U (W/m^2 K) C (£/(W/K))	NA	NA	NA	NA	NA	NA	NA	NA	NA
	Treated Cooling Water	U (W/m^2 K) C (£/(W/K))	315 3.10	491 3.10	NA	NA	NUS	NUS	NUS	NUS	NUS
	Low Viscosity Hydrocarbon Liquid	U (W/m^2 K) C (£/(W/K))	NUS	NUS	NA	NA	NUS	NUS	NUS	NUS	NUS
	High Viscosity Hydrocarbon Liquid	U (W/m^2 K) C (£/(W/K))	NUS	NUS	NA	NUS	NUS	NUS	NUS	NUS	NUS
	Boiling water	U (W/m^2 K) C (£/(W/K))	NA	NA	NA	NA	NA	NA	NA	NA	NA
	Boiling Hydrocarbon	U (W/m^2 K) C (£/(W/K))	270 3.10	402 3.10	NA	NA	453 3.10	NUS	NA	530 3.10	NUS
10 000	Low Pressure Gas (< 1 bar)	U (W/m^2 K) C (£/(W/K))	163 1.57	217 1.55	NA	NA	264 1.55	NUS	NUS	270 1.55	NUS
	Medium Pressure Gas (20 bar)	U (W/m^2 K) C (£/(W/K))	217 1.55	325 1.55	NA	NA	377 1.55	NUS	NUS	402 1.55	NUS
	High Pressure Gas (150 bar)	U (W/m^2 K) C (£/(W/K))	NA	NA	NA	NA	NA	NA	NA	NA	NA
	Treated Cooling Water	U (W/m^2 K) C (£/(W/K))	315 1.55	491 1.55	NA	NA	NUS	NUS	NUS	NUS	NUS
	Low Viscosity Hydrocarbon Liquid	U (W/m^2 K) C (£/(W/K))	NUS	NUS	NA	NA	NUS	NUS	NUS	NUS	NUS
	High Viscosity Hydrocarbon Liquid	U (W/m^2 K) C (£/(W/K))	NUS	NUS	NA	NUS	NUS	NUS	NUS	NUS	NUS
	Boiling water	U (W/m^2 K) C (£/(W/K))	NA	NA	NA	NA	NA	NA	NA	NA	NA
	Boiling Hydrocarbon	U (W/m^2 K) C (£/(W/K))	270 1.55	402 1.55	NA	NA	453 1.55	NUS	NA	530 1.55	NUS

TABLE 9.8 *U* & *C* VALUES FOR WELDED-PLATE HEAT EXCHANGERS (courtesy Johnson Hunt Ltd)

$\dot{Q}/\Delta T$ (W/K)	Cold Side Fluid	Parameter	Hot-side Fluid: Low Pressure Gas (< 1 bar)	Medium Pressure Gas (20 bar)	High Pressure Gas (150 bar)	Process Water	Low Viscosity Organic Liquid	High Viscosity Liquid	Condensing Steam	Condensing Hydrocarbon	Condensing Hydrocarbon with Inert Gas
1 000	Low Pressure Gas (< 1 bar)	U (W/m^2 K)	78	106	NA	220	153	122	243	225	190
		C (£/(W/K))	8.1	8.0		4.9	5.7	5.9	4.7	4.9	5.8
	Medium Pressure Gas (20 bar)	U (W/m^2 K)	103	256	NA	381	308	202	1021	708	600
		C (£/(W/K))	8.1	7		4.9	4.9	5.7	4.0	3.7	4.36
	High Pressure Gas (150 bar)	U (W/m^2 K)	NA	NA	NA	NA	NA	NA	NA	NA	NA
		C (£/(W/K))									
	Treated Cooling Water	U (W/m^2 K)	158	349	NA	1328	380	228	1750	511	400
		C (£/(W/K))	5.6	4.9		3.7	4.1	4.9	3.5	3.9	4.98
	Low Viscosity Organic Liquid	U (W/m^2 K)	143	297	NA	343	534	283	1085	701	600
		C (£/(W/K))	5.1	5.6		4.1	3.9	4.4	3.8	3.7	4.36
	High Viscosity Liquid	U (W/m^2 K)	103	188	NA	215	271	167	363	311	220
		C (£/(W/K))	6.1	5.9		4.8	4.6	5.1	3.9	4.1	4.91
	Boiling Water	U (W/m^2 K)	170	350	NA	1400	500	250	1800	800	500
		C (£/(W/K))	5.20	4.90		3.60	3.90	4.32	3.50	3.28	4.36
	Boiling Organic Liquid	U (W/m^2 K)	150	300	NA	1000	400	200	1200	600	400
		C (£/(W/K))	5.50	5.6		3.80	4.98	4.46	3.70	4.36	4.98
5 000	Low Pressure Gas (< 1 bar)	U (W/m^2 K)	77	114	NA	220	168	119	243	226	190
		C (£/(W/K))	4.02	3.4		2.8	2.74	2.8	4.4	1.92	2.28
	Medium Pressure Gas (20 bar)	U (W/m^2 K)	108	616	NA	1223	1110	357	1400	787	440
		C (£/(W/K))	3.58	1.5		1.18	1.22	1.98	1.02	1.3	1.98
	High Pressure Gas (150 bar)	U (W/m^2 K)	NA	NA	NA	NA	NA	NA	NA	NA	NA
		C (£/(W/K))									
	Treated Cooling Water	U (W/m^2 K)	215	1187	NA	4252	1551	454	3880	1272	600
		C (£/(W/K))	2.54	1.22		0.74	1.0	1.52	0.7	0.9	1.22
	Low Viscosity Organic Liquid	U (W/m^2 K)	140	975	NA	1650	1692	522	2676	1281	600
		C (£/(W/K))	2.94	1.22		0.82	0.78	1.22	0.78	0.92	1.22
	High Viscosity Liquid	U (W/m^2 K)	131	357	NA	411	400	242	621	434	300
		C (£/(W/K))	2.8	0.9		1.3	1.8	2.04	1.26	1.36	1.66
	Boiling Water	U (W/m^2 K)	195	1000	NA	4500	2000	600	4000	1200	550
		C (£/(W/K))	2.8	1.28		0.78	0.58	1.04	0.75	1.28	1.35
	Boiling Organic Liquid	U (W/m^2 K)	155	800	NA	2000	1500	500	2500	900	450
		C (£/(W/K))	2.65	1.48		0.79	0.80	1.25	0.78	1.38	1.52

Selection of Double-pipe Heat Exchanger

A heat exchanger is required for the cooling of a high viscosity oil with a specific heat of 2 kJ/kg K flowing at a rate of 0.5 kg/s and at a pressure of 30 bar (3.0 MPa). The oil enters the exchanger at 70°C and leaves at 30°C. It is cooled by water flowing at 2 kg/s and entering at 20°C. The specific heat capacity of water is 4.2 kJ/kg K. The oil stream is subject to moderate fouling. Evaluate the alternative heat exchangers for this duty and select between the feasible types on the basis of cost estimated from Tables 9.2 to 9.8.

Solution. For this case, the pressure is too high for a gasketed-plate exchanger, and the plate-fin, welded-plate and printed-circuit heat exchangers are eliminated on the grounds of fouling. The options considered are the shell-and-tube and double-pipe heat exchangers. The heat load, $\dot{Q}$, is given by:

$$\dot{Q} = \dot{M}_h c_{p,h}(T_{h,in} - T_{h,out}) = 0.5 \times 2.0 \times 10^3 \times (70 - 30) = 40\ 000\ \mathrm{W}.$$

The water outlet temperature is given by:

$$T_{c,out} = T_{c,in} + \frac{\dot{Q}}{\dot{M}_c c_{p,c}} = 20 + \frac{40000}{2 \times 4.2 \times 10^3} = 24.8\,°\mathrm{C}.$$

Assuming that the shell-and-tube exchanger is multi-pass (the most widely used) then the value of $\dot{Q}/\Delta T_m$ is calculated using the procedure given in Section 6. The values of E and R are calculated as follows (Equation (6.7)):

$$E = \frac{(T_{in} - T_{out})_{larger}}{(T_{h,in} - T_{c,in})}.$$

The oil has the larger value of $(T_{in} - T_{out})$ and thus:

$$E = \frac{(70 - 30)}{(70 - 20)} = 0.8.$$

The values of $\dot{M}c_p$ are calculated for the oil and for the water as follows:

$$(\dot{M}c_p)_{oil} = 0.5 \times 2.0 = 1.0\ \mathrm{kW/K}$$

$$(\dot{M}c_p)_{water} = 2.0 \times 4.2 = 8.4\ \mathrm{kW/K}.$$

The value of R is then estimated from Equation (6.9):

$$R = \frac{(\dot{M}c_p)_{smaller}}{(\dot{M}c_p)_{larger}} = \frac{1.0}{8.4} = 0.1190 .$$

Reading off from Figure 21, the value of N_{TU} corresponding to these values of E and R is 1.86. Thus, from Equation (6.10),

$$\begin{aligned}\dot{Q}/\Delta T_m &= (\dot{M}c_p)_{smaller} N_{TU} \\ &= 1000 \times 1.86 = 1860 \text{ W/K} .\end{aligned}$$

The flows in double-pipe exchangers are normally countercurrent. Thus $\Delta T_m = \Delta T_{lm}$. The logarithmic mean temperature difference is first calculated from Equation (6.1) as follows:

$$\begin{aligned}\Delta T_{lm} &= \frac{[(T_{h,\,in} - T_{c,\,out}) - (T_{h,\,out} - T_{c,\,in})]}{\log_e [(T_{h,\,in} - T_{c,\,out})/(T_{h,\,out} - T_{c,\,in})]} \\ &= \frac{[(70 - 24.8) - (30 - 20)]}{\log_e [(70 - 24.8)/(30 - 20)]} = 23.33 \text{ K}.\end{aligned}$$

Since $\Delta T_m = \Delta T_{lm}$ for the double-pipe exchanger, it follows that

$$\dot{Q}/\Delta T_m = \dot{Q}/\Delta T_{lm} = \frac{40000}{23.33} = 1715 \text{ W/K}.$$

To estimate costs of these alternative designs, the C-value tables, Tables 9.2 and 9.3, are used. Suppose there is a value C_1 at $(\dot{Q}/\Delta T_m)_1$ and a value C_2 at $(\dot{Q}/\Delta T_m)_2$; the C value for the calculated $(\dot{Q}/\Delta T_m)$ is given by logarithmic interpolation and is as follows:

$$C = \exp\left\{\log_e C_1 + \frac{\log_e (C_1/C_2)\log_e [(\dot{Q}/\Delta T_m)/(\dot{Q}/\Delta T_m)_1]}{\log_e (\dot{Q}/\Delta T_m)_1/(\dot{Q}/\Delta T_m)_2}\right\}.$$

For the shell-and-tube heat exchanger from Table 9.2 (for "Treated Cooling Water" and "High Viscosity Fluid" as the cold and hot side fluids, respectively) $C_1 = 4.59$ at $(\dot{Q}/\Delta T_m)_1$ and $C_2 = 1.41$ at $(\dot{Q}/\Delta T_m)_2 = 5000$. Thus for $\dot{Q}/\Delta T_m = 1800$ the value of C is given by

$$\begin{aligned} C &= \exp\left[\log_e 4.59 + \frac{\log_e (4.59/1.41)\log_e (1860/1000)}{\log_e (1000/5000)}\right] \\ &= 2.912 \text{ £/(W/K)}. \end{aligned}$$

Similarly, for the double-pipe exchanger, the values $C_1 = 2.5$ at $(\dot{Q}/\Delta T_m)_1 = 1000$ and $C_2 = 1.0$ at $(\dot{Q}/\Delta T_m)_2 = 5000$ are read from Table 9.3. Here, the C value for $\dot{Q}/\Delta T_m = 1715$ is calculated by logarithmic interpolation as 1.839 £/(W/K). Thus, the costs of the two exchangers at 1992 prices are as follows:

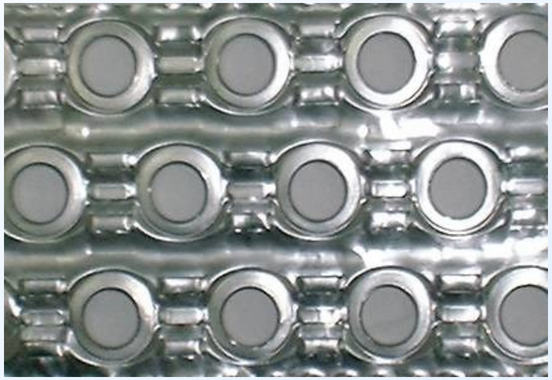
$$\text{Shell-and-tube: Cost} = C \times \dot{Q}/\Delta T_m = 2.912 \times 1860 = \text{c.£5400}$$

$$\text{Double-pipe: Cost} = C \times \dot{Q}/\Delta T_m = 1.839 \times 1715 = \text{c.£3200}.$$

Thus, the double-pipe exchanger is likely to be the best option for this small, high pressure duty where modular construction procedures are a positive cost advantage.

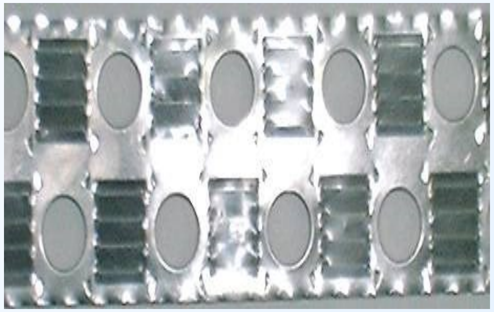
Extended surfaces

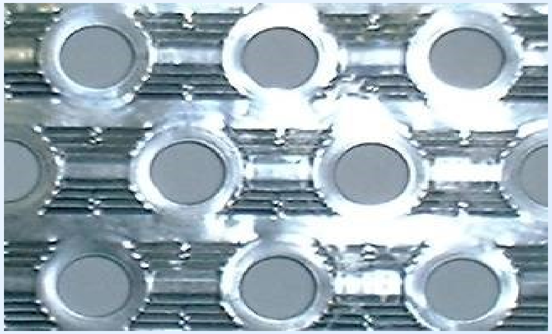
波浪鰭片
(wavy)



複合百葉窗鰭片－convex louver

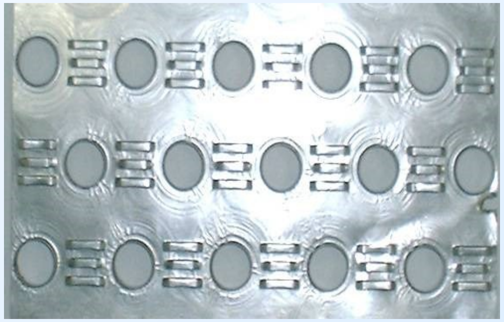
百葉窗鰭片－單向開口
(louver)

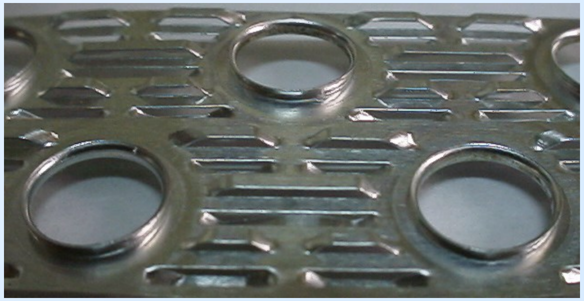




百葉窗鰭片－雙向開口
(louver)

裂口式鰭片－單向開口
(slit)





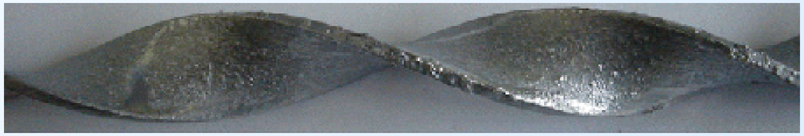
裂口式鰭片－雙向開口
(slit)

Which is better?

Intensified heat transfer techniques

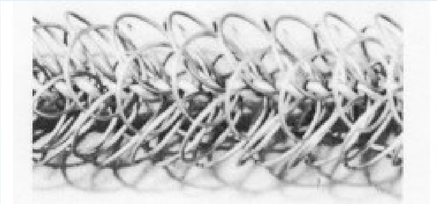
Tube-side:

- **Twisted-tape inserts**, which cause spiral flow along the tube length to increase turbulence



- **Coiled wire inserts**, which consist of a helical coiled spring and function as non-integral roughness

- **hiTRAN®**, which consist of a wire mesh with different densities. They are usually used to improve the heat transfer coefficient for the laminar regime

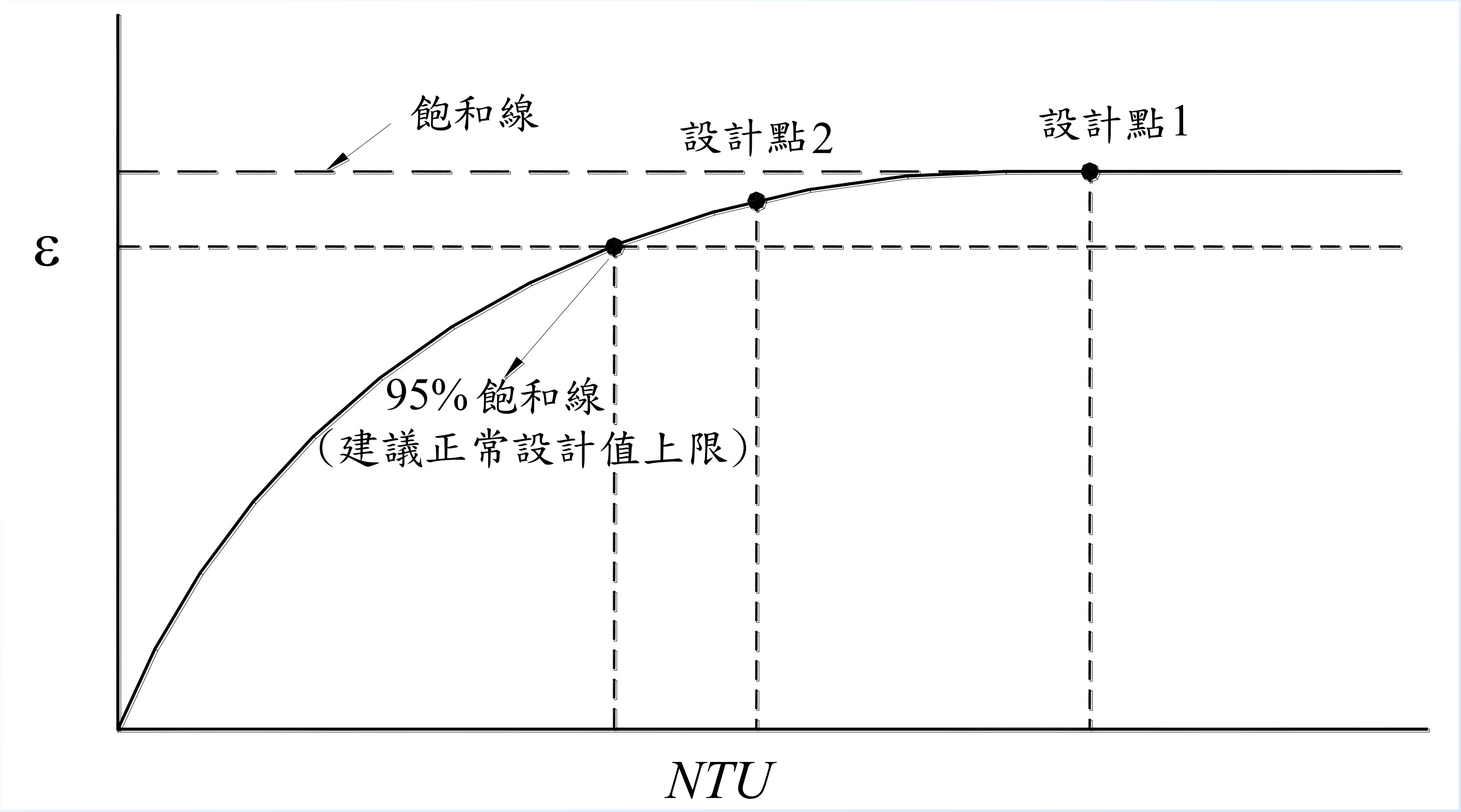


熱交換器的選擇原則（初步定性）

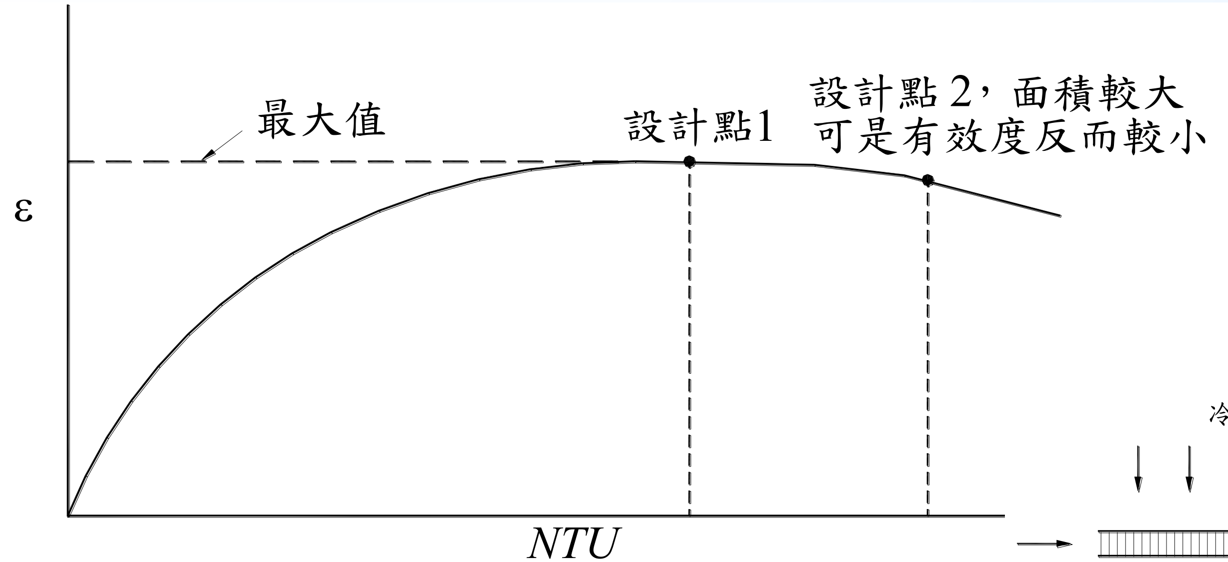
- Maximum Pressure
- Temperature range
- Fluids limitation
- Size range available
- Fouling & cleanability
- Plot area available
- Design life
- Location (maintenance)
- Is there a “temperature cross?”. If so, HX approaches counter flow is more appropriate

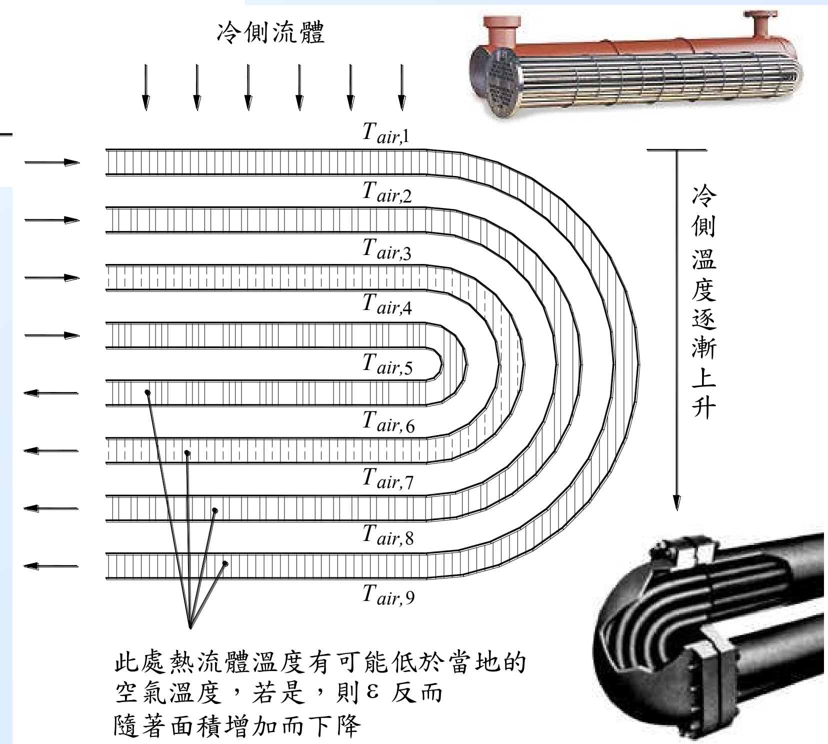
一些值得商榷的觀念

ε-NTU 關係是否已飽和？

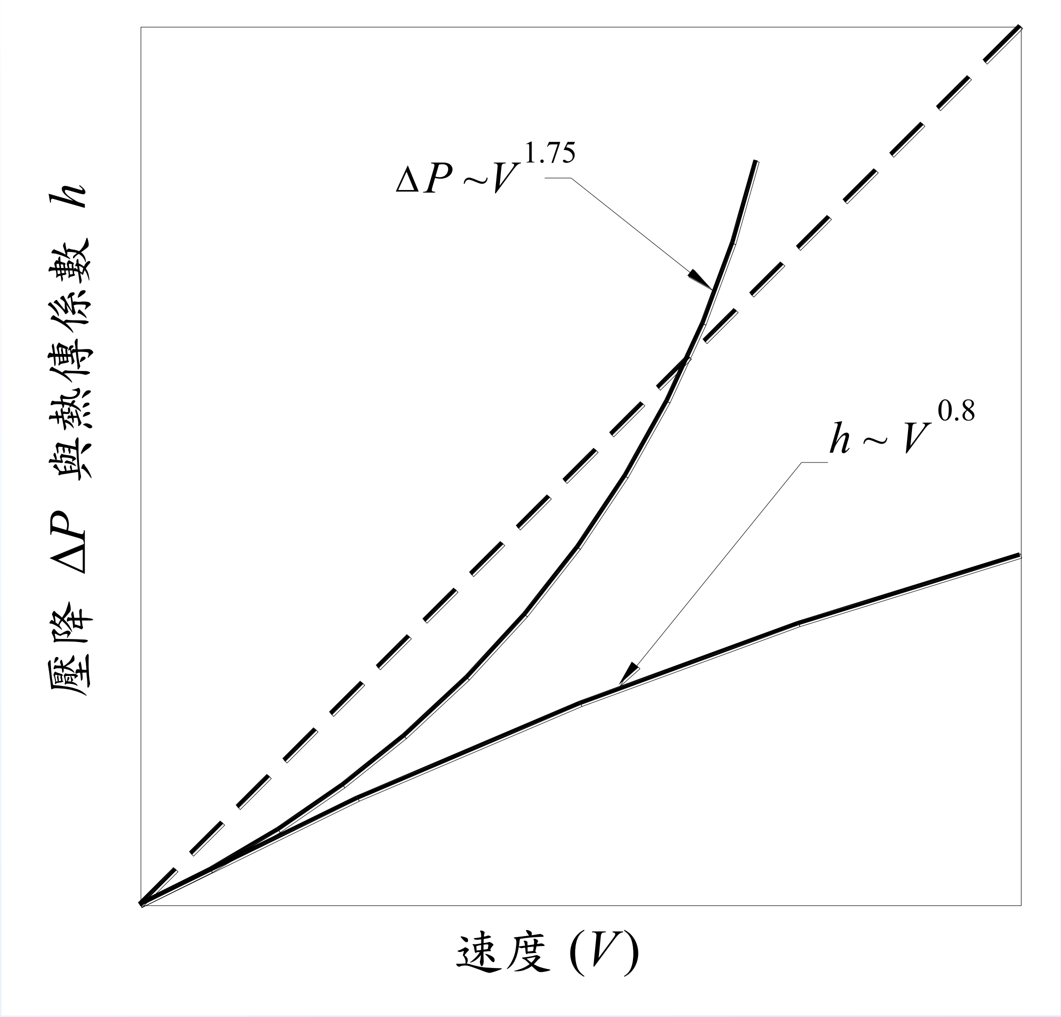


熱交換器越大熱傳效果越好？





熱交換器設計是否僅考慮熱傳量？

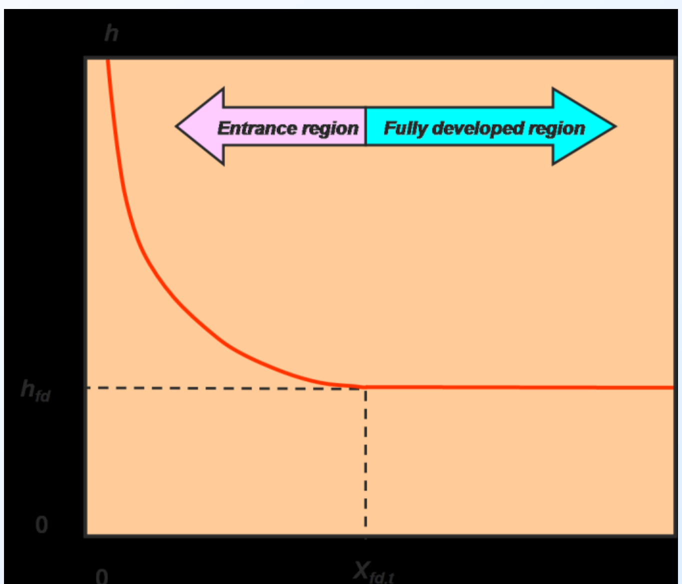


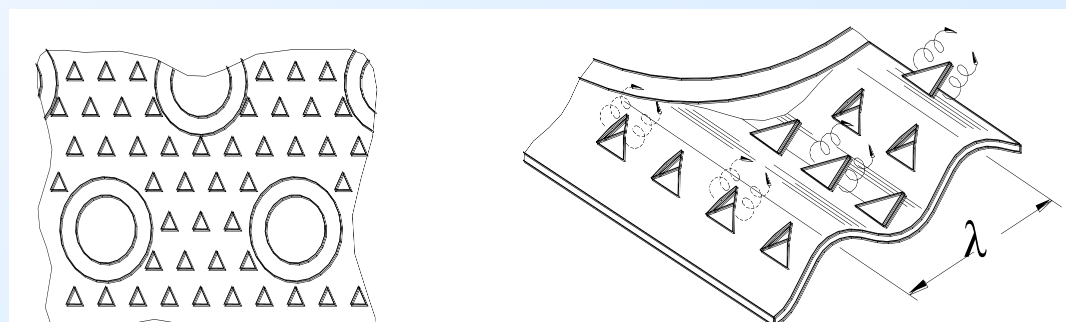
熱傳係數、壓降與速度的關係示意圖

熱傳增強技術
Airsdie

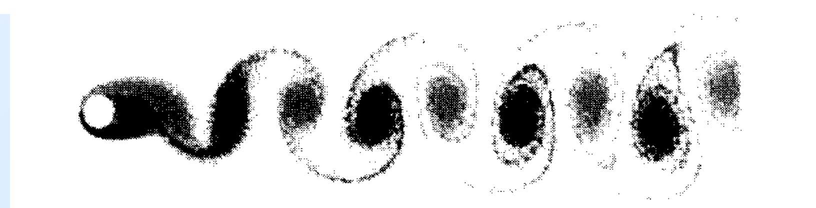
Some common ways for augmentation

- More Surface Area
- Thermal Boundary Layer Restart
- Instability
- Thermal Wake Management
- Swirl flow



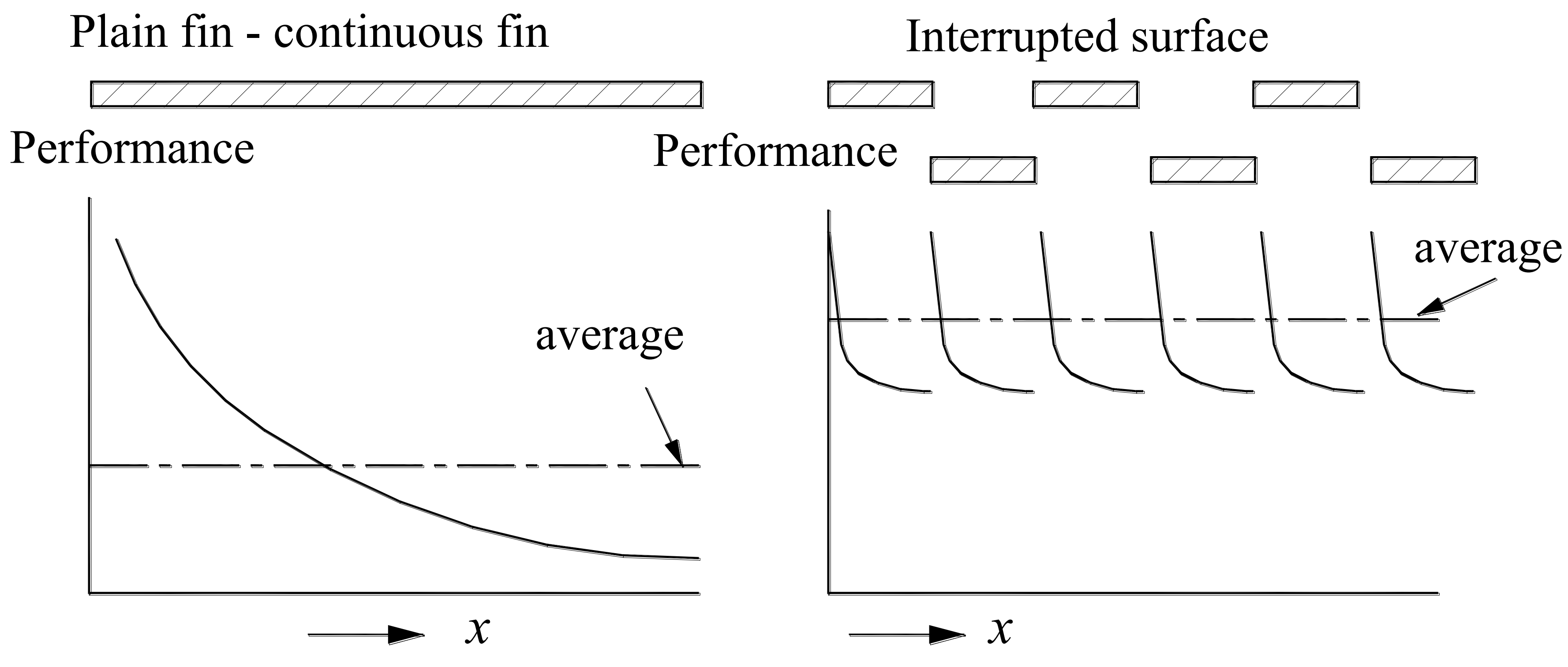


US patent 4817709



Concept of Interrupted surfaces

Boundary restart & Mixing



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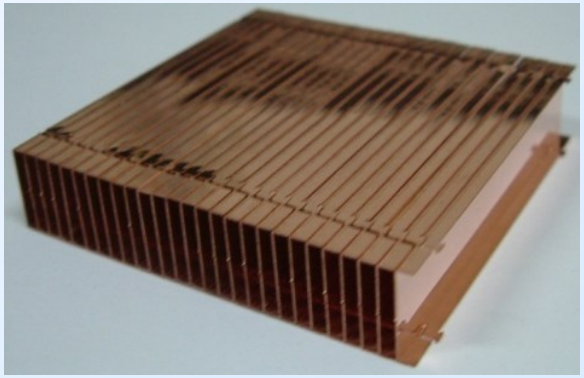
Various kinds of improvements - Implementations

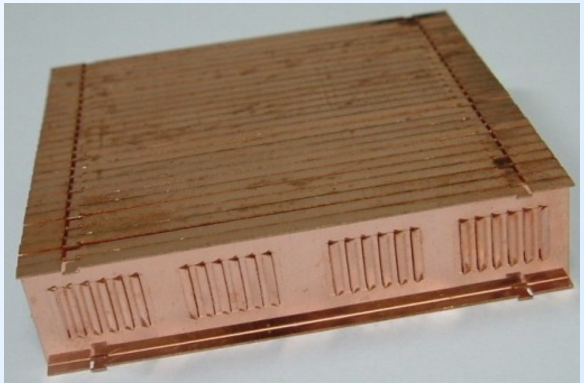
- Plate fin heat sink featuring heat transfer improvement by increasing heat dissipating surface. Generally, smaller fin spacing is used to accommodate more fin surface.

  Fin spacing can be lower than 1 mm

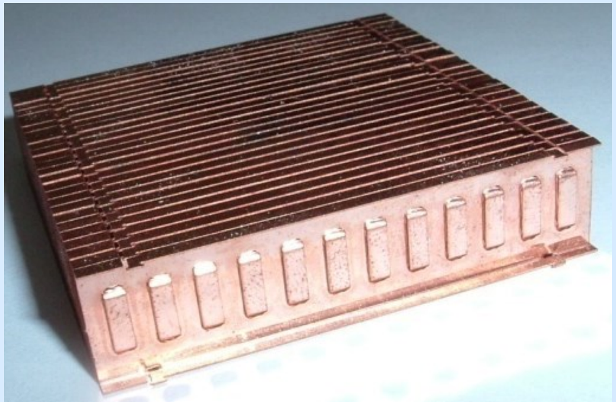


- Heat sink with interrupted fin geometry which improves convective heat transfer coefficient via periodical renewal of boundary layer such as slit or louver fin.

louver fin



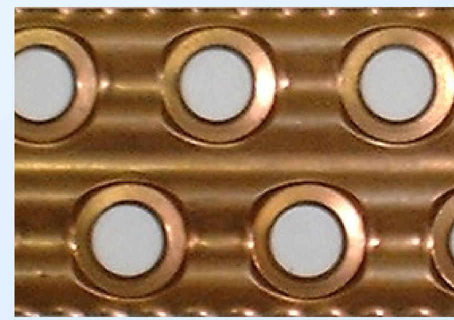
slit fin

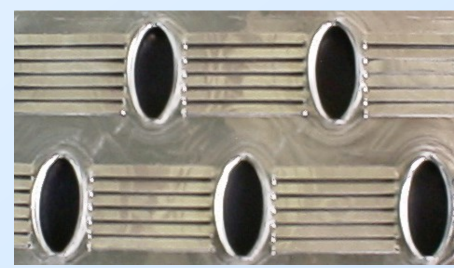


Various Fin Patterns





Interrupted surfaces..

- Provide effective heat transfer augmentations at medium and high velocity with significant pressure drop penalty.
- Nearly ineffective at low velocity but still suffer from considerable pressure drop.
  - Duct flow effect.

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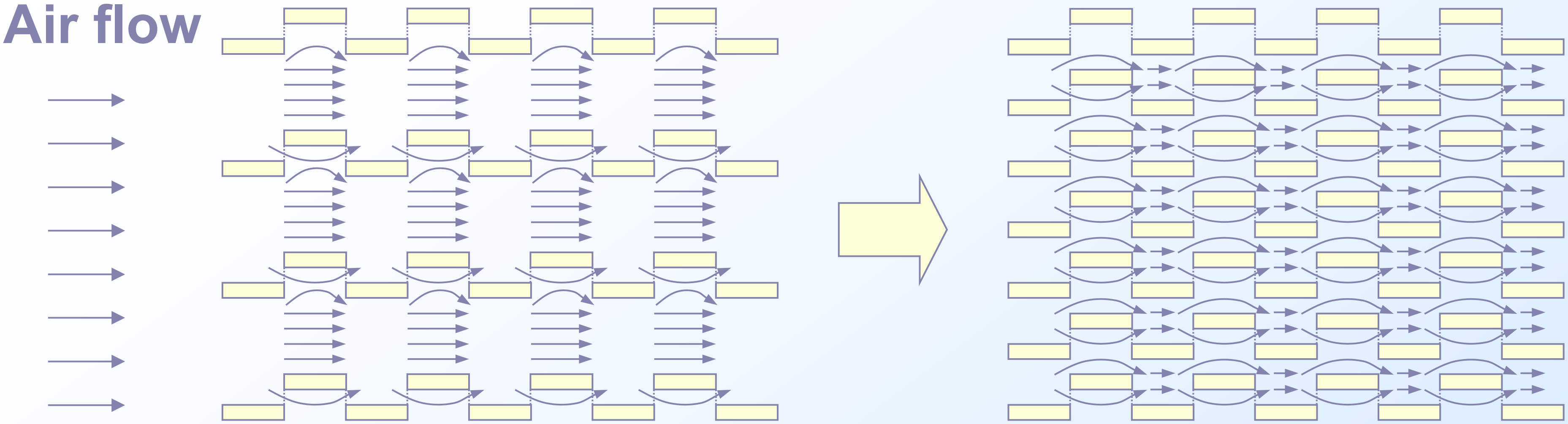
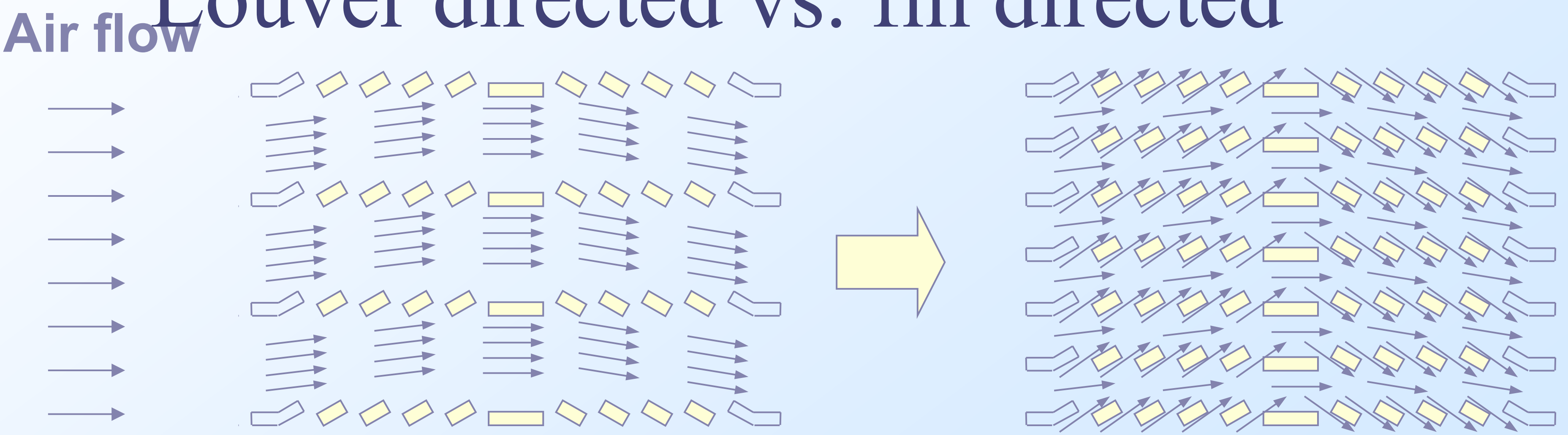
Effects of Periodic Entrance/Exit



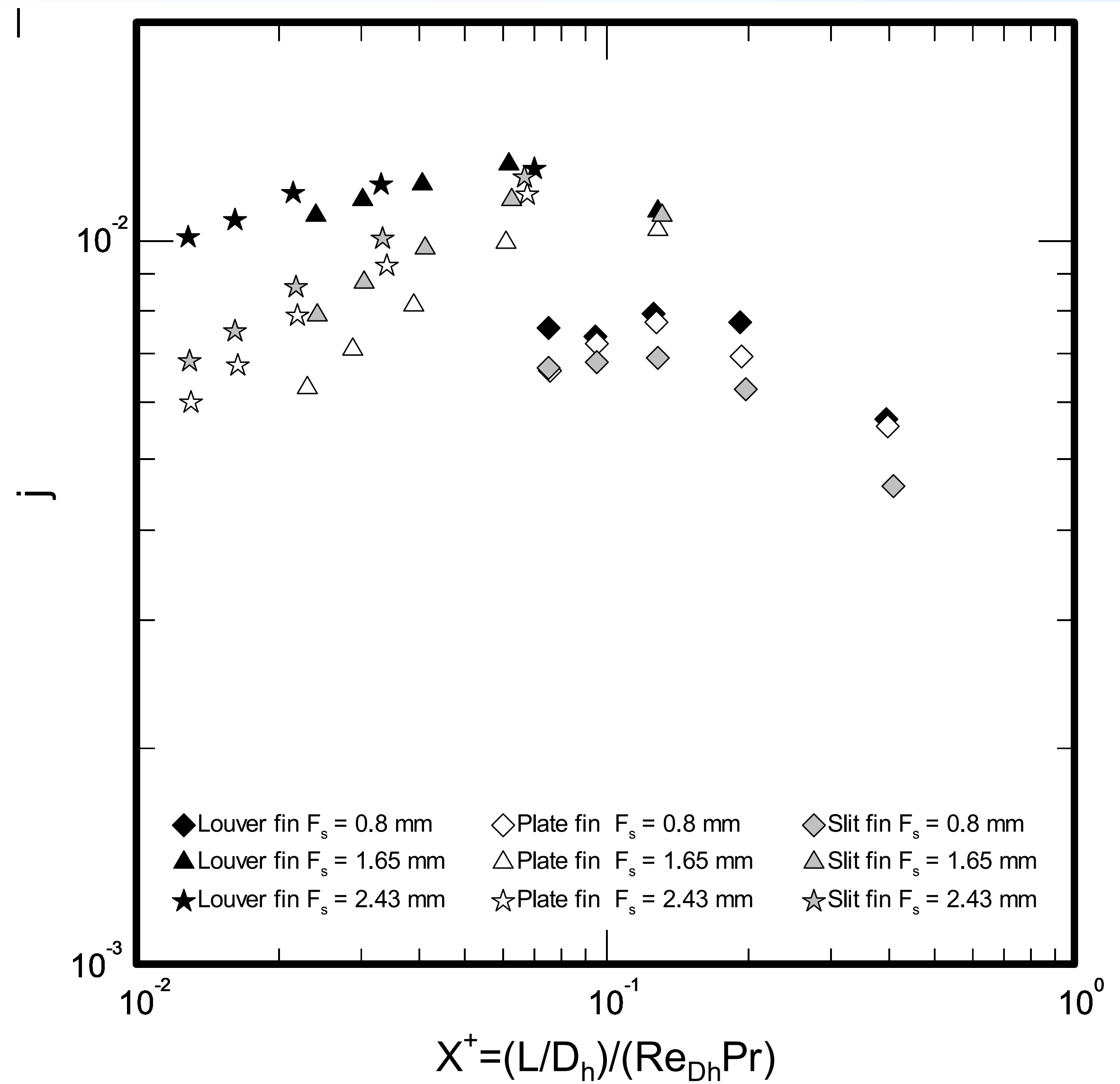
Louver directed vs. fin directed



SCHEMATIC OF DUCT FLOW VS. FIN-DIRECTED FLOW FOR LOUVER FIN GEOMETRY AT SMALLER AND LARGER FLOW VELOCITIES. (Yang et al. IJHMT, 2007)

Interrupted surfaces..

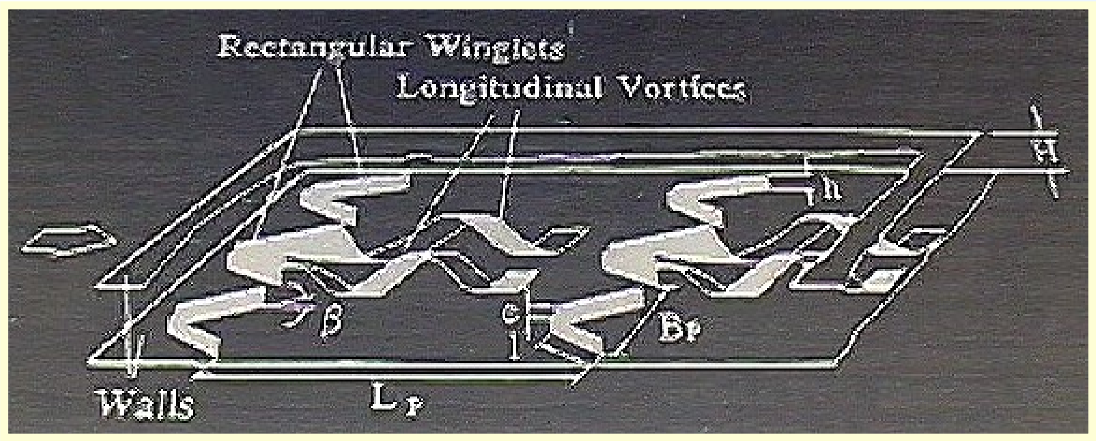
- Smaller fin spacing accentuates the duct flow effect, resulting in fully developed flow and deteriorate the heat transfer performance.



INVERSE GRAETZ NUUMBER NUMBER X$^+$ VS. *j* FOR LOUVER, SLIT AND PLATE FIN. (Yang et al., IJHMT, 2007)

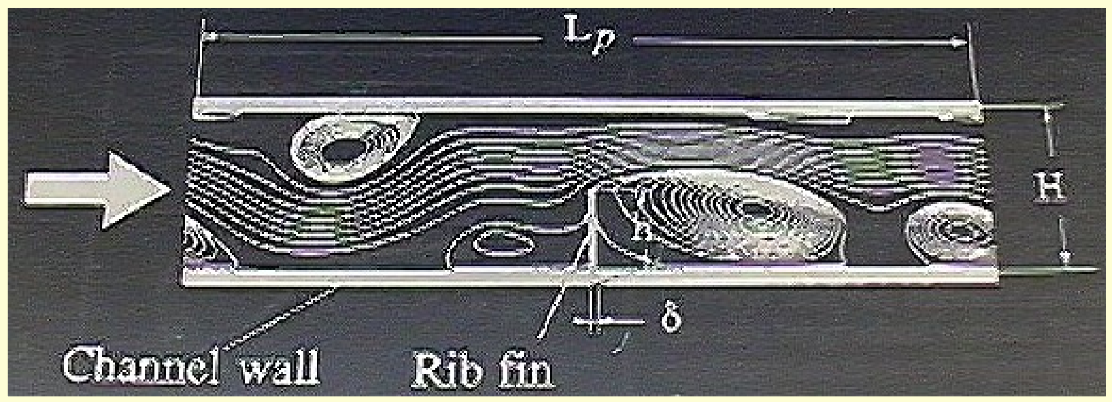
Type of vortex generators

Longitudinal vortex outperforms the transverse vortex



Longitudinal vortex

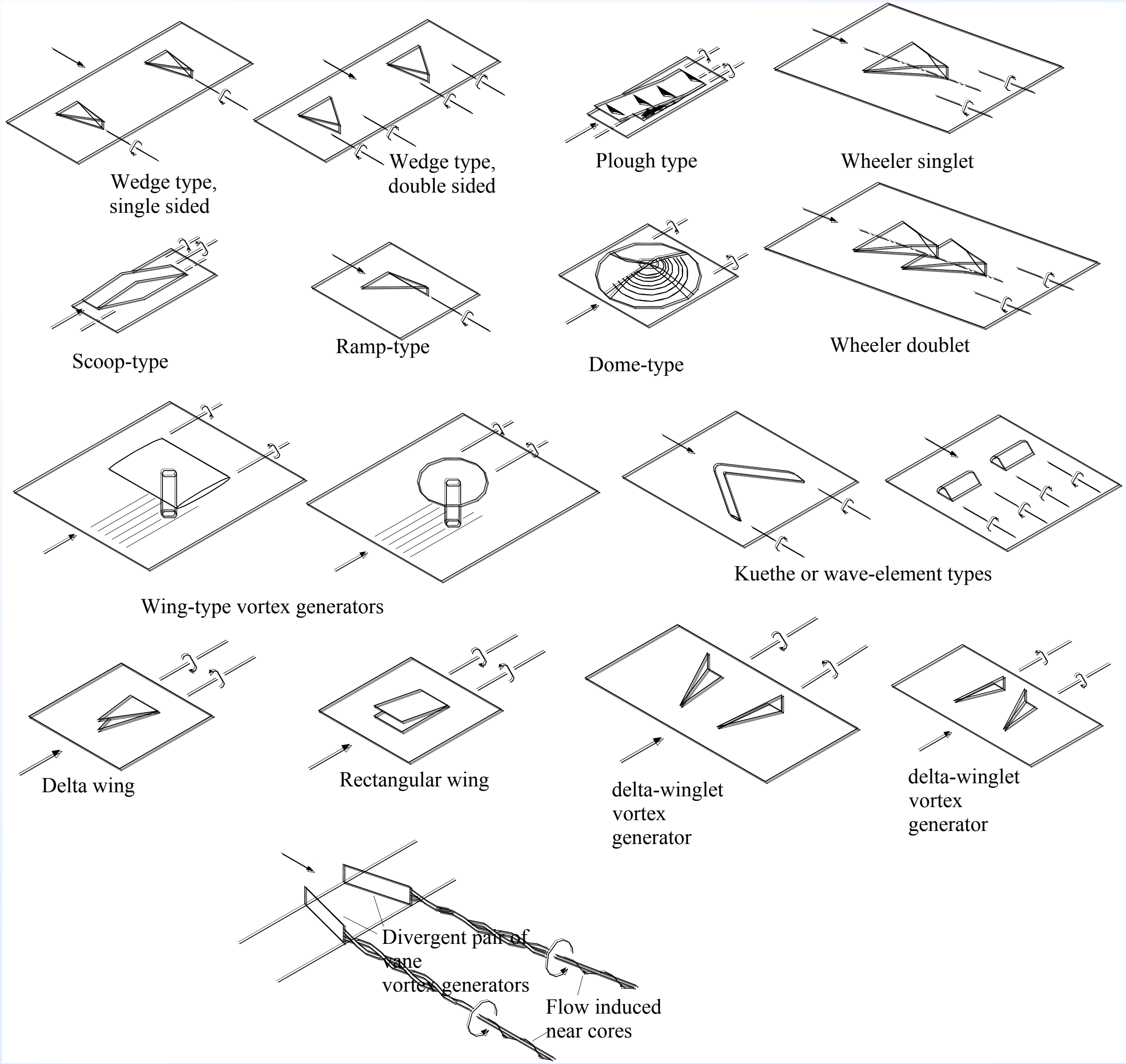
Transverse vortex



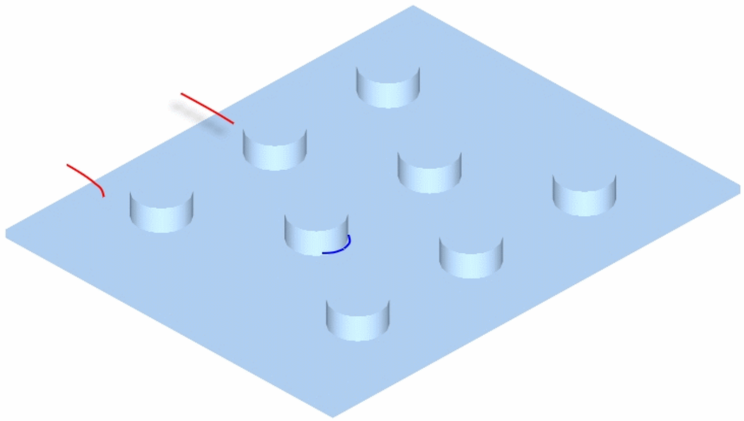
Typical LVGs

Benefits of vortex generator

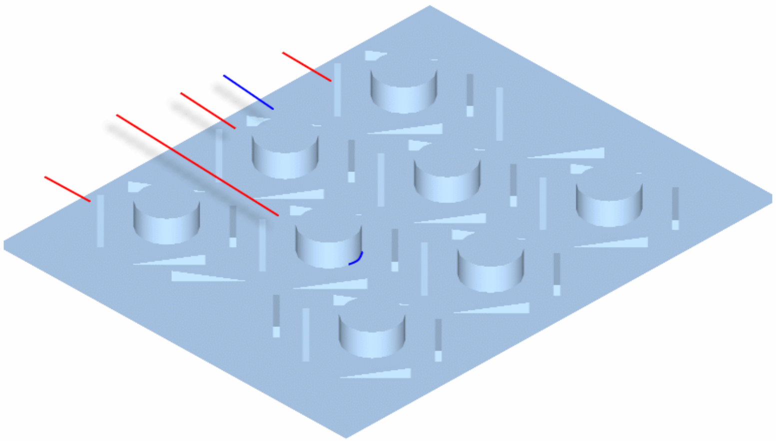
- Prevent Boundary Layer separation
- Improve heat transfer performance with acceptable pressure drop



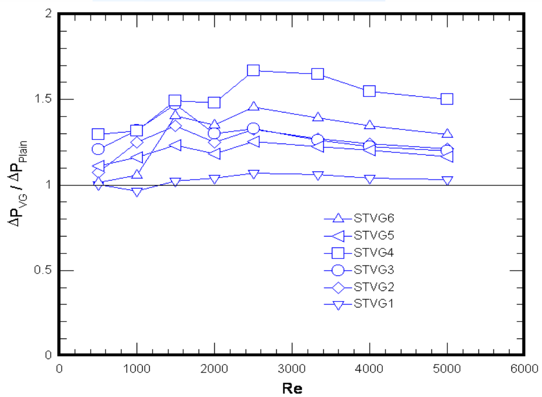
Int. J. of Heat and Mass Transfer, Vol. 45, pp. 1933-1944.
Int. J. of Heat and Mass Transfer, Vol. 45, pp. 3803-3815.

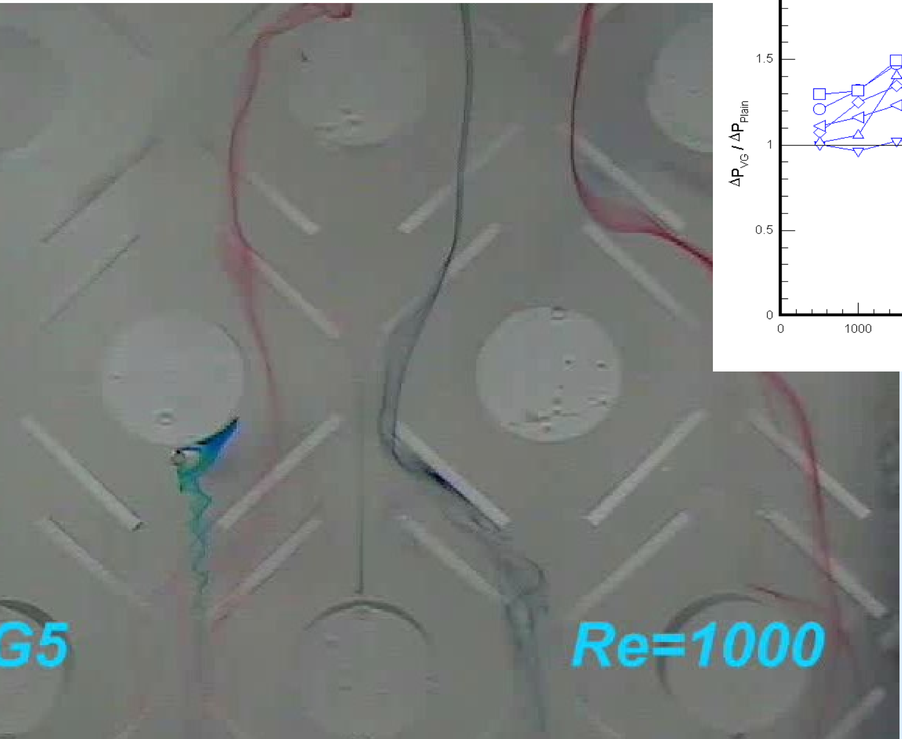


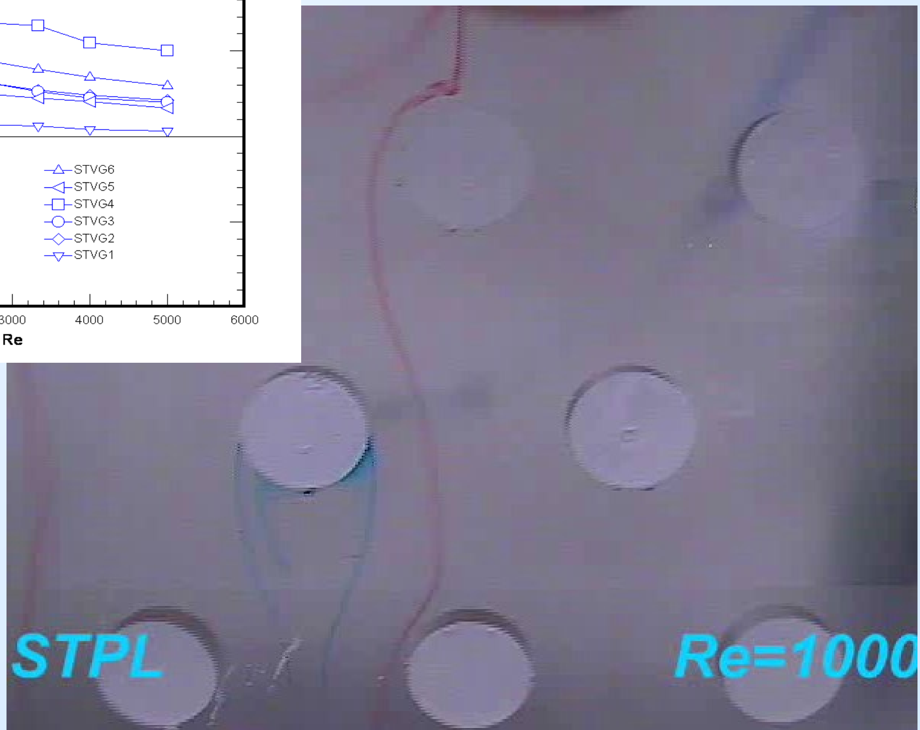
Re=1500,STPL



Re=1000,STVG5

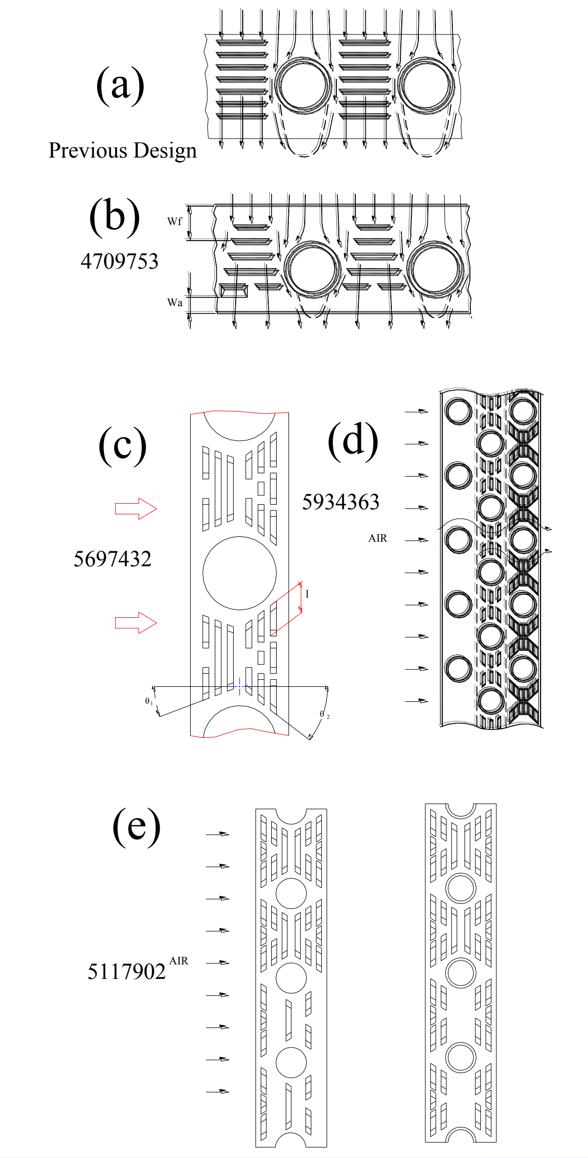






Design by Non-uniformity

1. Place the enhancement at low heat transfer region.
2. Check the effective local temperature difference. Placing enhancements at those having lower temperature difference are generally more effective.

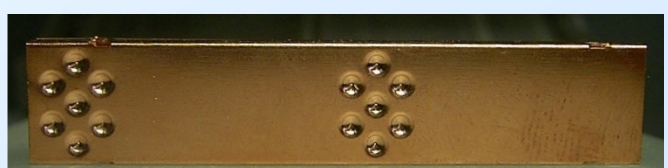


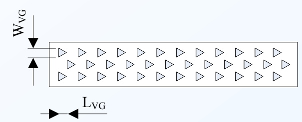
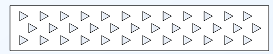
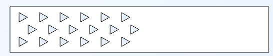
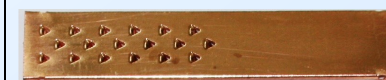
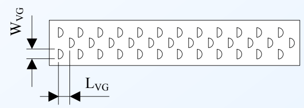
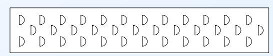
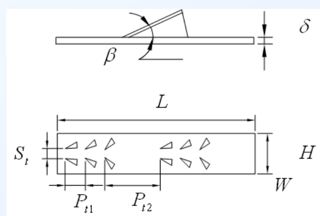
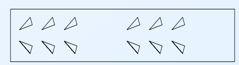
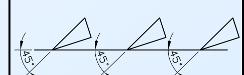
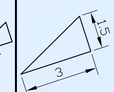
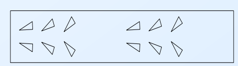
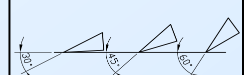
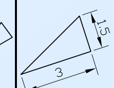
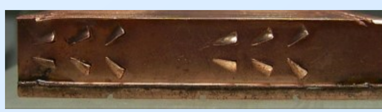
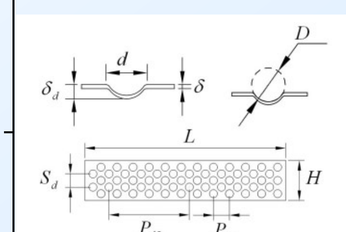
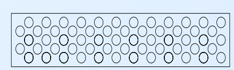
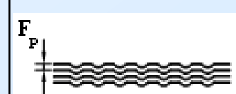
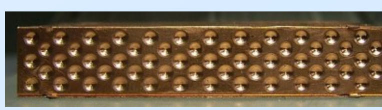
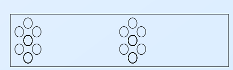
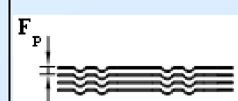
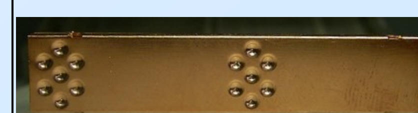
Vortex Generators..

- Implementations

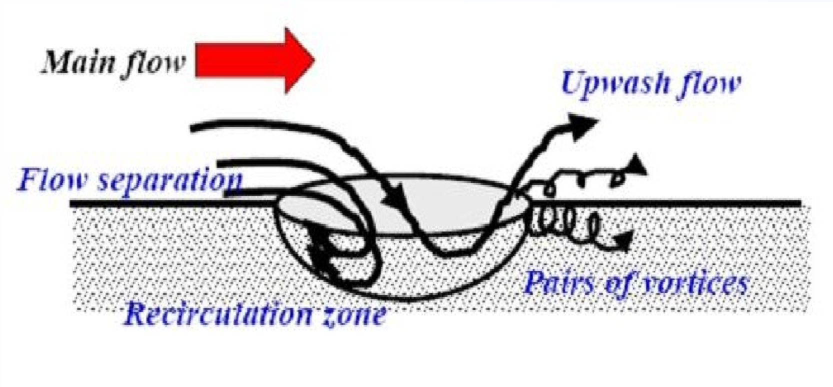
Type III: Heat sink with dense vortex generator. The enhancements introduce swirl flow, Coanda deflection flow or destabilized flow field from vortex generators or dimple/protrusion structure. The general arrangement is using inline or staggered layout such as semi-circular, delta and dimple vortex generator.

Type IV: Heat sink with loose vortex generator: The enhancements of this category are still vortex generators or dimple/protrusion structure but with sparse arrangement of vortex generator.

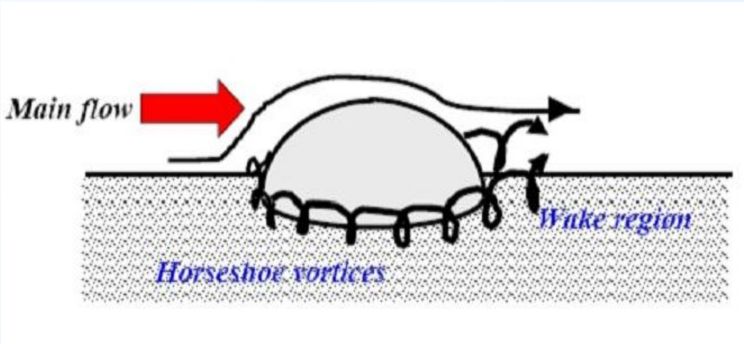


Heat sink	Nomenclature	Side view	Dimension		Photos of test sample
(a) Plate	-		-	-	
(b) Delta VG			-	-	
(c) Delta VG+Plate			-	-	
(d) Semi-circular VG			-	-	
(e) Triangular VG					
(f) Triangular Attack VG					
(g) Dimple VG				-	
(h) Two Groups Dimple VG				-	

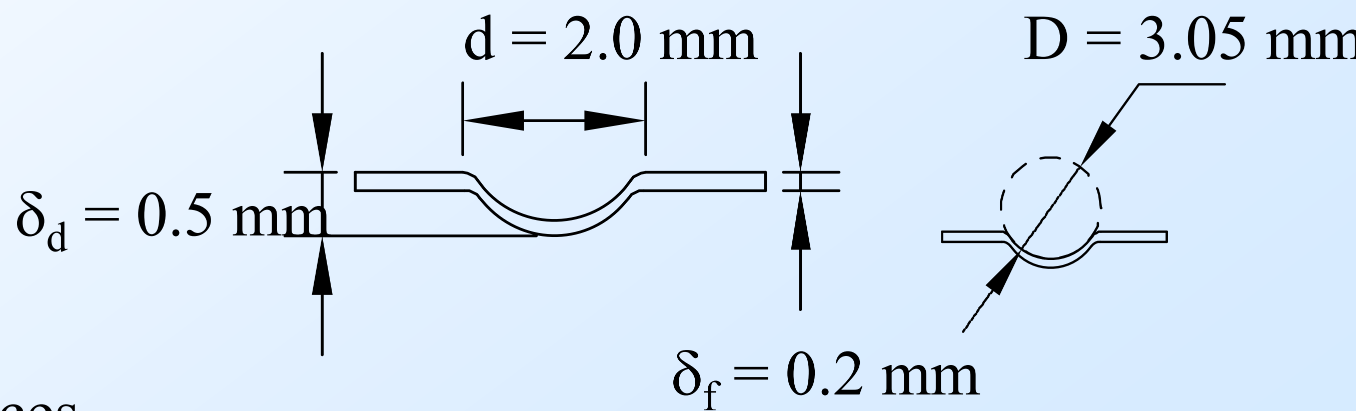
The original concept of Using dimple..



(a) Dimple



(b) Protrusion



- Drag reduction
- Longitudinal Vortices
- In this study, fin thickness is 0.2 mm,
- the length of cavity is 2 mm, effective cavity depth is 0.3 mm

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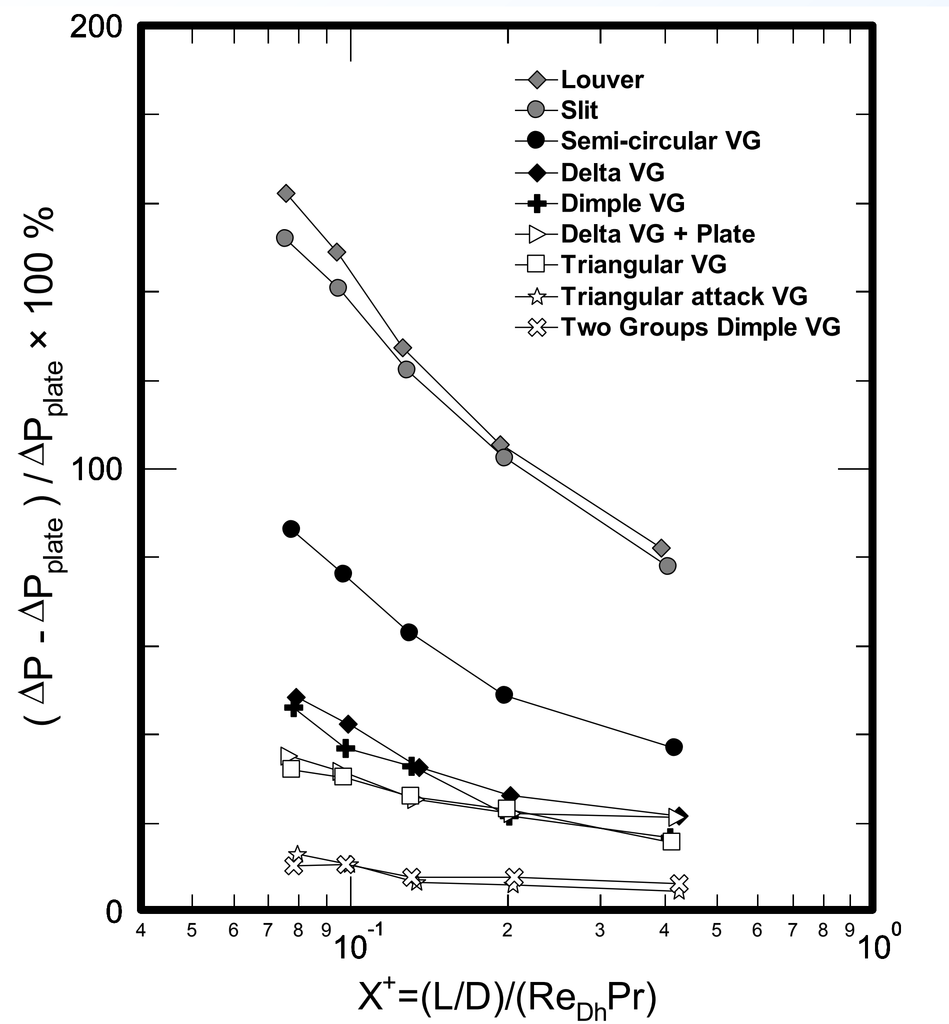
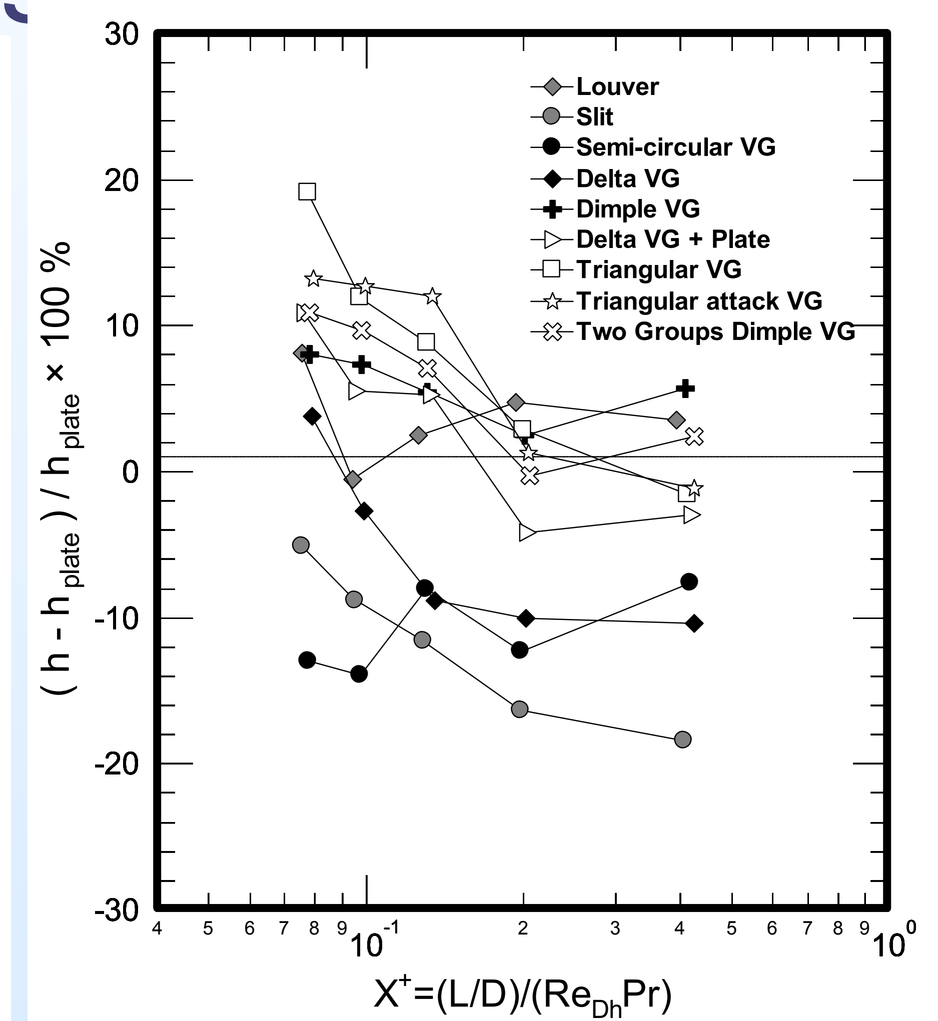
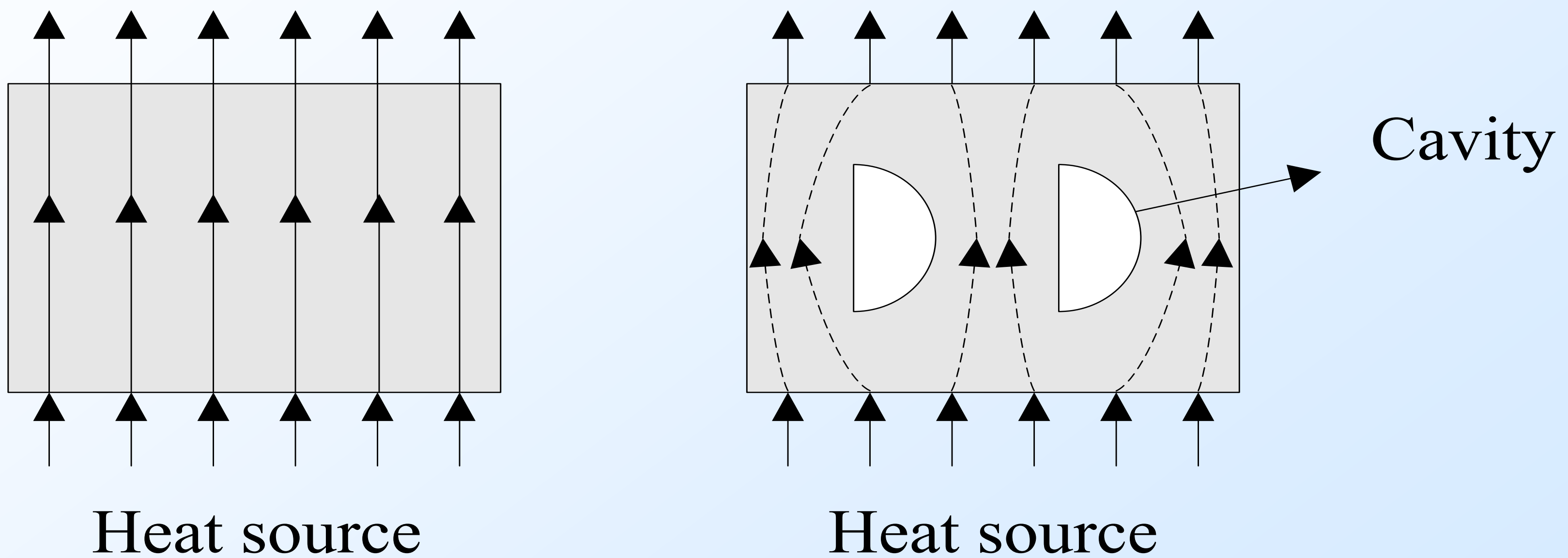
Performance comparison
Fin spacing = 0.8 mm





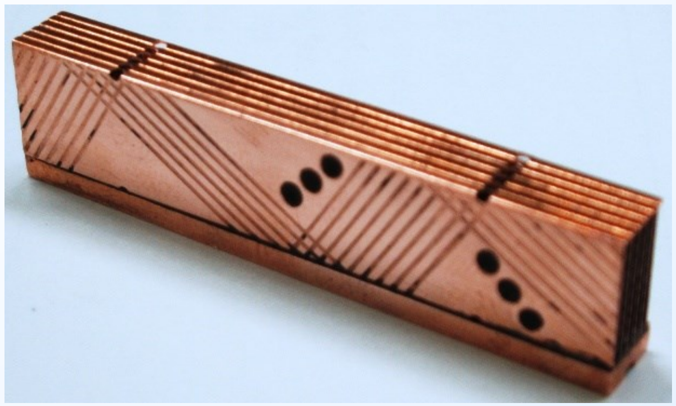
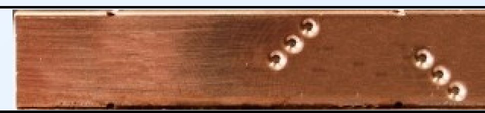
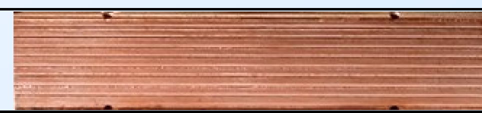
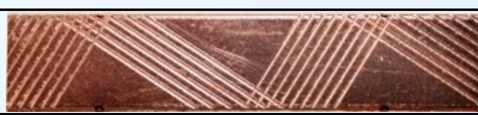
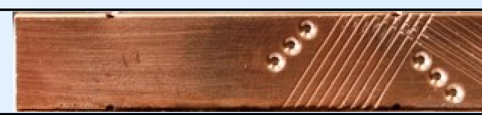
An extra problem for some VG & interrupted surfaces

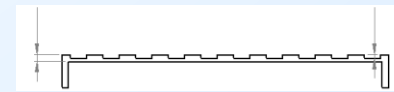


Very small fin spacing also jeopardize the formation of LVG

So, what's next?

Oblique Dimples with cannelure structure

	
(plate fin)	(oblique dimple gap 4-12fin)
	
(oblique dimple gap 6-12 fin)	(cannelure fin I)
	
(cannelure fin II)	(oblique dimple gap 4-12 cannelure fin)
	
(oblique dimple gap 6-12 cannelure fin I)	(oblique dimple gap 6-12 cannelure fin II)
	



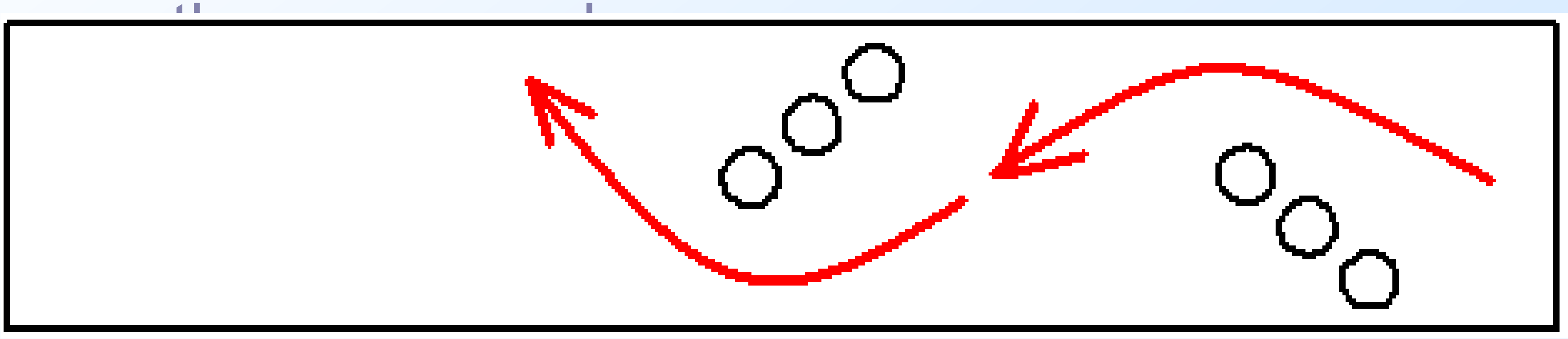
Cannelure channel

Depth: 0.1 mm

Width: 0.4 mm

The original idea for oblique dimple..

- Concavity + Dimple
- Lengthen the flow path
- No need for significant amount dimples
  - Reduce the number of dimples to decrease



The idea of cannelure channels..

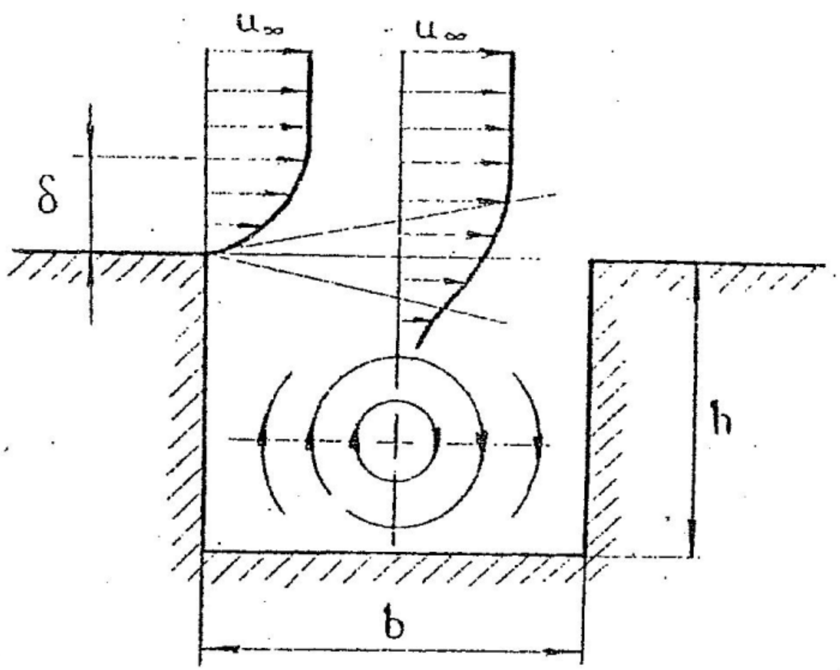


Fig. 1. Model of flow in an isolated rectangular pit [2] in a wall.

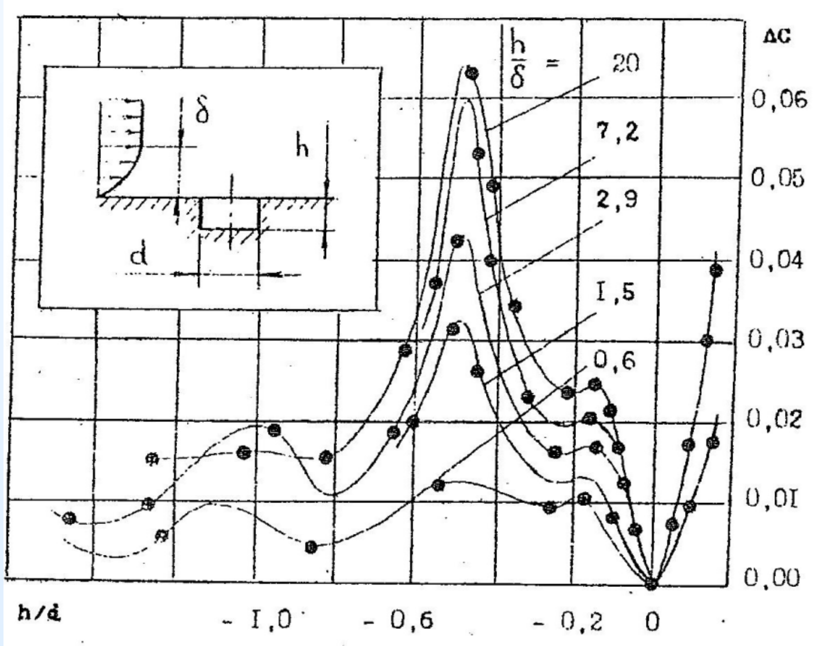


Fig. 2. Increment of the drag coefficient as a function of the dimensionless depth of the pit [4].

Heat Transfer Research 25(1), 1993, 22-56

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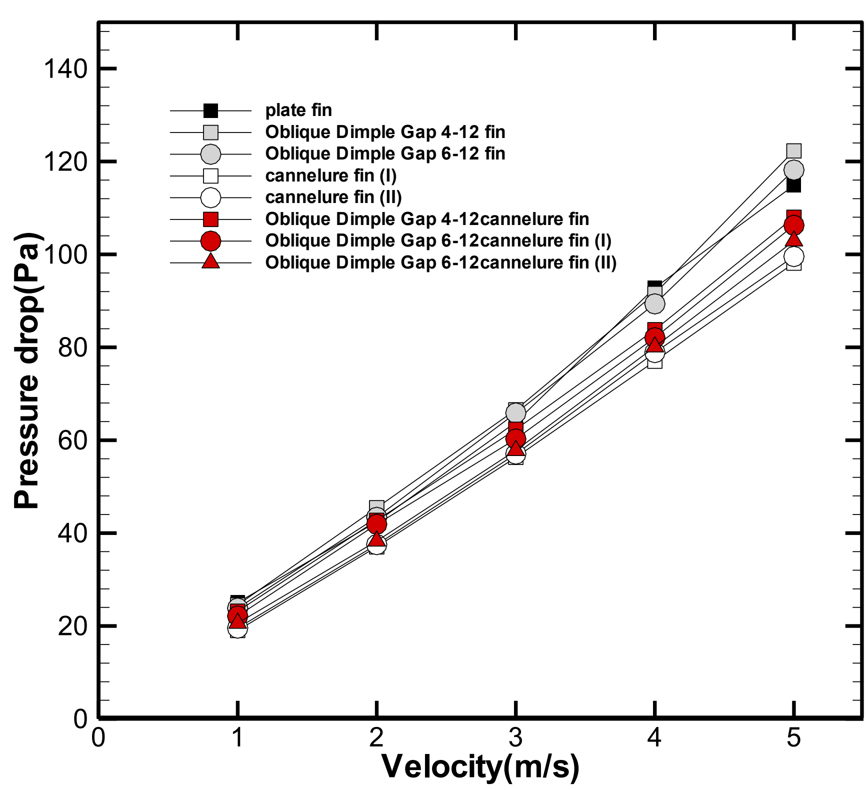
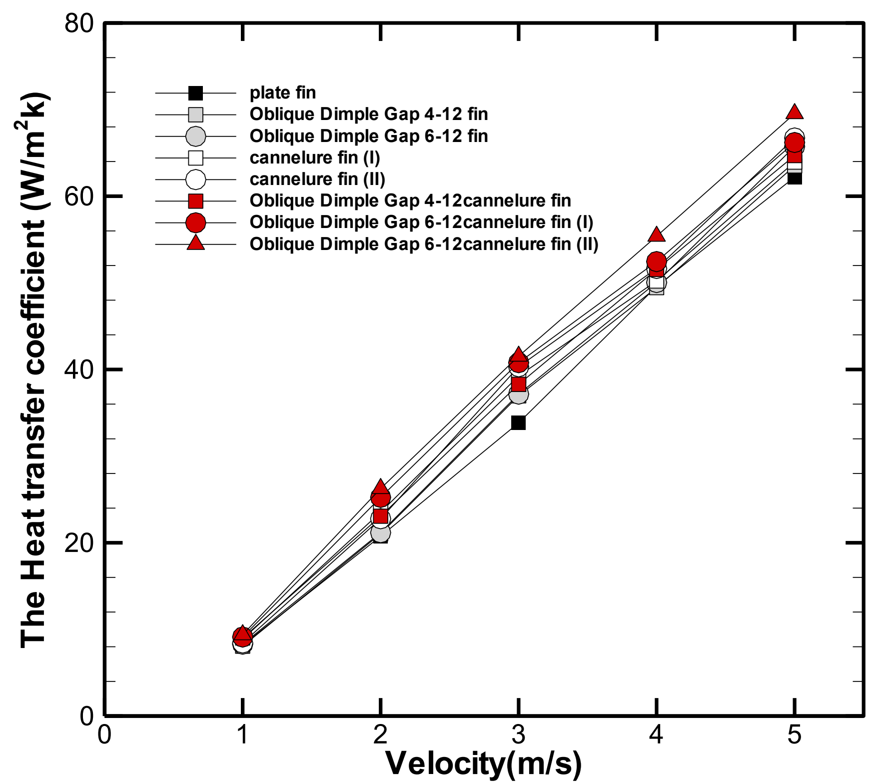
Results:

More than 20% increase HTC





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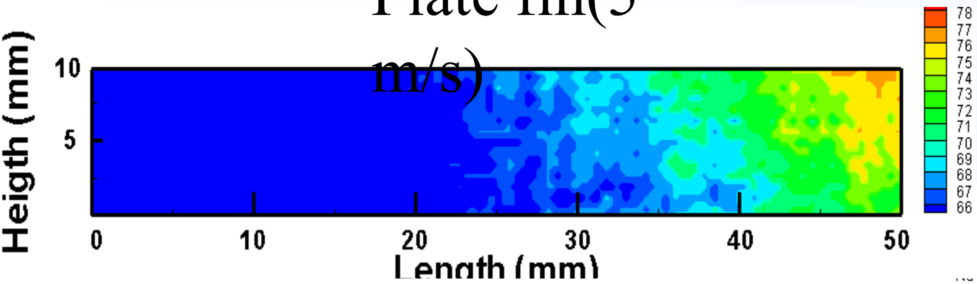
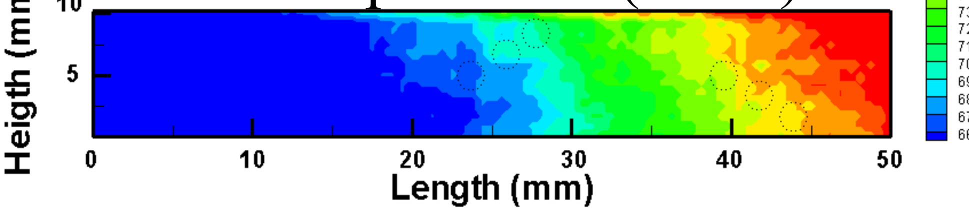
Performance & IR

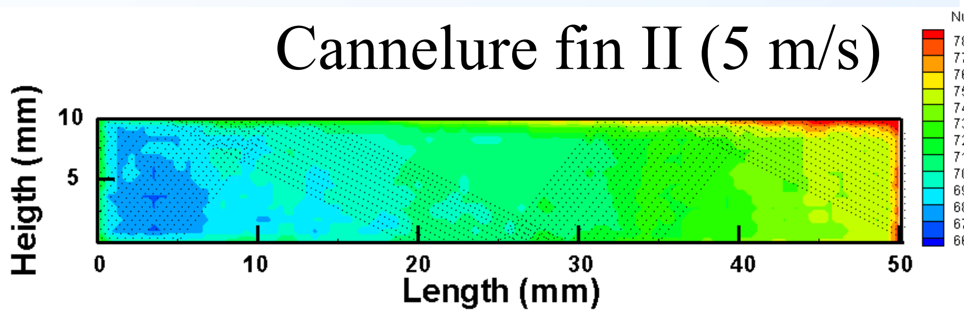
Plate fin(5 m/s)



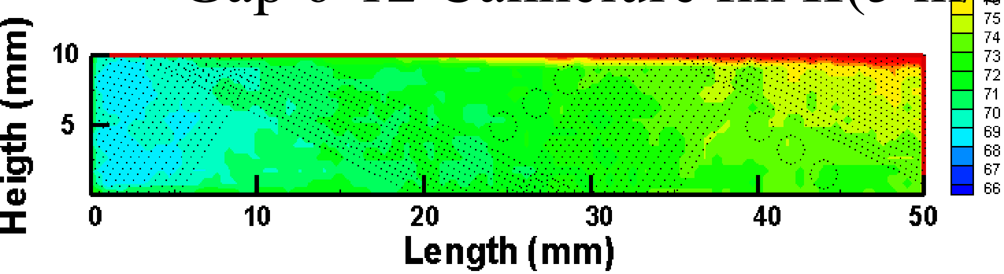
Gap 6-12 fin(5 m/s)

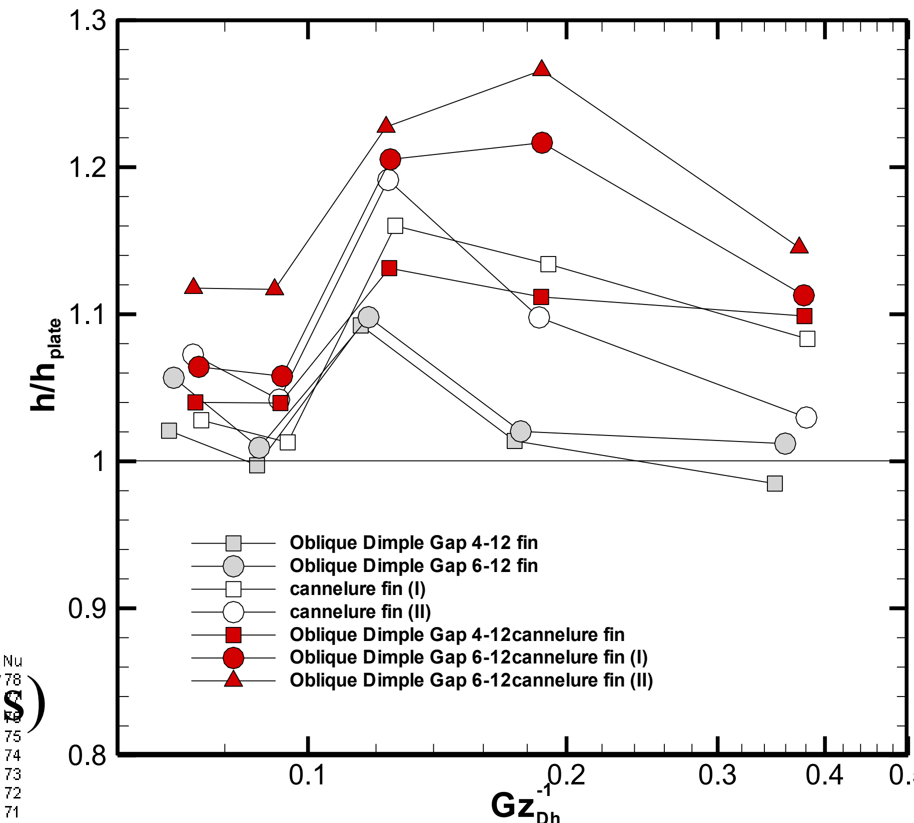


Cannelure fin II (5 m/s)



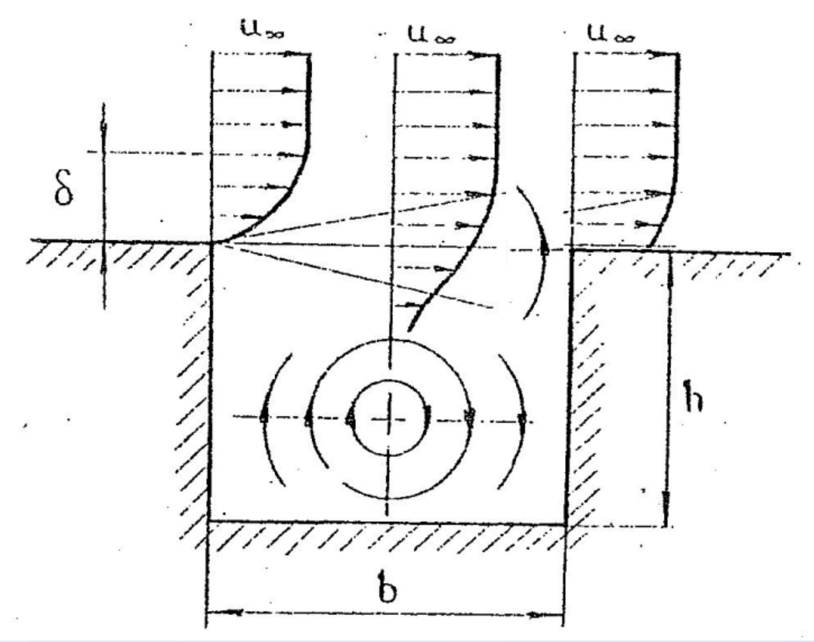
Gap 6-12 Cannelure fin II(5 m/s)





Why cannelure structure is working? – One possible reason

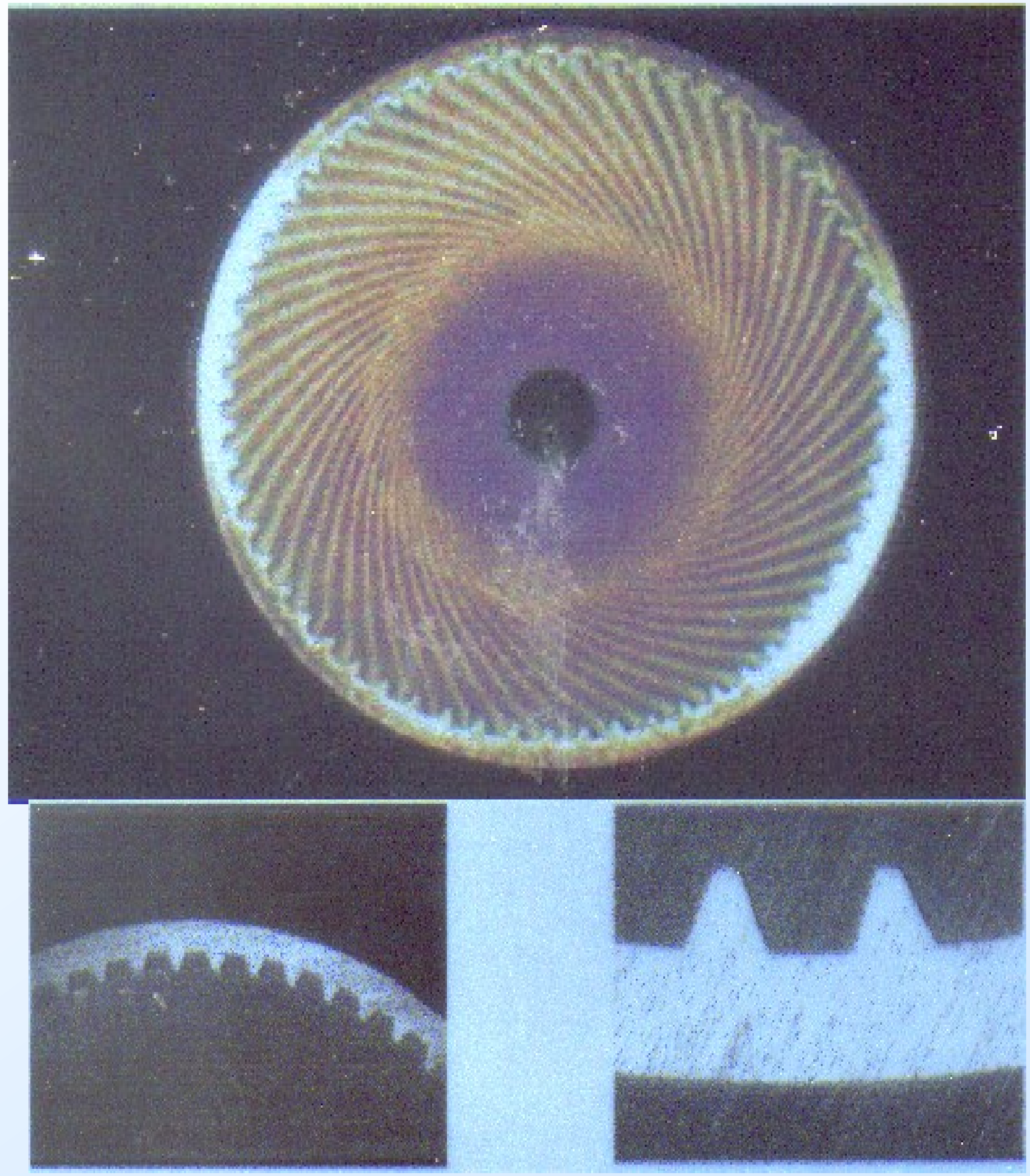
- Reduce the BL thickness to improve the heat transfer performance for fully developed region.
- It acts like a "suction" device.



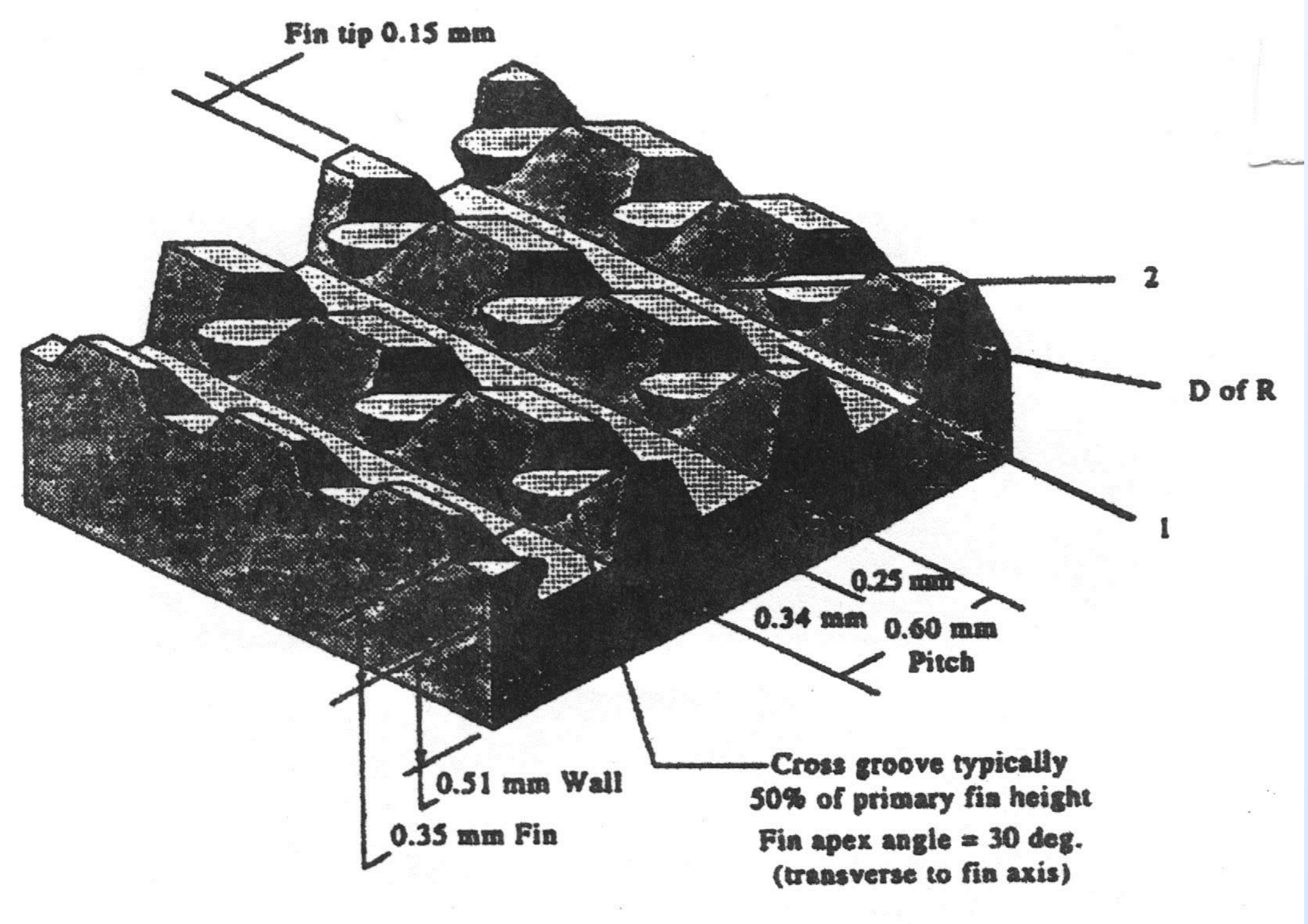
熱傳增強技術 - Liquid Side

TABLE Ⅱ The generations of heat transfer technology

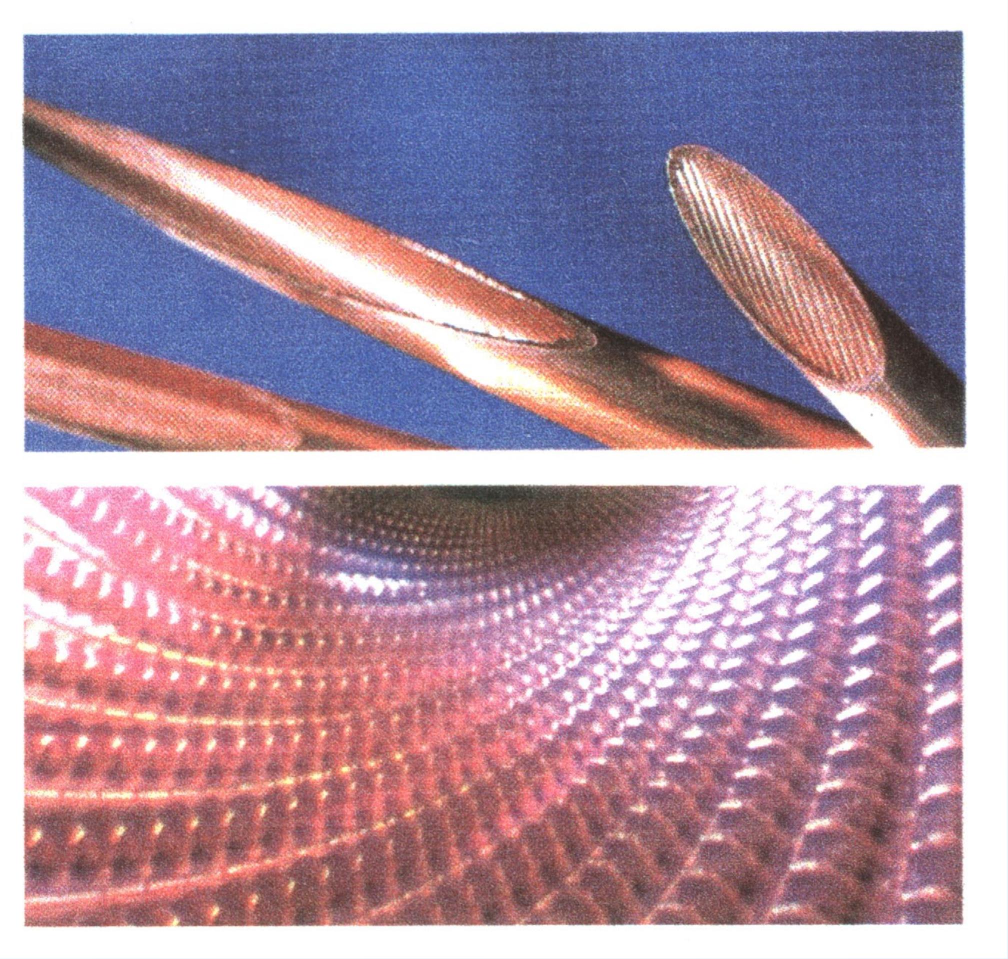
Tube-and-plate fins, single-phase	
1st generation	bare tube
2nd generation	plain fins
3rd generation	longitudinal vortex generators on fins
In channel, single-phase	
1st generation	smooth channel
2nd generation	2-D roughness
3rd generation	3-D roughness
Outside tubes, boiling	
1st generation	smooth tube
2nd generation	2-D fins
3rd generation	3-D fins and metallic matrices
In-tube, evaporation	
1st generation	smooth tubes
2nd generation	massive fins and inserts
3rd generation	micro-fins
Outside tubes, condensing	
1st generation	smooth tubes
2nd generation	2-D fins
3rd generation	3-D fins and metallic matrices



Two dimensional microfin tube



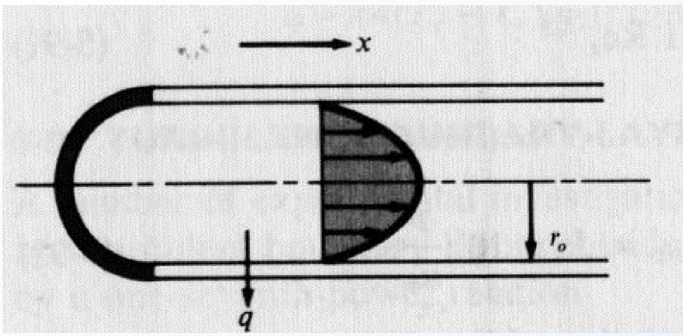
- Three dimensional microfin tube (Olin)

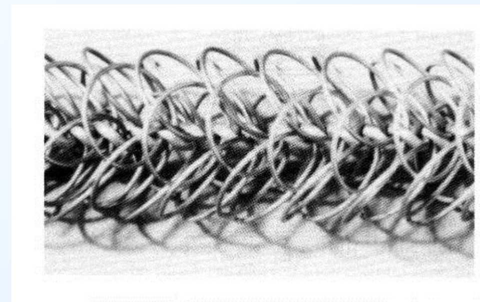


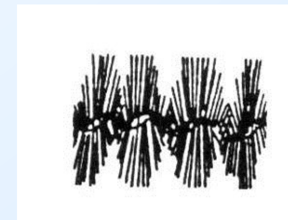
德国 **KME** 公司开发的三维微肋管

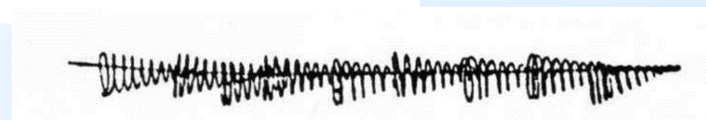
-
- Three dimensional microfin tube(KME)

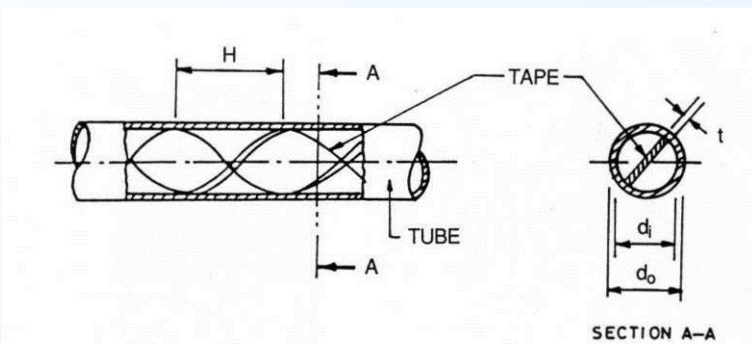
Augmentation of single-phase flow – Laminar Flow

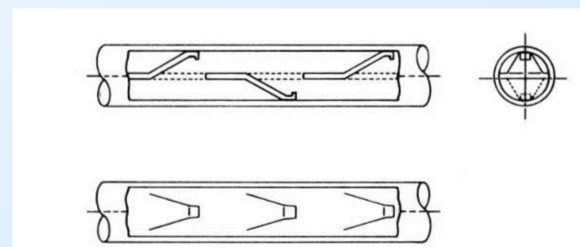


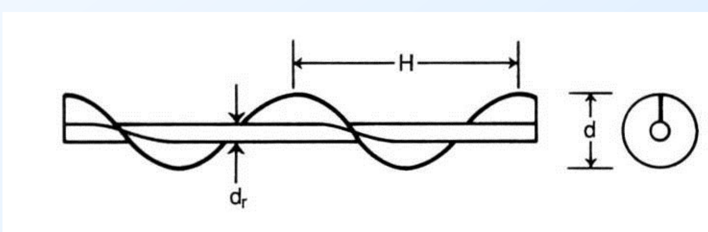


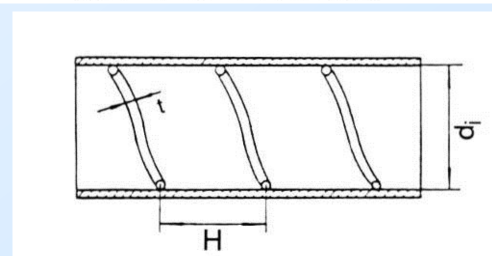




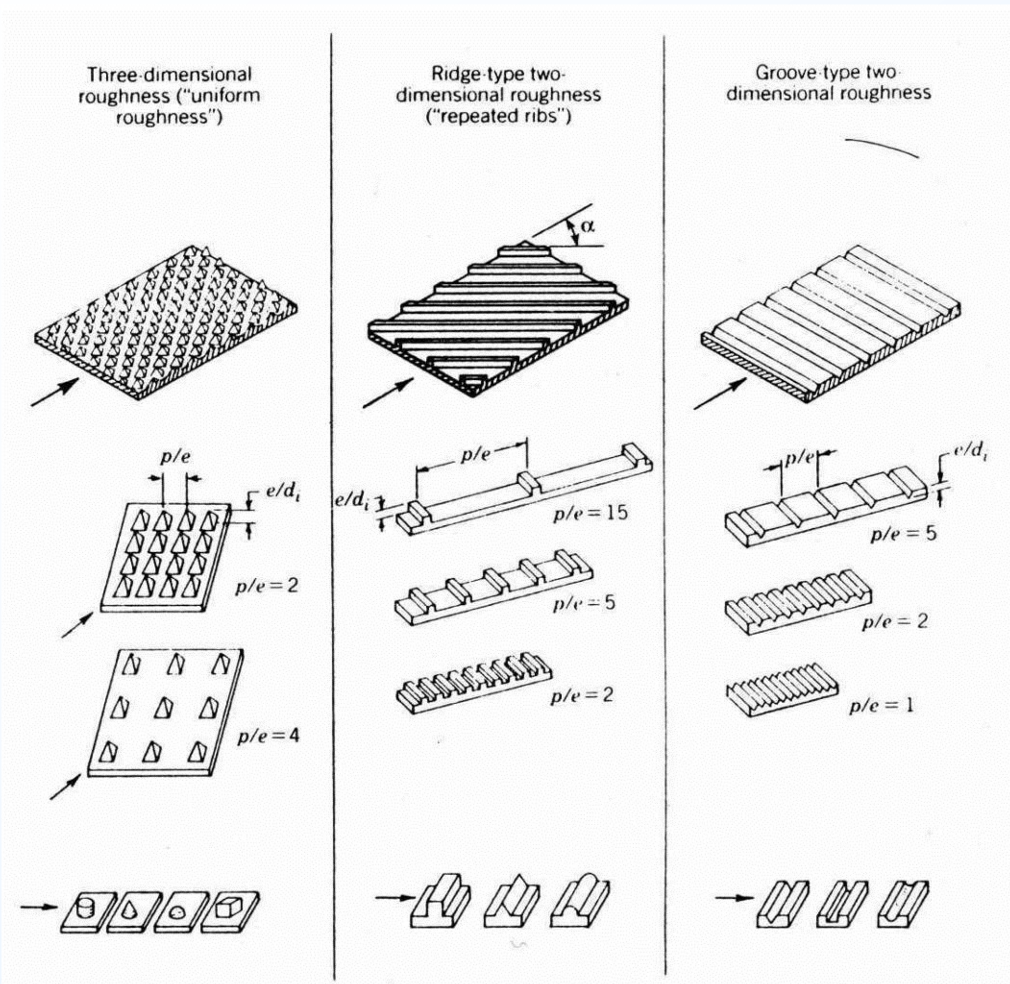


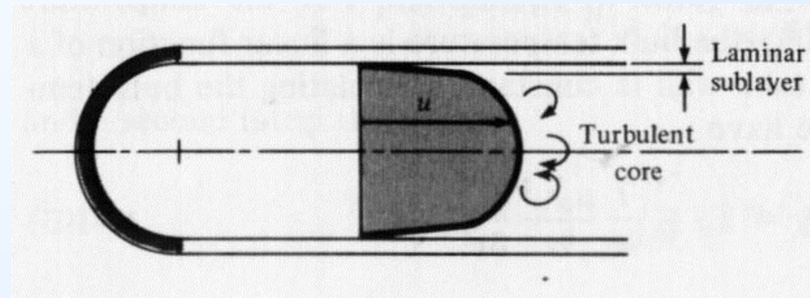


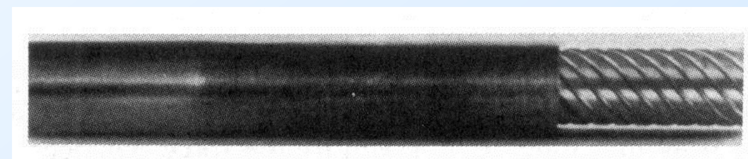


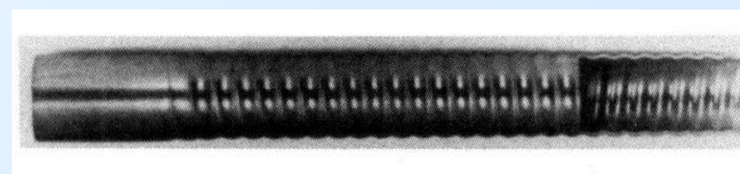


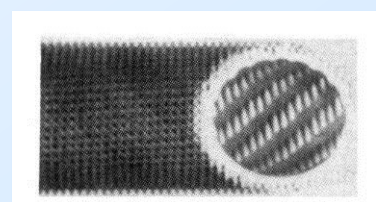
Augmentation in turbulent flow

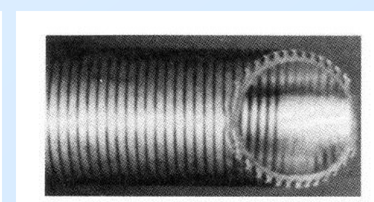






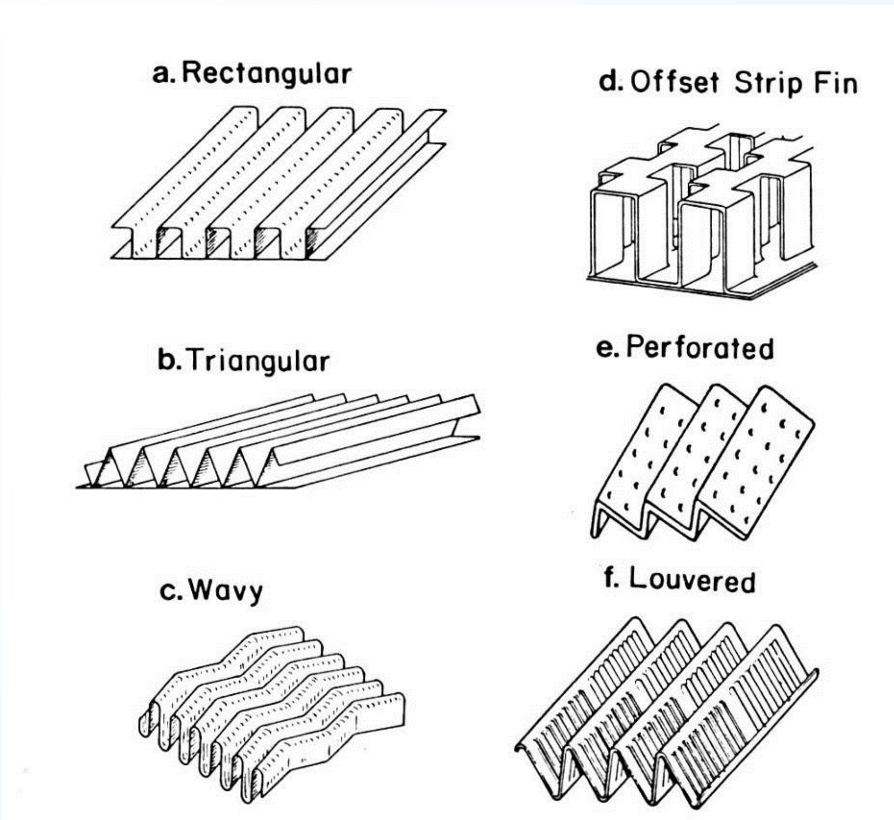


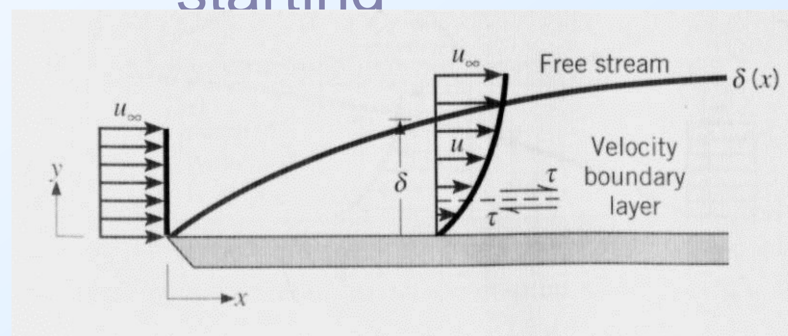


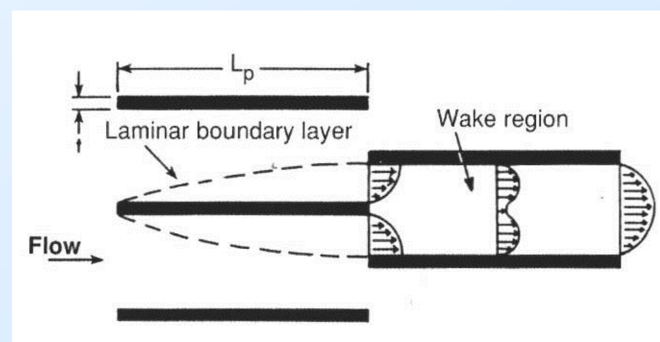


Augmentation – with the presence of fins

- OSF interrupted surface
- Boundary layer re-starting

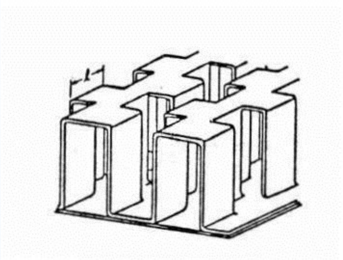


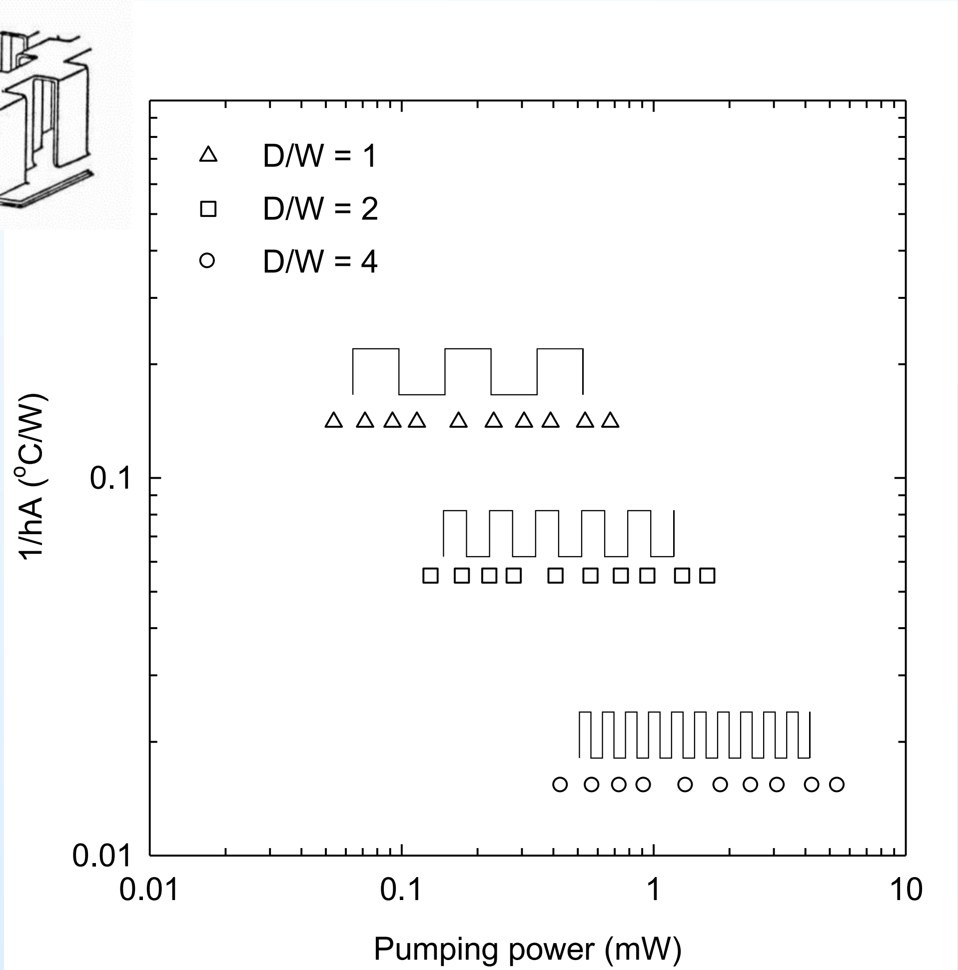


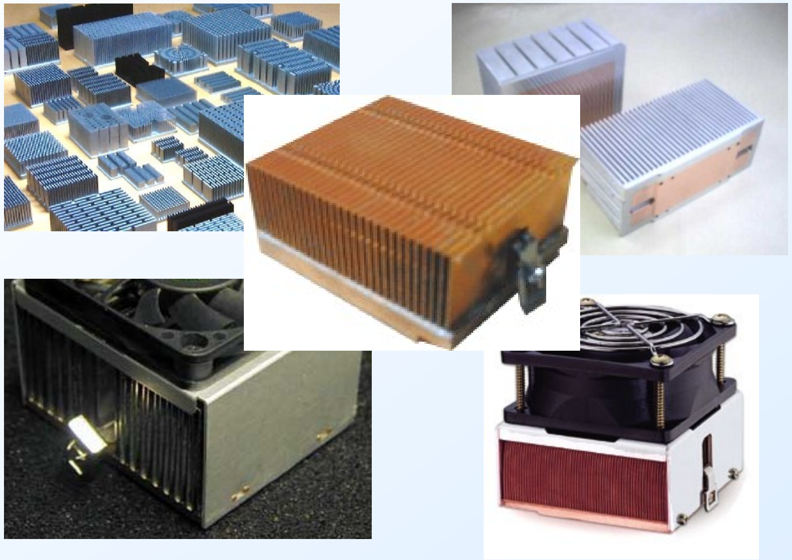


Example of Increased A

- Increase aspect ratio
  - Limit to manufacturing
  - Mal-distribution is likely
- Increase fin type







Increased heat transfer coeff.

- Air cool to liquid cool, single-phase to two-phase

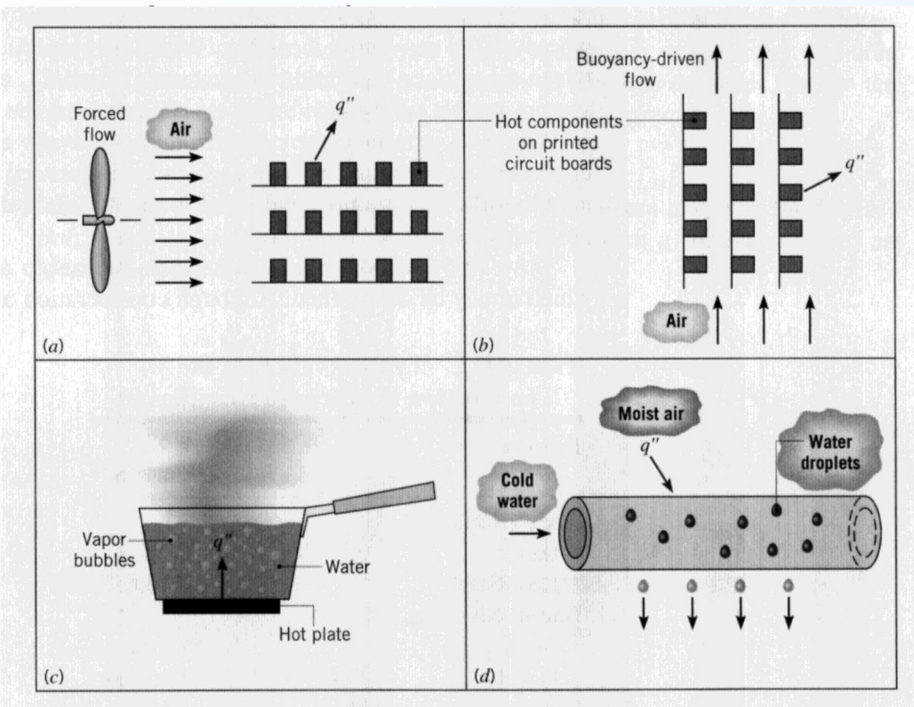
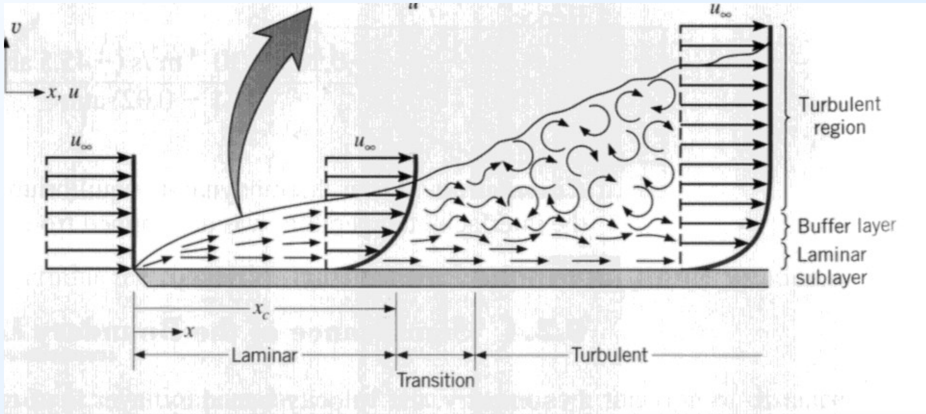


TABLE 1.1 Typical values of the convection heat transfer coefficient

Process	h $(W/m^2 \cdot K)$
Free convection	
Gases	2–25
Liquids	50–1000
Forced convection	
Gases	25–250
Liquids	100–20,000
Convection with phase change	
Boiling or condensation	2500–100,000

Single phase flow pattern

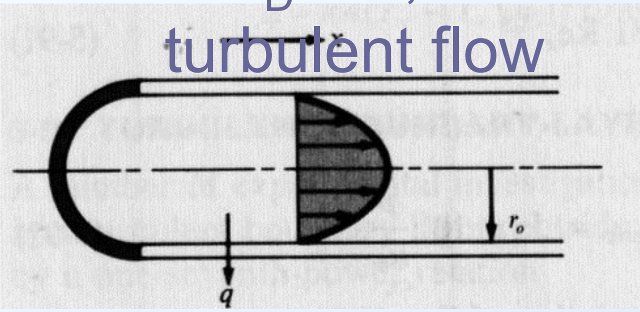
$$q = hA(T_s - T_\infty) = -kA\frac{dT}{dy}\Big|_{y=0}$$

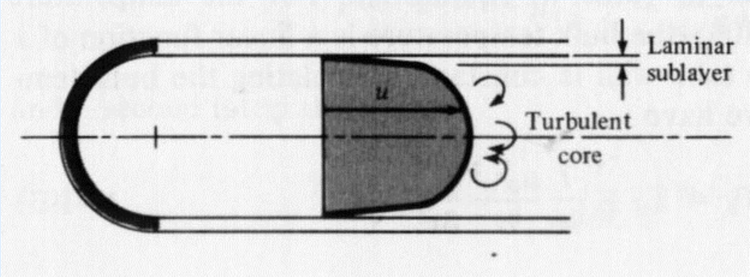


- $Re_D = \rho uD/\mu$

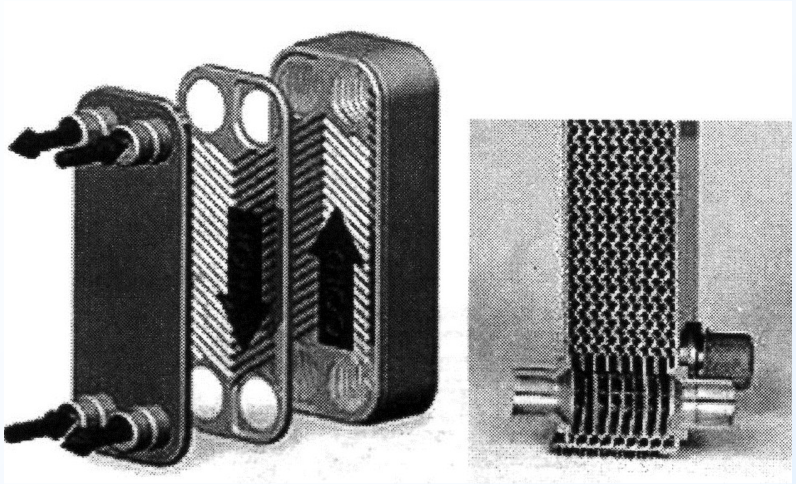
For smaller diameter tube (or micro tube) flow pattern is mostly operated at laminar flow

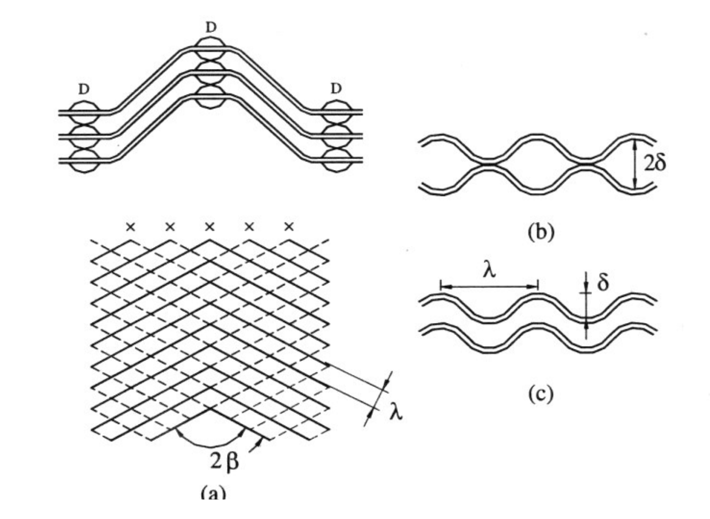
$Re_D < 2,300$ laminar flow $Re_D > 2,300$ turbulent flow

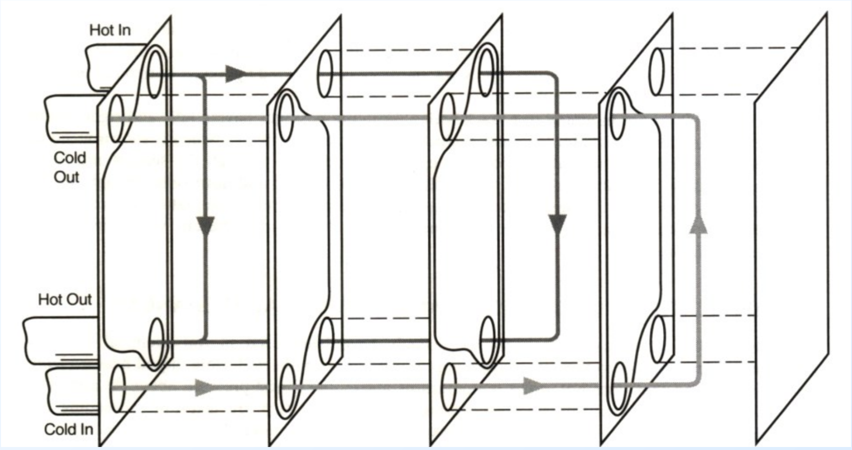


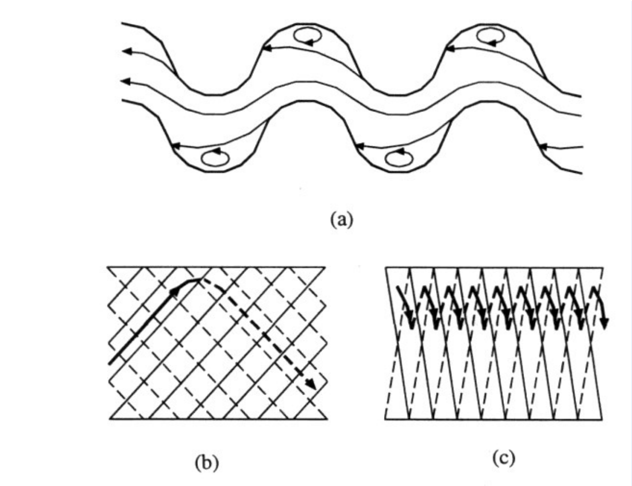


Apply the conventional Plate HX









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。

$$h = c_{p,a}\mathrm{Pr}^{-2/3} jG \tag{7-3}$$

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：

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

同時，推動流體所需要的消耗功率可表示如下：

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

因此，新的熱傳增強表面與參考表面的耗功比值為：

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

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：

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

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：

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

另外將式7-8代入式7-6可得到：

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

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

PEC的應用法則與目的

法則	幾何參數	固定參數				目的
		r	W	Q	ΔT	
FG-1a	N，L	☆			☆	↑Q
FG-1b	N，L	☆		☆		↓ΔT
FG-2a	N，L		☆		☆	↑Q
FG-2b	N，L		☆	☆		↓ΔT
FG-3	N，L			☆	☆	↓W
FN-1	N		☆	☆	☆	↓L
FN-2	N	☆		☆	☆	↓L
FN-3	N	☆		☆	☆	↓W
VG-1		☆	☆	☆	☆	↓NL
VG-2a	N，L	☆	☆		☆	↑Q
VG-2b	N，L	☆	☆	☆		↓ΔT
VG-3	N，L	☆		☆	☆	↓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

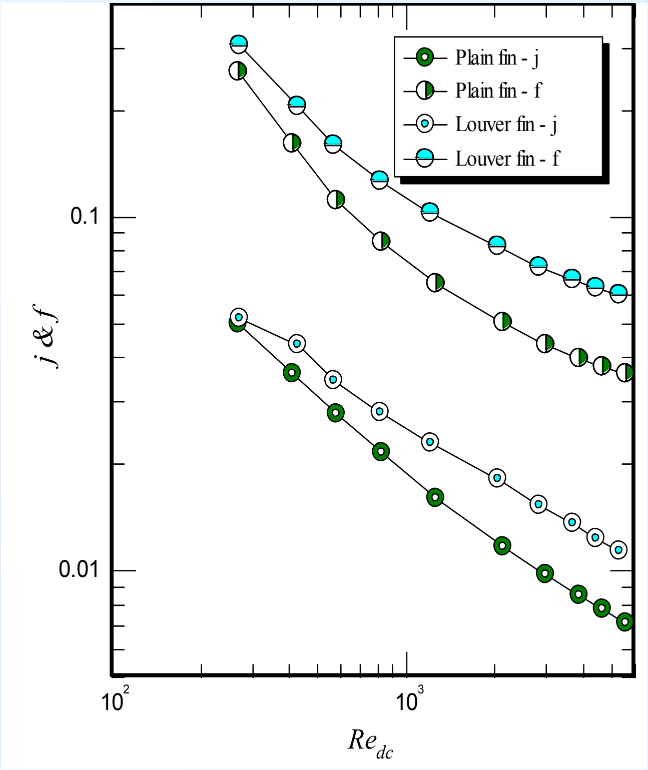
一平板式(plain)鰭管式熱交換器，原工作雷諾數為2000，若使用百葉窗式(louver)鰭片 來取代，若以VG-1法則來評估其可行性，試問：(1) 百葉窗式鰭片 工作的雷諾數為何？(2)在該操作條件下，若使用百葉窗式鰭片 熱交換器，則可節省多少熱交換器的面積？本例的平板與百葉窗熱交換器的排數與鰭片 間距均相同。平板與百葉窗鰭片 的*j*與*f*的特性如圖所示，而*j*與*f*的測試資料與回歸方程式如下：

$$j_{louver} = 0.0084715195 + \frac{18.858621}{\mathrm{Re}_{d_c}} - \frac{1920.6343}{\mathrm{Re}_{d_c}^2}$$

$$f_{louver} = 0.050337437 + \frac{59.224153}{\mathrm{Re}_{d_c}} + \frac{2727.0893}{\mathrm{Re}_{d_c}^2}$$

$$j_{plain} = 0.00482626 + \frac{14.4216}{\mathrm{Re}_{d_c}} - \frac{627.44798}{\mathrm{Re}_{d_c}^2}$$

$$f_{plain} = 0.02961772 + \frac{38.410523}{\mathrm{Re}_{d_c}} + \frac{6174.5523}{\mathrm{Re}_{d_c}^2}$$



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

由於 $\mathrm{Re}_{ref}=\mathrm{Re}_{plain}=2000$；

$$j_{ref}=j_{plain}=0.00482626+\frac{14.4216}{\mathrm{Re}_{d_c}}-\frac{627.44798}{\mathrm{Re}_{d_c}^2}$$

$$=0.00482626+\frac{14.4216}{2000}-\frac{627.44798}{2000^2}=0.0118802$$

$$f_{ref}=f_{plain}=0.02961772+\frac{38.410523}{\mathrm{Re}_{d_c}}+\frac{6174.5523}{\mathrm{Re}_{d_c}^2}$$

$$=0.02961772+\frac{38.410523}{2000}+\frac{6174.5523}{2000^2}=0.050367$$

$$\because \frac{\mathrm{Re}}{\mathrm{Re}_{ref}}=\left(\frac{j}{j_{ref}}\frac{f_{ref}}{f}\right)^{1/2} \Rightarrow \frac{\mathrm{Re}}{2000}=\left(\frac{j}{0.0118802}\frac{0.050367}{f}\right)^{1/2}$$

要解上述方程式則需要一些疊代工作，經過一番努力後，上述方程式的解為 $\mathrm{Re}_{dc}\approx 1919$。 此時的$j\approx 0.01777$而$f\approx 0.08194$。由於鰭片間距相同，所以$h\sim j$，在VG-1法則要求$h_{ref}A_{ref}=hA$

$$\Rightarrow \frac{A_{ref}}{A}=\frac{h}{h_{ref}}\approx\frac{j}{j_{ref}}=\frac{0.01777}{0.0118802}=1.496$$

換句話說，使用百葉窗鰭片可節省約1/3(≈ 1 – 1/1.496)的面積。不過筆者必須在這裡特別提醒讀者，本計算例是在兩個假設的前提下所得，即 (1) 熱交換器的阻抗全部在空氣側；(2)假設平板與百葉窗的鰭片效率是相同的。因此讀者不可以指望換了一個百葉窗鰭片就可節省1/3的熱交換器面積。

FG-3法則

在FG-3的需求中，熱交換器的面積大小是固定的，而Q/Q_{ref}與$\Delta T_m/\Delta T_{m,ref}$為固定，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運轉成本，即$W/W_{ref}<1$，這個設計在長期運轉上是很重要的。由$Q/Q_{ref}=1$，$hA/h_{ref}A_{ref}=1$與$A/A_{ref}=1$，則式7-4可簡化如下：

$$\frac{hA}{h_{ref}A_{ref}}=\frac{j}{j_{ref}}\frac{A}{A_{ref}}\frac{G}{G_{ref}}$$

$$\Rightarrow 1=\frac{j}{j_{ref}}\frac{A}{A_{ref}}\frac{G}{G_{ref}}$$

$$\Rightarrow 1=\frac{j}{j_{ref}}\frac{G}{G_{ref}}$$

$$\Rightarrow \frac{G}{G_{ref}}=\frac{j_{ref}}{j} \qquad (7\text{-}11)$$

所以式7-6可簡化如下：

$$\frac{W}{W_{ref}}=\frac{f}{f_{ref}}\frac{A}{A_{ref}}\left(\frac{G}{G_{ref}}\right)^3=\frac{f}{f_{ref}}\left(\frac{G}{G_{ref}}\right)^3=\frac{f}{f_{ref}}\left(\frac{j_{ref}}{j}\right)^3 \qquad (7\text{-}12)$$

同上例，平板式(plain)鰭管式熱交換器工作雷諾數為2000，若使用百葉窗式(louver)鰭片來取代，且熱交換器大小一樣，試以FG-3法則來評估運轉動力是否可以降低？

由式7-11，$\dfrac{G}{G_{ref}} = \dfrac{j_{ref}}{j} \Rightarrow \dfrac{\mathrm{Re}_{d_c}}{\mathrm{Re}_{dc,ref}} = \dfrac{j_{ref}}{j}$

由於 $\mathrm{Re}_{ref} = \mathrm{Re}_{plain} = 2000$；

同例7-2-1，

$$j_{ref} = j_{plain} = 0.0118802\text{，} \quad f_{ref} = f_{plain} = 0.050367$$

$$\frac{\mathrm{Re}_{d_c}}{\mathrm{Re}_{dc,ref}} = \frac{j_{ref}}{j}$$

$$\Rightarrow \frac{\mathrm{Re}_{d_c}}{2000} = \frac{0.0118802}{0.0084715195 + \dfrac{18.858621}{\mathrm{Re}_{d_c}} - \dfrac{1920.6343}{\mathrm{Re}_{d_c}^2}}$$

$$\Rightarrow \mathrm{Re}_{d_c} \approx 845$$

$$\therefore j \approx 0.0281\text{，}f \approx 0.1242$$

$$\frac{W}{W_{ref}} = \frac{f}{f_{ref}}\left(\frac{j_{ref}}{j}\right)^3 = \frac{0.1242}{0.050367}\left(\frac{0.0118802}{0.0281}\right)^3 = 0.1864$$

此一結果告訴您使用FG-3法則設計，將可節省8成的運轉成本，不過這個例子與例7-2-1的假設均相同；讀者在使用上仍需注意。

通用熱交換器性能評價方法

$$\frac{1}{(UA)_{ref}} = \frac{1}{\eta_{o,ref} h_{o,ref} A_{o,ref}} + \frac{1}{h_i A_i} + \frac{\Delta X}{k A_{w,m}}$$

$$= \frac{1}{\eta_{o,ref} h_{o,ref} A_{o,ref}} \left[1 + \frac{\eta_{o,ref} h_{o,ref} A_{o,ref}}{h_i A_i} + \frac{\eta_{o,ref} h_{o,ref} A_{o,ref} \Delta X}{k A_{w,m}} \right] \qquad (7\text{-}13)$$

$$= \frac{1}{\eta_{o,ref} h_{o,ref} A_{o,ref}} \left[1 + \beta_{ref} \right]$$

其中

$$\beta_{ref} = \frac{\eta_{o,ref} h_{o,ref} A_{o,ref}}{h_i A_i} + \frac{\eta_{o,ref} h_{o,ref} A_{o,ref} \Delta X}{k A_{w,m}} \qquad (7\text{-}14)$$

$$\frac{1}{(UA)}=\frac{1}{\eta_o h_o A_o}+\frac{1}{h_i A_i}+\frac{\Delta X}{kA_{w,m}}=\frac{1}{\eta_o h_o A_o}\left[1+\frac{\eta_o h_o A_o}{h_i A_i}+\frac{\eta_o h_o A_o \Delta X}{kA_{w,m}}\right]$$

$$=\frac{1}{\eta_o h_o A_o}\left[1+\frac{\eta_o h_o A_o}{\eta_{o,ref} h_{o,ref} A_{o,ref}}\left(\frac{\eta_{o,ref} h_{o,ref} A_{o,ref}}{h_i A_i}+\frac{\eta_{o,ref} h_{o,ref} A_{o,ref}\Delta X}{kA_{w,m}}\right)\right]$$

$$=\frac{1}{\eta_o h_o A_o}\left[1+\frac{\eta_o h_o A_o}{\eta_{o,ref} h_{o,ref} A_{o,ref}}\beta_{ref}\right] \tag{7-15}$$

將式7-13除上式7-15可得到

$$\frac{UA}{(UA)_{ref}}=\frac{\eta_o h_o A_o}{\eta_{o,ref} h_{o,ref} A_{o,ref}}\left(\frac{1+\beta_{ref}}{1+\frac{\eta_o h_o A_o}{\eta_{o,ref} h_{o,ref} A_{o,ref}}\beta_{ref}}\right) \tag{7-16}$$

$$=\left(\frac{\eta_o h_o}{\eta_{o,ref} h_{o,ref}}\right)\left(\frac{A_o}{A_{o,ref}}\right)\left(\frac{1+\beta_{ref}}{1+\left(\frac{\eta_o h_o}{\eta_{o,ref} h_{o,ref}}\right)\left(\frac{A_o}{A_{o,ref}}\right)\beta_{ref}}\right)$$

在壓降部份，式7-6仍可適用：

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

以VG-1法則而言，設計的過程可簡述如下：

(1) 熱交換器的熱流特性，包括原參考熱交換器與「中意」的熱傳增強表面必須事先給定；另外，由於VG-1要求 $\dfrac{Q}{Q_{ref}}=\dfrac{W}{W_{ref}}=\dfrac{\Delta T_m}{\Delta T_{m,ref}}$

(2) $$\therefore\ \frac{UA}{(UA)_{ref}}=1=\left(\frac{\eta_o h_o}{\eta_{o,ref}h_{o,ref}}\right)\left(\frac{A_o}{A_{o,ref}}\right)\left(\frac{1+\beta_{ref}}{1+\left(\frac{\eta_o h_o}{\eta_{o,ref}h_{o,ref}}\right)\left(\frac{A_o}{A_{o,ref}}\right)\beta_{ref}}\right)$$

(3) $$\frac{W}{W_{ref}}=1=\frac{f}{f_{ref}}\frac{A}{A_{ref}}\left(\frac{G}{G_{ref}}\right)^3$$

(4) β_{ref} 為原參考條件故可算出，即：

$$\beta_{ref}=\frac{\eta_{o,ref}h_{o,ref}A_{o,ref}}{h_iA_i}+\frac{\eta_{o,ref}h_{o,ref}A_{o,ref}}{kA_{w,m}}$$

(5) 猜新增強表面的一個合適的工作 G(或 Re)值，因此 $(\eta_o h_o)/(\eta_{o,ref}h_{o,ref})$，$G/G_{ref}$，$f/f_{ref}$ 可一併算出，再由

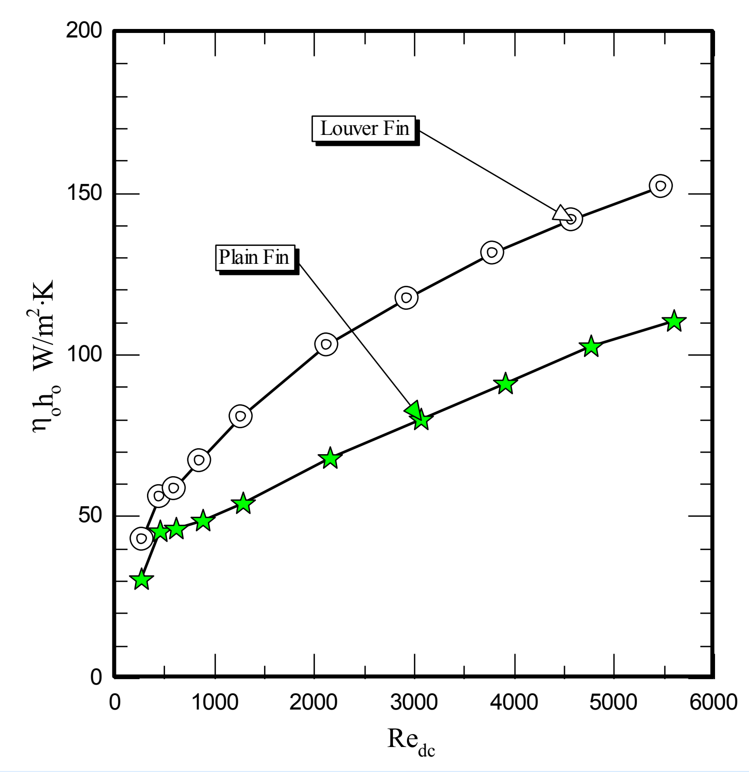
$$\frac{W}{W_{ref}}=1=\frac{f}{f_{ref}}\frac{A}{A_{ref}}\left(\frac{G}{G_{ref}}\right)^3$$，算出 A/A_{ref}

檢查是否 $\left(\frac{\eta_o h_o}{\eta_{o,ref}h_{o,ref}}\right)\left(\frac{A_o}{A_{o,ref}}\right)\left(\frac{1+\beta_{ref}}{1+\left(\frac{\eta_o h_o}{\eta_{o,ref}h_{o,ref}}\right)\left(\frac{A_o}{A_{o,ref}}\right)\beta_{ref}}\right)=1$，若是，代表猜測的 G 值正確；若否，則需重複5~6步驟。請特別注意這個過程未必有解，若無法取得解答，則代表VG-1法則無法滿足此一熱傳增強表面的選擇。

同上例，平板式(plain)鰭管式熱交換器，原工作雷諾數為5000，若使用百葉窗式鰭片來取代，以VG-1法則來評估其可行性，試問：(1) 百葉窗式鰭片工作的雷諾數為何？(2)在該操作條件下，若使用百葉窗式鰭片熱交換器，則可節省多少熱交換器的面積？本例的平板與百葉窗熱交換器的排數與鰭片間距均相同，而管內熱傳係數$h_i \approx 4000$ W/m²·K，$A_{o,ref}/A_i \approx 12.7$。平板與百葉窗鰭片的$j$與$f$的特性如前圖所示而$j$與$f$的方程式與上例同，$\eta_o h_o$的測試資料見下圖，相關的熱流方程式經回歸後如下：

$$\eta_{o,louver} h_{o,louver} = 2.3948796 + 3.4730182 \mathrm{Re}_{d_c}^{0.43766997}$$

$$\eta_{o,plain} h_{o,plain} = 31.64771 + 0.046902626 \mathrm{Re}_{d_c}^{0.86292333}$$



β_{ref}為原參考條件故可算出，並假設管壁阻抗甚小可忽略，即：

$$\beta_{ref}=\frac{\eta_{o,ref}h_{o,ref}A_{o,ref}}{h_iA_i}+\frac{\eta_{o,ref}h_{o,ref}A_{o,ref}}{kA_m}\approx\frac{\eta_{o,ref}h_{o,ref}A_{o,ref}}{h_iA_i}+\frac{\eta_{o,ref}h_{o,ref}}{h_i}\frac{A_{o,ref}}{A_i}$$

$$=\frac{31.64771+0.046902626\times5000^{0.86292333}}{4000}\times12.7=\frac{104.61}{4000}\times12.7=0.3321$$

接下來，猜新增強表面的一個合適的工作Re_{dc}值，再進行疊代，例如

Re_{dc} = 4500

$\Rightarrow(\eta_o h_o)/(\eta_{o,ref}h_{o,ref})$ = 1.3412

G/G_{ref} = 0.9

f/f_{ref} = 1.694

由 $\dfrac{W}{W_{ref}}=1=\dfrac{f}{f_{ref}}\dfrac{A}{A_{ref}}\left(\dfrac{G}{G_{ref}}\right)^3$，可算出 A/A_{ref} = 0.8094，再檢查

$$\left(\frac{\eta_o h_o}{\eta_{o,ref}h_{o,ref}}\right)\left(\frac{A_o}{A_{o,ref}}\right)\left(\frac{1+\beta_{ref}}{1+\left(\dfrac{\eta_o h_o}{\eta_{o,ref}h_{o,ref}}\right)\left(\dfrac{A_o}{A_{o,ref}}\right)\beta_{ref}}\right)$$

$$=1.3412\times0.8094\times\left(\frac{1+0.3321}{1+1.3412\times0.8094\times0.3321}\right)$$

$$=1.063\neq1$$

如此幾次進行疊代，可得到最後結果如下：

Re_{dc} = 4659

A_o/A_{ref} = 0.7346

換句話說，在此操作條件下約可減少26.54%的面積

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：

(1) 直接評價*j*與*f*，作為判別。

(2) 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。

(3) 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)。

(4) 其他評價方法(例如Cowell, 1990的方法)。

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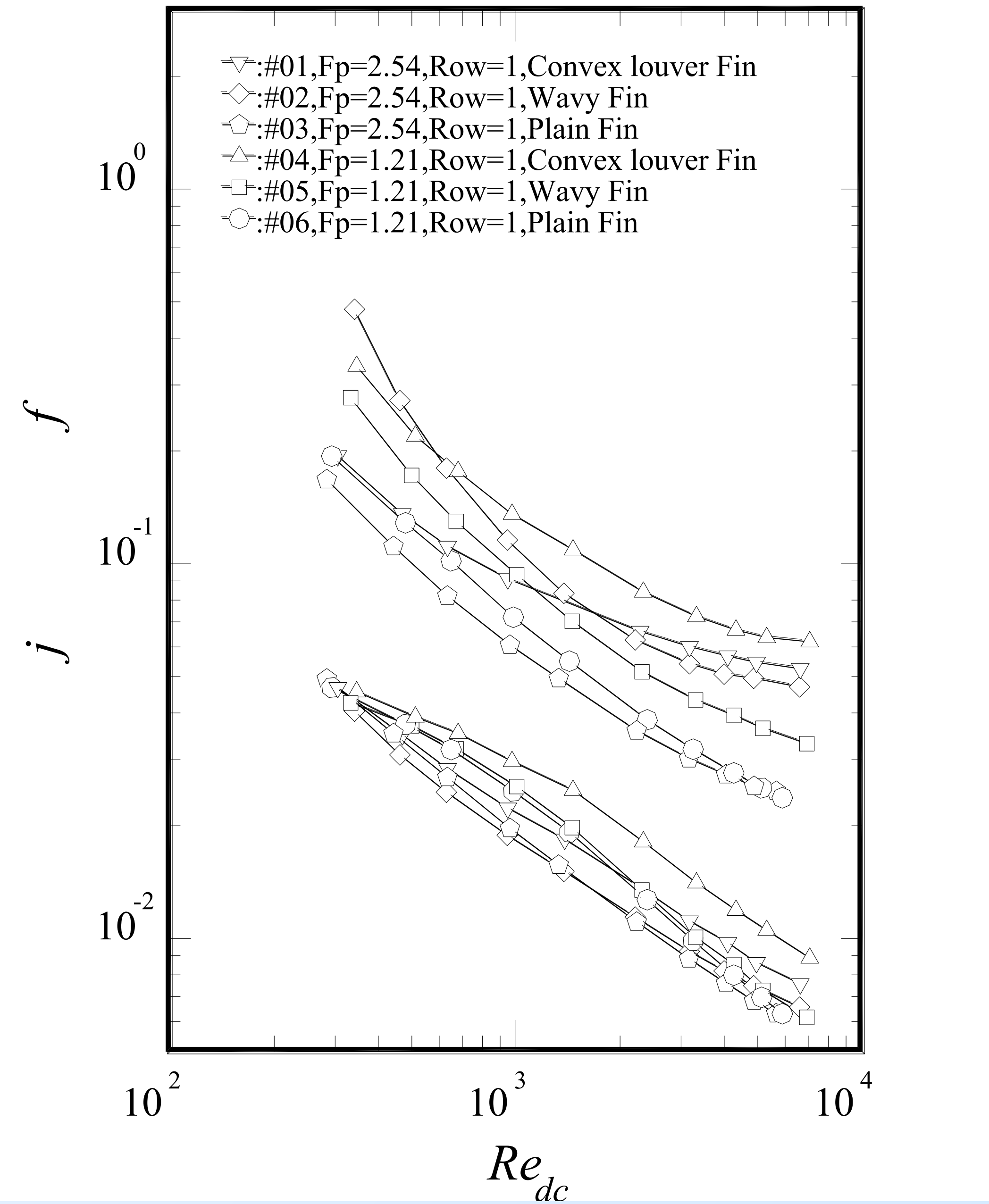
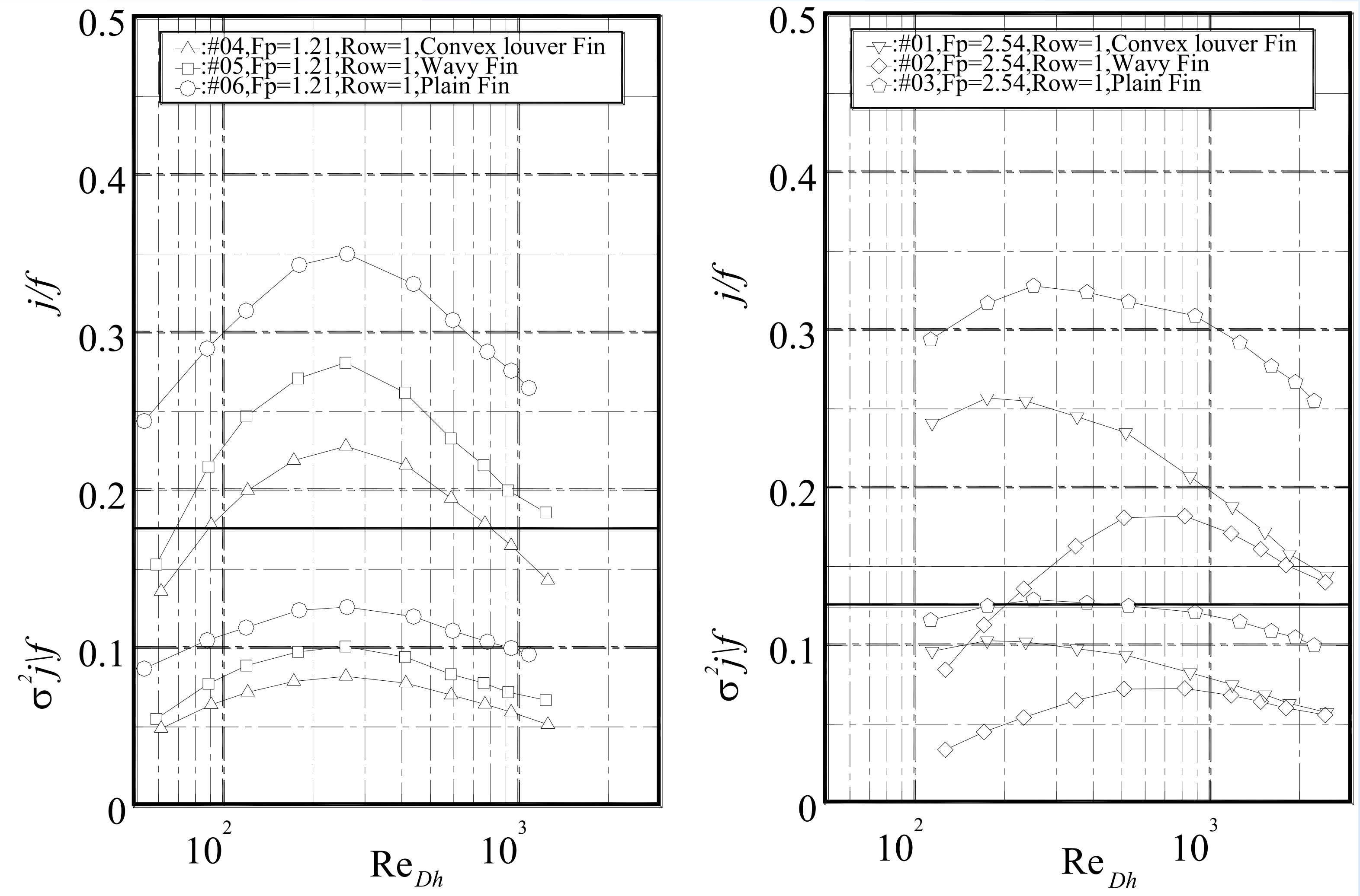


圖7-7　複合式百葉窗片(convex-louver)、波浪鰭片(wavy)與平板型鰭片(plain)的性能直接評價

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

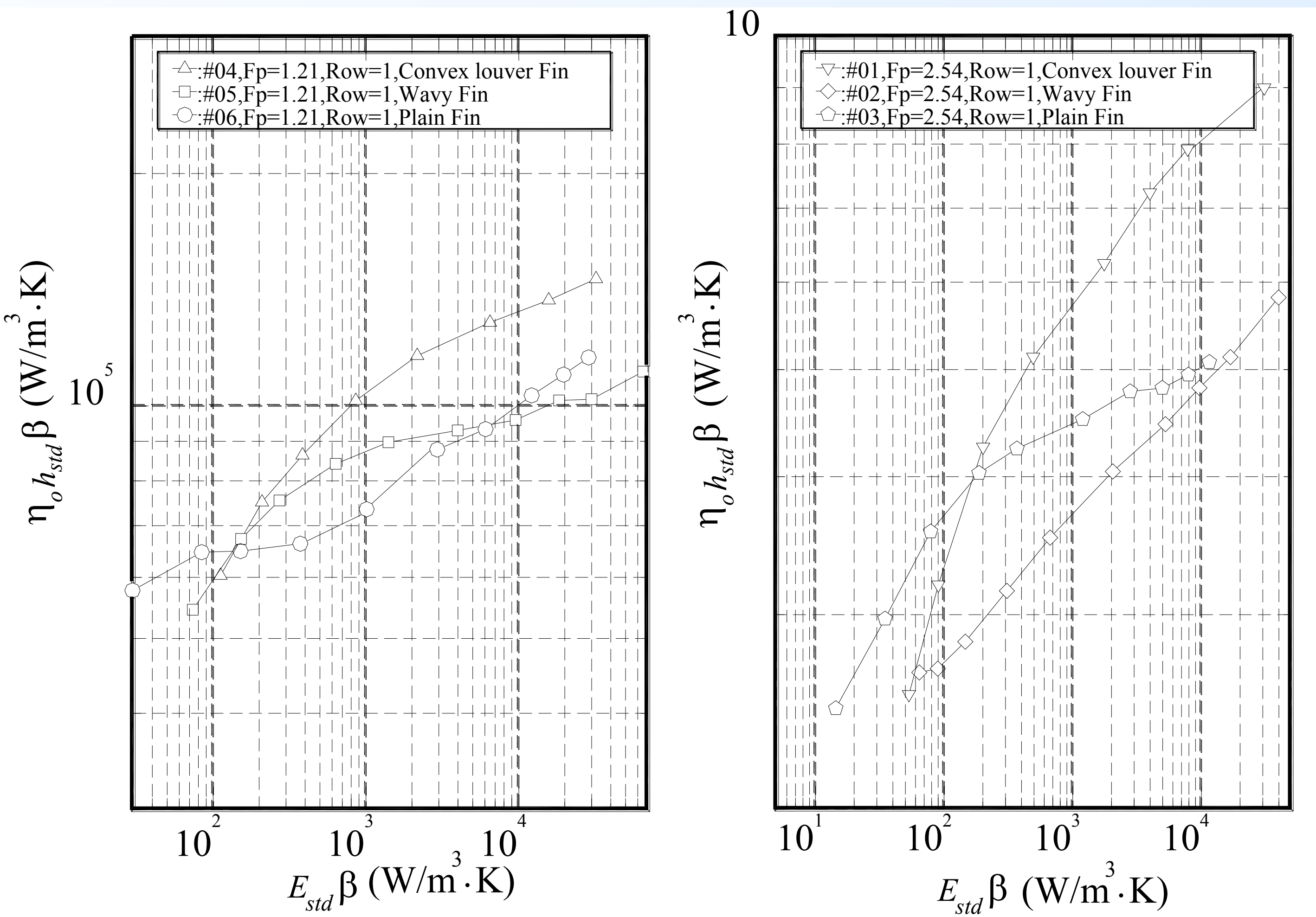


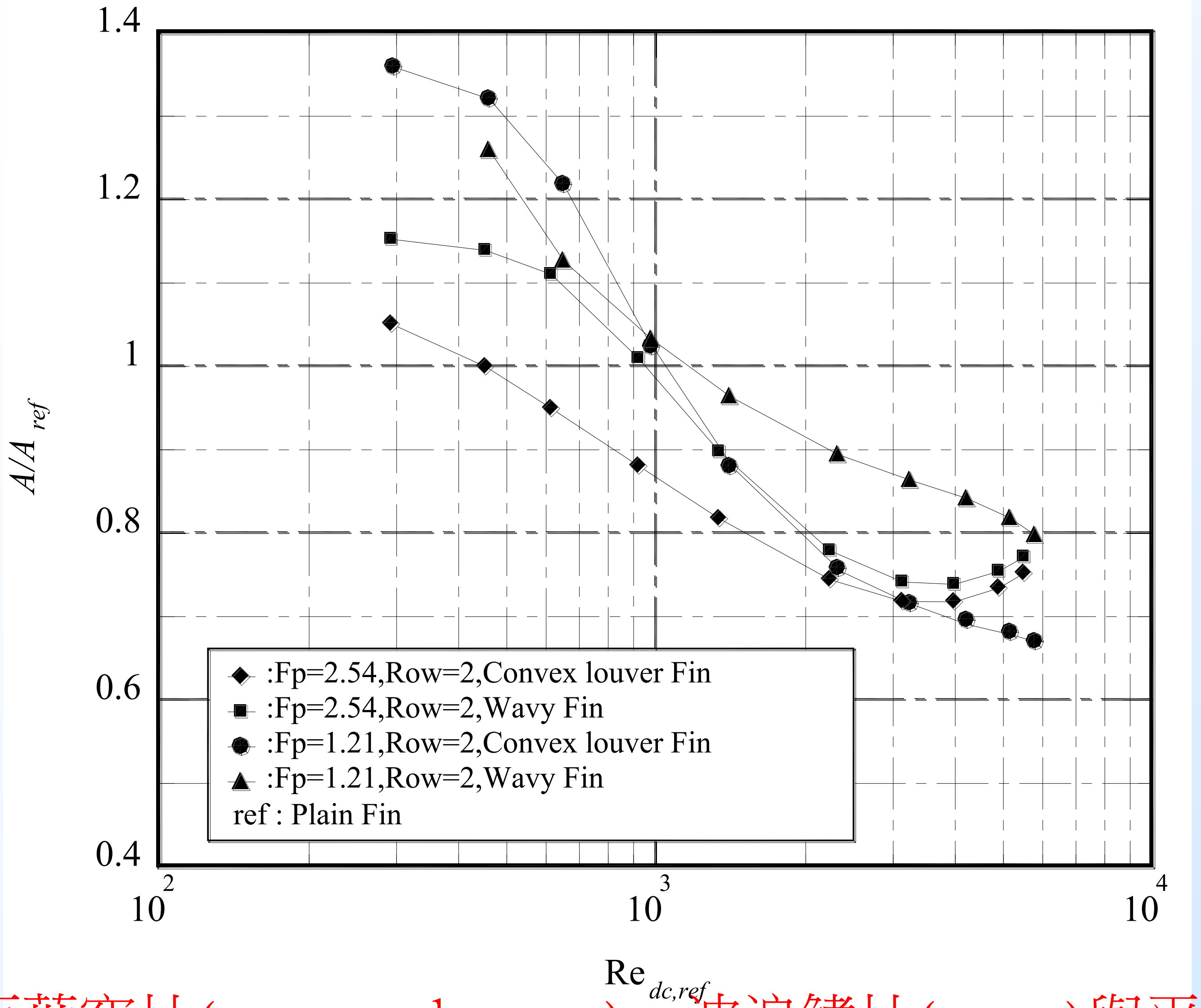
另外一種常用的評價方法為「體積優先」評價法(volume goodness factor comparison, Kays and London ,1950)，採用的評價參數為：

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

$$E_{std}\beta = \frac{\mu^3}{2g_c\rho^2}\frac{4\sigma}{D_h^4}f\,\mathrm{Re}^3 \tag{7-18}$$

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

結語

熱交換器性能的評價方法相當多，本章節主要是以 Webb教授的PEC法為主軸，讀者可針對使用上的設計需求使用合理的評價方法，而取得定量的數據，對工程師而言，個人極力推崇使用這種方法，過去廣為人用的j/f或$h/\Delta P$法，建議少用以免誤導。

錢是否花在刀口上？

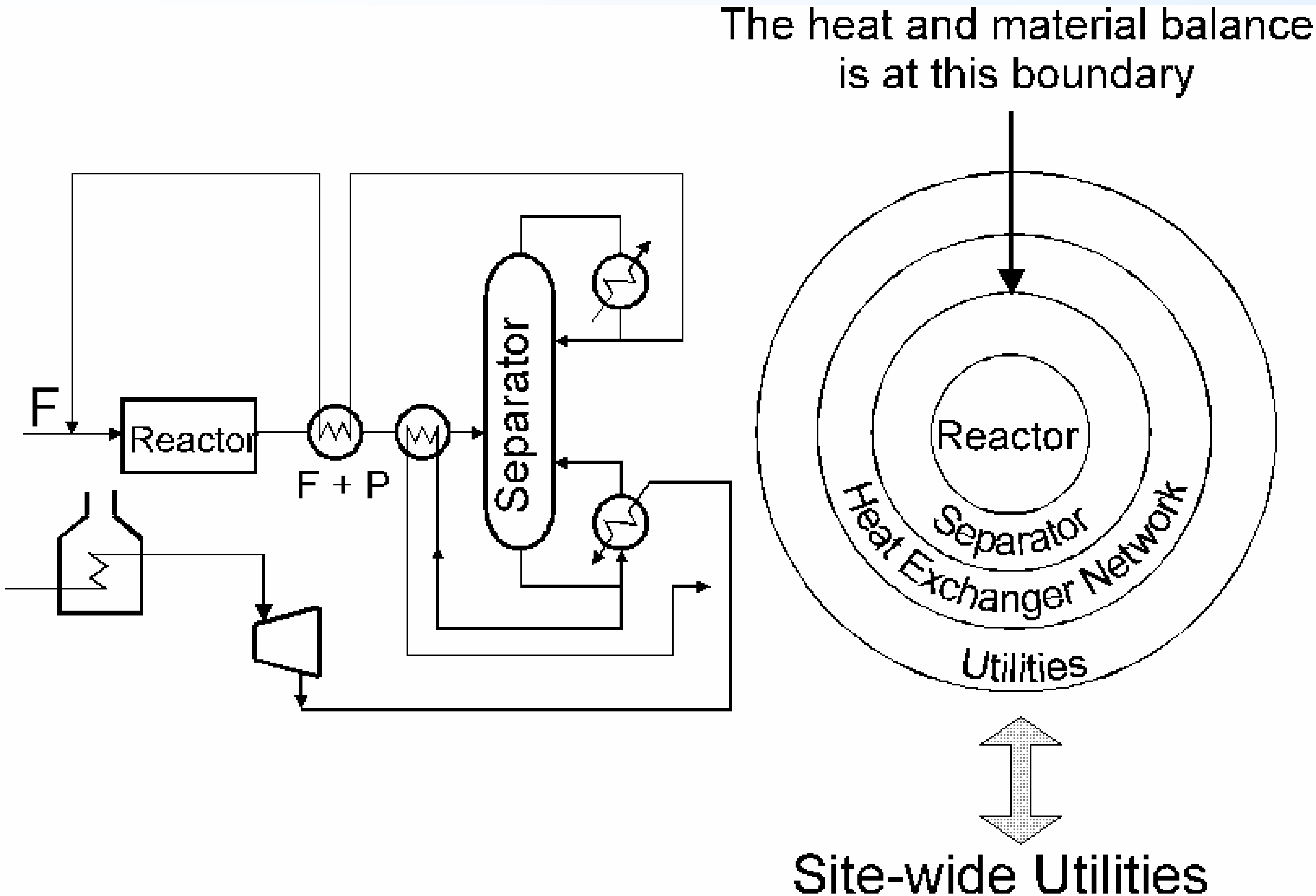
- 原則

  → 解決最大熱阻

製程整合節能技術

- 在實際應用上，不管是一般化工廠、石化工廠、食品製造工廠、或是機械工廠；通常這些工廠中都有相當多的熱交換器，這是因為熱交換器是最常見的能源轉換單元，在這麼多的熱交換器中，免不了有一些廢熱與廢冷，如果將這些廢熱與廢冷直接排放掉，不僅浪費寶貴的能源，也可能污染環境；因此回收製程系統中的這些廢熱廢冷就是本部分的主題

- Pinch Technology provides a systematic methodology for energy saving in processes and total sites. The methodology is based on thermodynamic principles. The role of Pinch Technology in the overall process design. The process design hierarchy can be represented by the “onion diagram” as shown below. The design of a process starts with the reactors (in the “core” of the onion). Once feeds, products, recycle concentrations and flow rates are known, the separators (the second layer of the onion) can be designed.


The heat and material balance is at this boundary
F
Reactor
F + P
Separator
Reactor
Separator
Heat Exchanger Network
Utilities
Site-wide Utilities

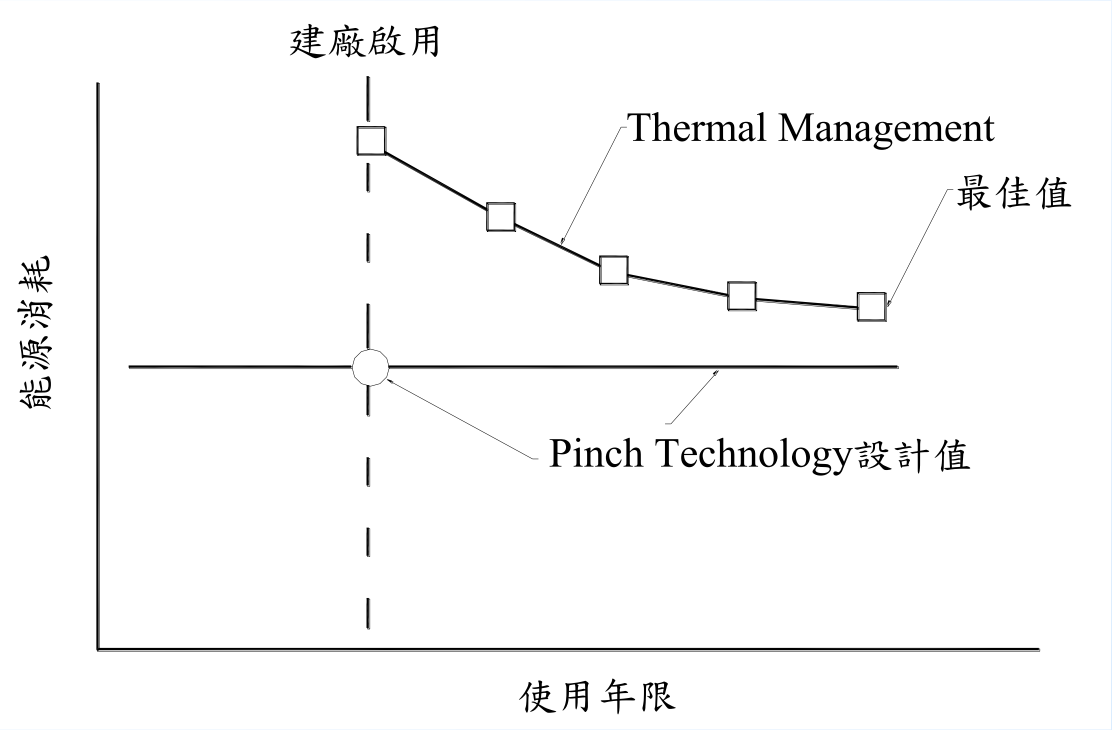
- A Pinch Analysis **starts with the heat and material balance for the process**. Using Pinch Technology, it is possible to identify appropriate changes in the core process conditions that can have an impact on energy savings (onion layers one and two). **After the heat and material balance is established, targets for energy saving can be set prior to the design of the heat exchanger network**. The Pinch Design Method ensures that these targets are achieved during the network design. Targets can also be set for the utility loads at various levels (e.g. steam and refrigeration levels).

- **The basic process heat and material balance is now in place, and the heat exchanger network (the third layer) can be designed. The remaining heating and cooling duties are handled by the utility system (the fourth layer).** The process utility system may be a part of a centralized site-wide utility system.

製程整合系統化的設計流程

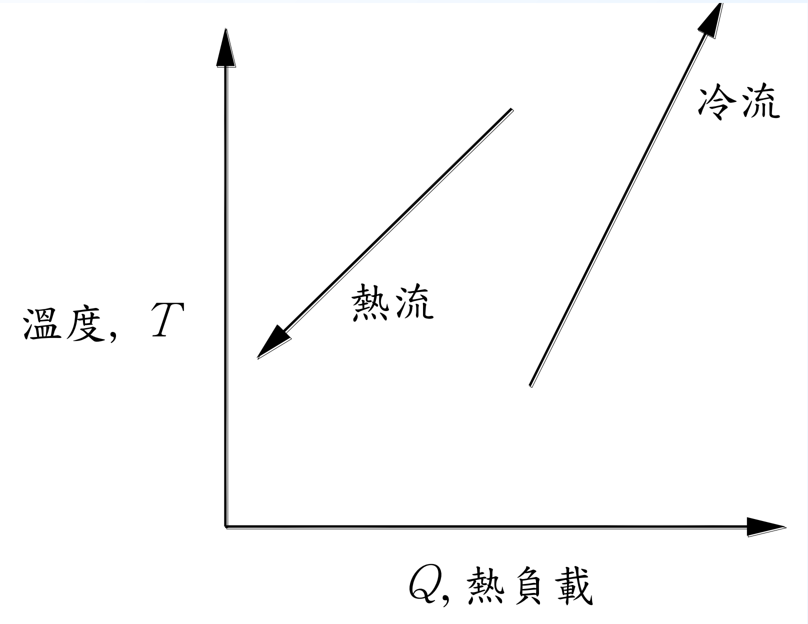
(1)收集製程資料。
(2)進行能量與質量的平衡。
(3)從這些資料中取出Pinch Technology所需要的資料。
(4)由這些資料中，選擇所需要的最小溫差。
(5)算出能源設計的標的與Pinch點。
(6)檢查當製程改變時，此一設計是否得當，如果需要，可以再次進行調整，重新計算。
(7)設計符合能源標的的熱交器網路。
(8)根據實際節能的考量，將此一熱交換器網路適度的調整。
(9)進行經濟分析，根據分析結果來判斷最佳的選擇。

Thermal Management vs. Pinch Technology



Thermal management 是藉由建廠後長期的監控與查核，再逐步地去調整與改善耗能與沒效率的設計(如圖所示)，透過系統監控與能源查核，沒有效率的能源耗費得以逐步改善

熱流、冷流示意圖
Hot & Cold Streams



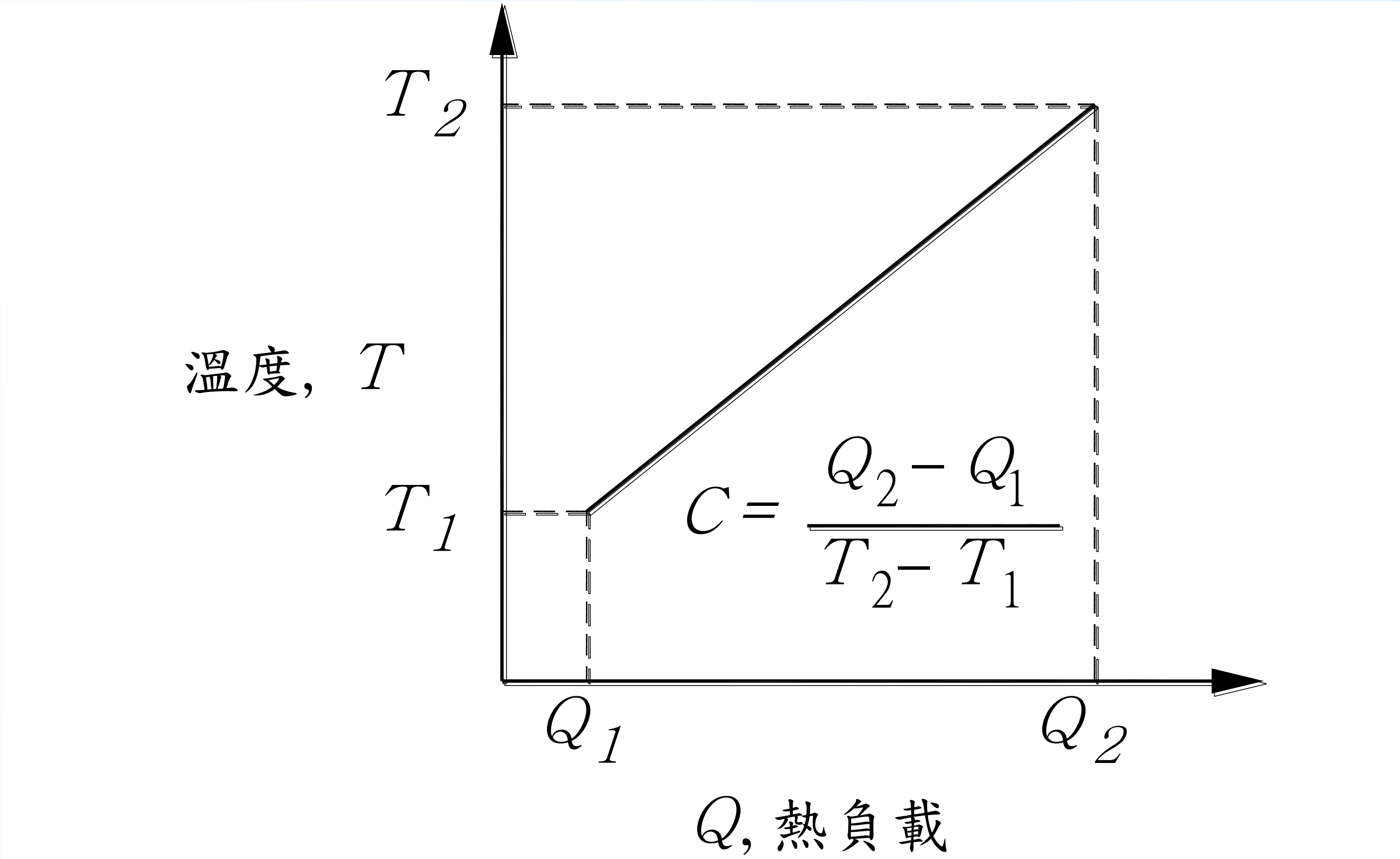
Stream係指任何物流(Material flow)在製程中經過熱焓量改變的過程

Hot stream(熱流)為製程中，熱焓量下降的物流。而Cold stream (冷流)為製程中，熱焓量上升的物流

熱流與冷流 (Conti..)

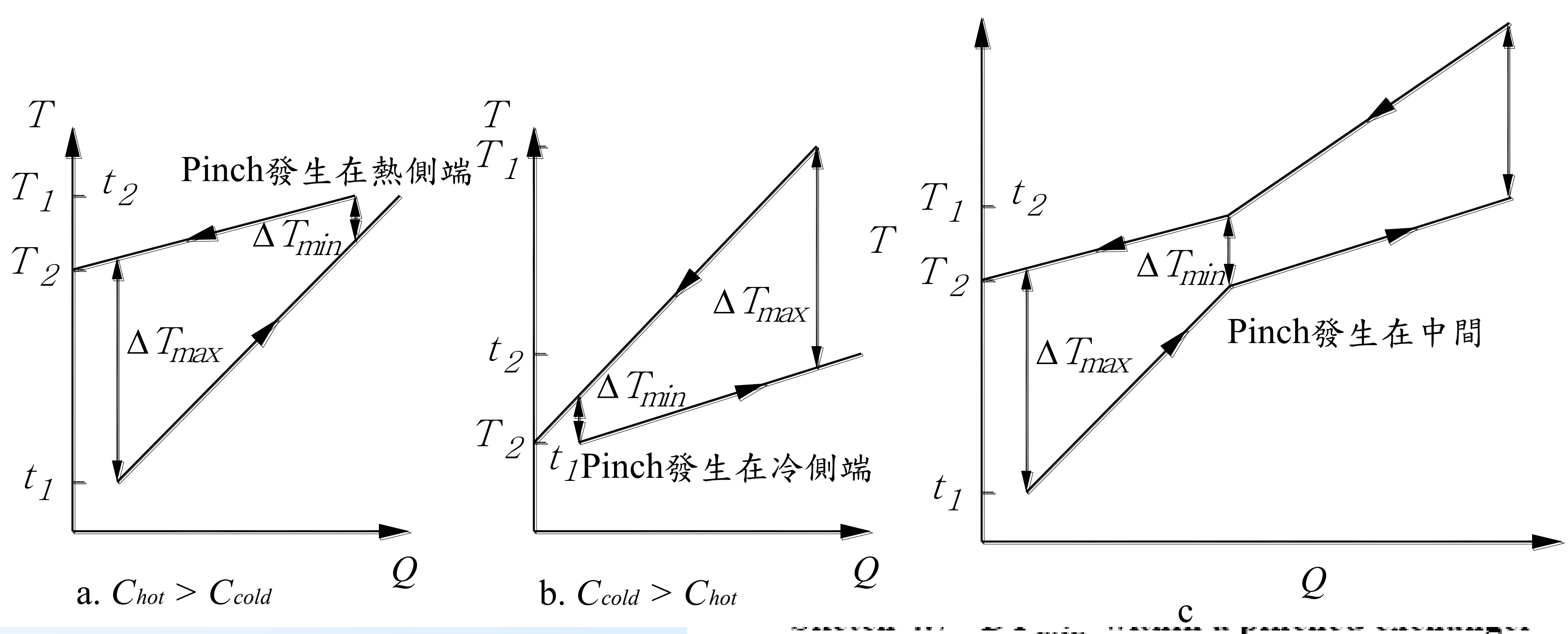
- Composite curve for hot & cold streams
- **Pinch usually occurs at one point between the hot and cold stream. This one is normally referred as heat recovery pinch.**
- How to decide the pinch?
  - Normally larger ΔT_{min} reduces capital cost but gives rise to higher running cost, and vice versa.
  - **A good initialization of the heat exchanger network design is to assume no individual heat exchanger has a temperature difference smaller than ΔT_{min} between composite curve.**

C與熱負載及溫差間的關係



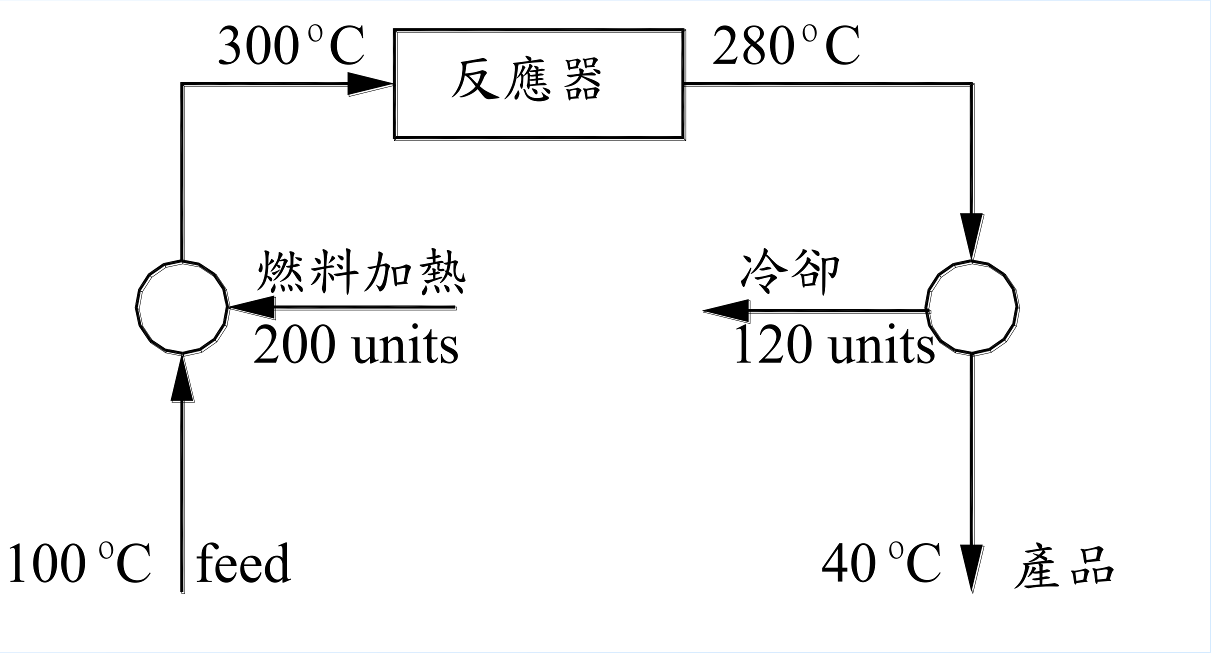
Pinch

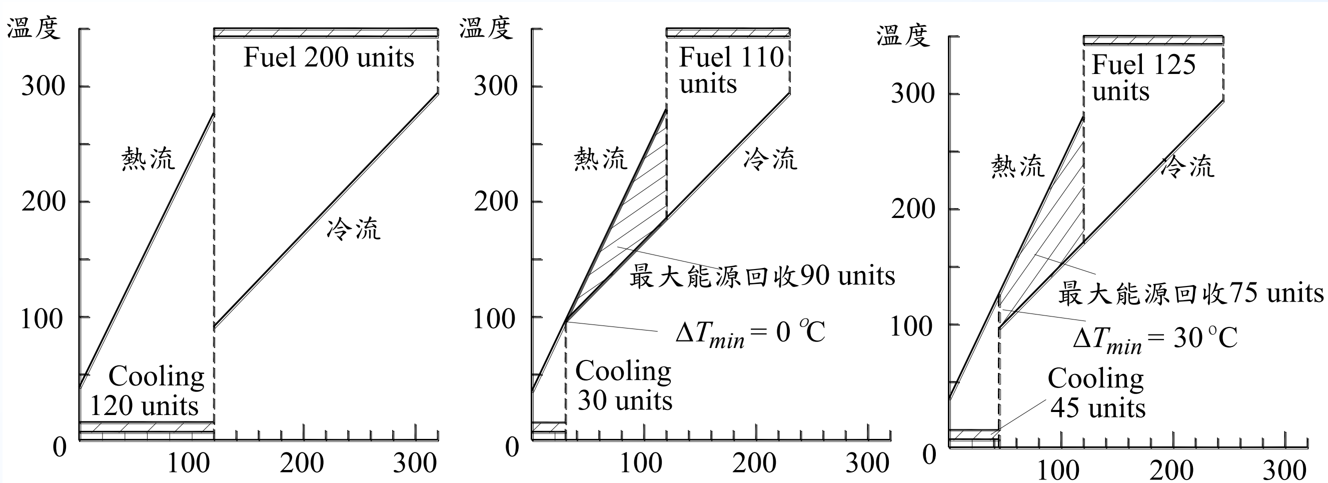
- 所謂Pinch(狹點)，就是熱流與冷流間於溫度-負載曲線(*T-Q* curve)中的最小溫度差，

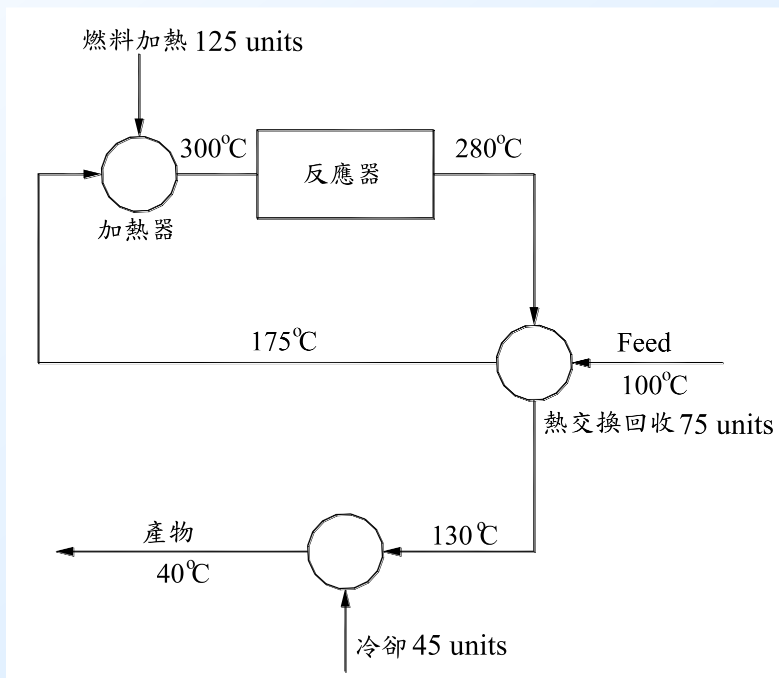


簡單案例

某一原料的進溫為100°C，經過外加的燃料加熱預熱到300°C然後置入反應器反應後轉變為半成品(半成品的溫度為280°C)隨後必須將半成品降溫成最後的40°C成品。假設原料的 c_p 為半成品與成品的一半；顯然，整個過程中需要外加燃料的加熱與冷卻的降溫，如果不考慮熱回收的問題，則原料從100°C加熱到300°C需要200個單位的燃料，而半成品從280°C降溫到40°C需要120個單位的冷卻量




溫度
300
200
100
0
Fuel 200 units
熱流
冷流
Cooling
120 units
100
200
300
Fuel 110
units
熱流
冷流
最大能源回收90 units
ΔTmin = 0 ℃
Cooling
30 units
Fuel 125
units
熱流
冷流
最大能源回收75 units
ΔTmin = 30 °C
Cooling
45 units


燃料加熱 125 units
300°C
反應器
280°C
加熱器
175℃
Feed
100°C
熱交換回收 75 units
產物
40℃
130℃
冷卻 45 units

Two methods

- Graph method
- Problem table

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複合負載曲線計算步驟

(1) 首先將冷流與熱流的資料分開分別處理。

(2) 以熱流為例，列出各個stream的資料，本例中的c_p 為定值，但在許多實際應用中，c_p可能變化很大，此時要將各個溫度區間的 c_p 變化列入考慮，最簡單的方式可取該熱流或冷流的進出口平均值，但特別提醒讀者此一做法僅適用單相的流體，如果其中牽涉到相變化，必須考慮整體熱焓的變化。

(3) 計算每一個溫度區間的熱容量流率 $C = \dot{m}c_p$ 。

(4) 將熱流的溫度由小到大，重新排序。

(5) 將各個區間的各個熱容量加總，$C_{sum} = \sum C_i$，請特別注意一個原則，各個熱流的熱容量流率的貢獻僅在自己的溫度區間內，例如熱流C的貢獻僅在120度到50度間，在其他區間的貢獻度為0。

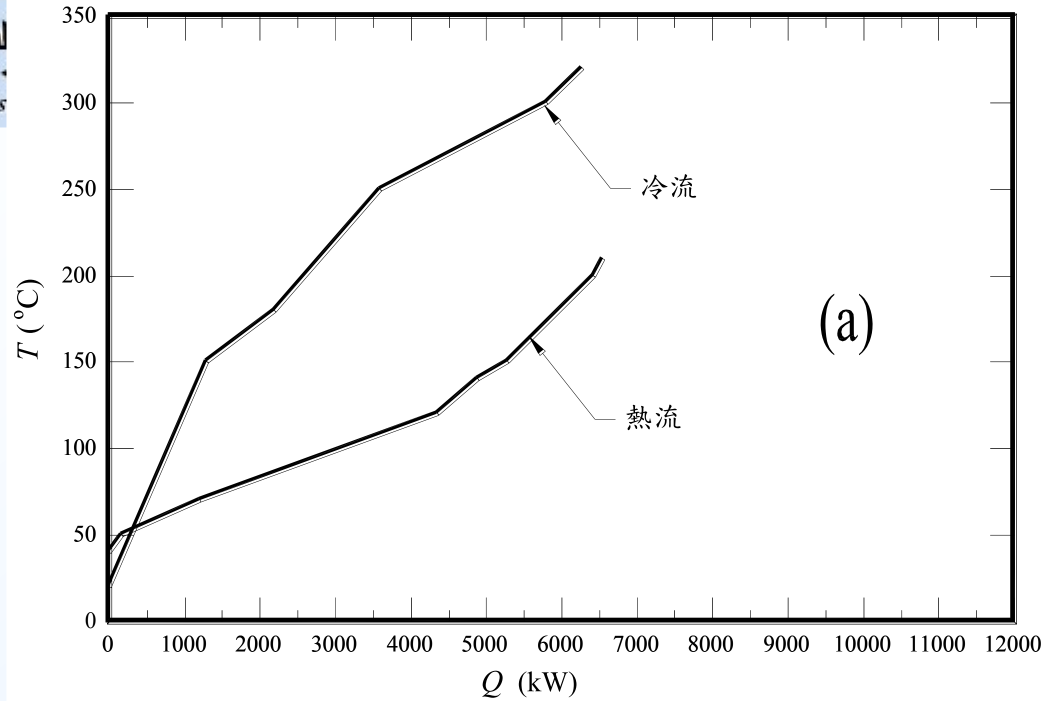
(6) 計算各個區段間的傳熱量，並累計期總熱傳量 $Q_i = Q_{i\text{-}1} + C_{sum}\Delta T$。

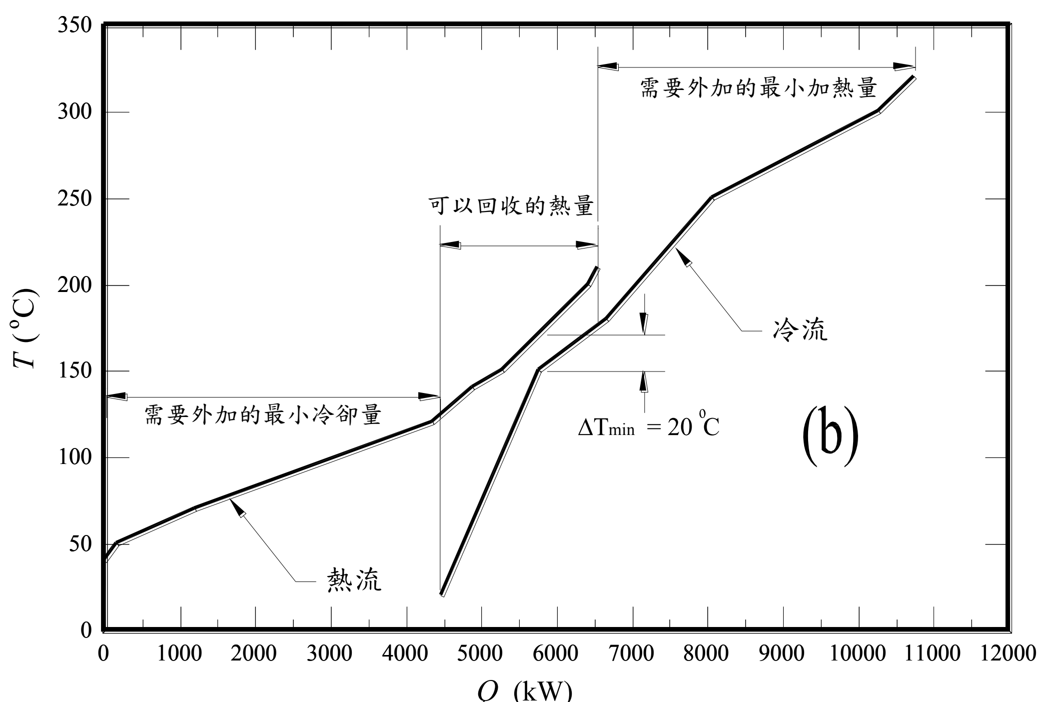
(7) 畫出排序溫度與Q_i的圖形。

一製程的工廠，擁有熱流A(要求從200°C降溫至70°C，其流量為10 kg/s，平均比熱為1.1 kJ/kg·K)、熱流B(要求從150°C降溫至40°C，其流量為8 kg/s，平均比熱為2.0 kJ/kg·K)、熱流C(要求從120°C降溫至50°C，其流量為12 kg/s，平均比熱為3.0 kJ/kg·K)、熱流D(要求從210°C降溫至140°C，其流量為2 kg/s，平均比熱為6.0 kJ/kg·K)及冷流X(要求從20°C昇溫至180°C，其流量為16 kg/s，平均比熱為2.0 kJ/kg·K)、冷流Y(要求從20°C昇溫至180°C，其流量為4 kg/s，平均比熱為1.8 kJ/kg·K)、冷流Z(要求從250 °C昇溫至320°C，其流量為16 kg/s，平均比熱為2.0 kJ/kg·K)，假設熱流與冷流的Pinch溫度為20°C，根據上述製程資料，試繪出複合耗能曲線

流 stream	溫度 (°C)	$\dot{m}$ (kg/s)	c_p (kJ/kg K)	C $\dot{m}c_p$	排序溫度	C	C_{sum}	ΔQ_i (kW)	Q (kW)
熱流						A+B+C+D			
A	200				40				0
		10	1.1	11		0+16+0+0	16	= 16×(50−40) = 160	
	70				50				= 0+160 = 160
						0+16+36+0	52	= 52×(70−50) = 1040	
B	150				70				=160+1040 = 1200
		8	2	16		11+16+36+0	63	= 63×(120−70) = 3150	
	40				120				= 1200+3150 =4350
						11+16+0+0	27	= 27×(140−120) = 540	
C	120				140				= 4350+540 = 4890
		12	3	36		11+16+0+12	39	= 39×(150−140) = 390	
	50				150				= 4890+390 = 5280
						11+0+0+12	23	= 23×(200−150) = 1150	
D	210				200				= 5280+1150 = 6430
		2	6	12		0+0+0+12	12	= 12×(210−200) = 120	
	140				210				= 6430+120 = 6550

流 stream	溫度 (°C)	$\dot{m}$ (kg/s)	c_p (kJ/kg K)	$C = \dot{m} c_p$	排序溫度	C	C_{sum}	ΔQ_i (kW)	Q (kW)
冷流						X+Y+Z			
X	20				20				0
		10	1	10		10+0+0	10	= 10×(150−20) = 1300	
	180				150				= 0+1300 = 1300
						10+20+0	30	= 30×(180−150) = 900	
Y	150				180				= 1300+900 = 2200
		8	2.5	20		0+20+0	20	= 20×(250−180) = 1400	
	300				250				= 2200+1400 = 3600
						0+20+24	44	= 44×(300−250) = 2200	
Z	250				300				= 3600+2200 = 5800
		6	4	24		0+0+24	24	= 24×(320−300) = 480	


冷流
熱流
(a)
T (°C)
Q (kW)


需要外加的最小加熱量
可以回收的熱量
冷流
需要外加的最小冷卻量
ΔTmin = 20 °C
(b)
熱流
T (°C)
Q (kW)

Problem Table Construction

- 將熱流與冷流的溫度以中間溫度表示，即原有的熱流溫度減掉 *ΔTmin/2*，而原有的冷流溫度加上*ΔTmin/2*，並將各個區段間的*Ci*值列入表格中。
- 將所有的中間溫度混和排序(包括熱流與冷流)，由最高溫度排到最低溫度。
- 將各個區間的各個熱容量流率加總，$C_{sum,i} = \sum(C_{hot,i} - C_{cold,i})$。
- 算出各個區間的熱量 = $\Delta Q_i = C_{sum,i}\Delta T$ 。
- 由最高溫度開始，首先將最高溫度的*Qi*定義為零，然後依序隨溫度下降時，累計實際的總熱傳量$Q_i = Q_{i-1} + C\Delta T$。
- 找出步驟(5)計算出的最小值，然後將此值定義為零(因此其他區間中不是零的值，必須加上這個最小值，例如最小值為–5000 kW，某一區間的值為 –2000 kW，則將最小值(–5000 kW)定義為零後，此一「某一區間」的值將變為–2000–(–5000) = 3000 kW。
- 由步驟(6)算出的結果，表的第一與最後一欄代表所需要外加的最小熱源與外加的最小冷源，此一表格稱之為問題表。
- 將問題表的最後一個欄位與中間溫度作圖，稱之為總複合負載曲線。

The Composite Curves
Problem Table

(1) Make a list of all the temperatures at which streams begin or end or change heat capacity flowrate, *CP*. Then rank them in ascending order, as in the first column of Table 6.2.

(2) List all the heat capacity flowrates of streams that have temperatures within each temperature interval. This is the second column of Table 6.2.

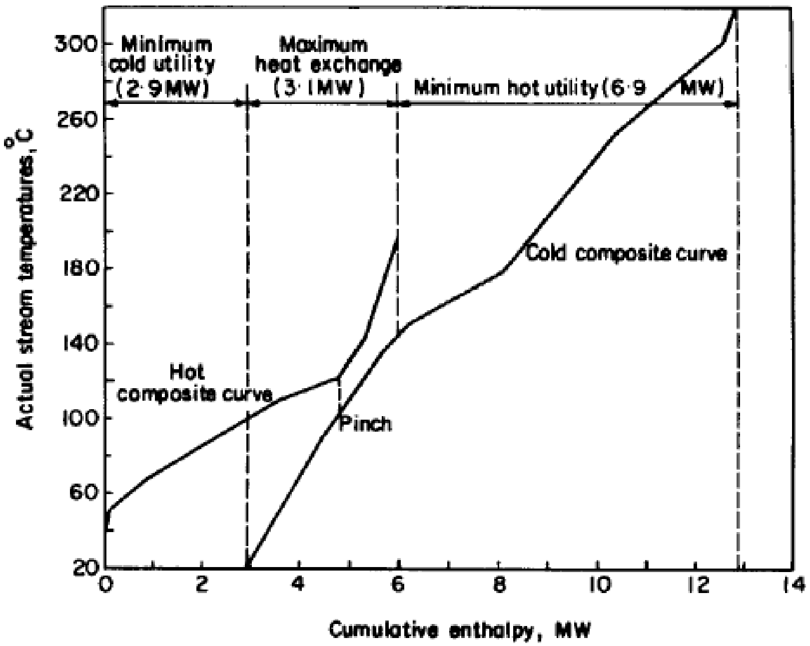
(3) Where more than one cold stream exists in an interval, as between 152°C and 179°C and between 254°C and 302°C, add together the *CP*s to give the total in column 3.

(4) Multiply the total *CP* by the temperature change from beginning to end of the interval. This gives the change in enthalpy of all the cold streams over the interval (column 4).

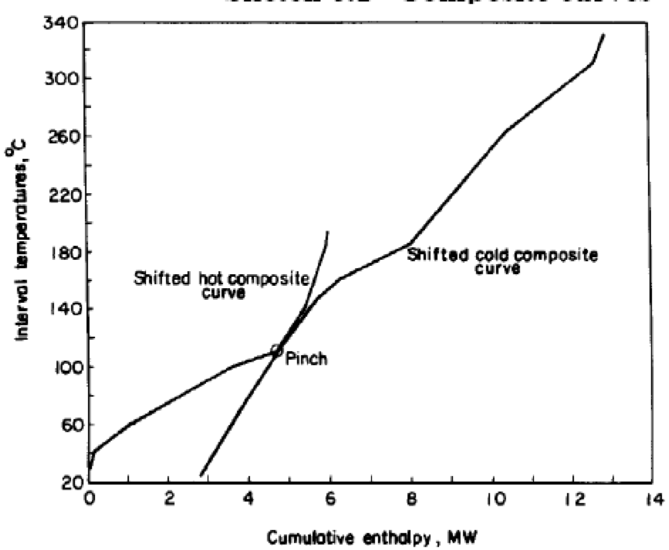
TABLE 6.1 Stream data for example problem

	1	2	3	4
Stream	Temperature °C	Enthalpy MW	Heat Capacity Flowrate (*CP*) MW/K	Interval Temperature °C
HOT STREAM				
A	199	1.35	0.0101	189
	123	0.58	0.0109	113
	70	0		60
B	144	1.51	0.0150	134
	112	1.03	0.0169	102
	70	0.32	0.0110	60
	41	0		31
C	123	3.02	0.0773	113
	112	2.17	0.0362	102
	52	0		42
D	204	0.12	0.0020	194
	144	0		134
COLD STREAM				
X	21	0	0.0218	31
	60	0.85	0.0241	70
	92	1.62	0.0253	102
	111	2.10	0.0263	121
	138	2.81	0.0376	148
	179	4.35		189
Y	152	0	0.0313	162
	302	4.70		312
Z	254	0	0.0146	264
	319	0.95		329

Note: $DT_{min} = 20°C$
Interval Temperatures are defined as follows:
for Hot Stream, $TI = T - DT_{min}/2 = T - 10$,
for Cold Stream, $TI = T + DT_{min}/2 = T + 10$.



Sketch 6.1 Composite curves



Sketch 6.2 Shifted composite curves (interval temperature basis)

TABLE 6.2 Calculation of Cold Composite Curve

1	2	3	4	5
Temperature	Heat Capacity Flowrate, MW/K		Enthalpy, MW	
°C	Individual Streams X + Y + Z	Total CP_i	Increment	Cumulative
21				0
	0.022	0.022	0.85	
60				0.85
	0.024	0.024	0.77	
92				1.62
	0.025	0.025	0.48	
111				2.10
	0.026	0.026	0.71	
138				2.81
	0.038	0.038	0.53	
152				3.34
	0.038 + 0.031	0.069	1.86	
179				5.20
	0.031	0.031	2.35	
254				7.55
	0.031 + 0.015	0.046	2.20	
302				9.75
	0.015	0.015	0.25	
319				10.00

The hot composite curve is derived in precisely the same way using the hot streams, A to D.

(1) Convert all stream temperatures to interval temperatures by adding (*i.e.* 10°C) to cold stream temperatures and subtracting from hot stream temperatures. This gives the last column in Table 6.1.

(2) List all the interval temperatures at which streams begin or end or change heat capacity flow rate, *CP*.

(3) Rank these interval temperatures in descending order. This is column 1 of Table 6.3.

(4) For each temperature interval, add together the heat capacity flowrates of all the hot streams which exist in that temperature interval and subtract the heat capacity flowrates of all the cold streams, to give the net heat capacity flowrates (columns 2 and 3 of Table 6.3).

(5) Multiply the net heat capacity flowrate by the temperature range of the interval to give the net heat required or released by all the streams in the interval (column 4 of Table 6.3).

(6) Starting from a zero input at the highest temperature, work down the table, adding on the heat change in each temperature interval to give a heat cascade, or cumulative heat passing through at the particular temperature (column 5). This preliminary problem table will normally have infeasibilities. These are temperatures at which the net heat cascaded through the system is negative. These infeasibilities must be removed.

(7) Find the largest negative number in column 5 of the preliminary problem table. Add this amount of heat, *Q*, as hot utility at the top of the cascade. All the other values in the cascade will also increase by *Q*, to give a new series, column 6 of Table 6.3, This is the feasible problem table. The heat *Q* added is equal to the target hot utility requirement. The pinch is the point at which there is zero net heat flow in the cascade, corresponding to the point where there was the largest infeasibility. This occurs at an interval temperature of 113°C.

(8) The first and last values in the final column of the problem table are respectively the minimum hot and cold utility requirements.

TABLE 6.3 Calculation of Heat Cascade (Problem Table)

1	2				3	4	5	6
Interval Temperature °C	Heat Capacity Flowrate, MW/K					Heat Increment MW	Total Heat Cascades	
	Individual Streams $CP_{hot}-CP_{cold}$				Total		Infeasible MW	Final MW
329							0.000	6.900
				− 0.0146	−0.0146	−0.248		
312							−0.248	6.652
			− 0.0313	− 0.0146	−0.0459	−2.203		
264							−2.451	4.449
			− 0.0313		−0.0313	−2.191		
194							−4.642	2.258
		+ 0.0020	− 0.0313		−0.0293	−0.147		
189							−4.789	2.111
	0.0101	+ 0.0020	− 0.0313	− 0.0376	−0.0568	−1.534		
162							−6.323	0.577
	0.0101	+ 0.0020		− 0.0376	−0.0255	−0.357		
148							−6.680	0.220
	0.0101	+ 0.0020		− 0.0263	−0.0142	−0.202		
134							−6.882	0.018
	0.0101	+ 0.0150		− 0.0263	−0.0012	−0.016		
121							−6.898	0.002
	0.0101	+ 0.0150		− 0.0253	−0.0002	−0.002		
113							−6.900	0.000
	0.0109	+ 0.0150	+ 0.0773	− 0.0253	0.0779	0.857		
102							−6.043	0.857
	0.0109	+ 0.0169	+ 0.0362	− 0.0241	0.0399	1.277		
70							−4.766	2.134
	0.0109	+ 0.0169	+ 0.0362	− 0.0218	0.0422	0.422		
60							−4.344	2.556
		+ 0.0110	+ 0.0362	− 0.0218	0.0254	0.457		
42							−3.887	3.013
		+ 0.0110		− 0.0218	−0.0108	−0.118		
31							−4.005	2.895

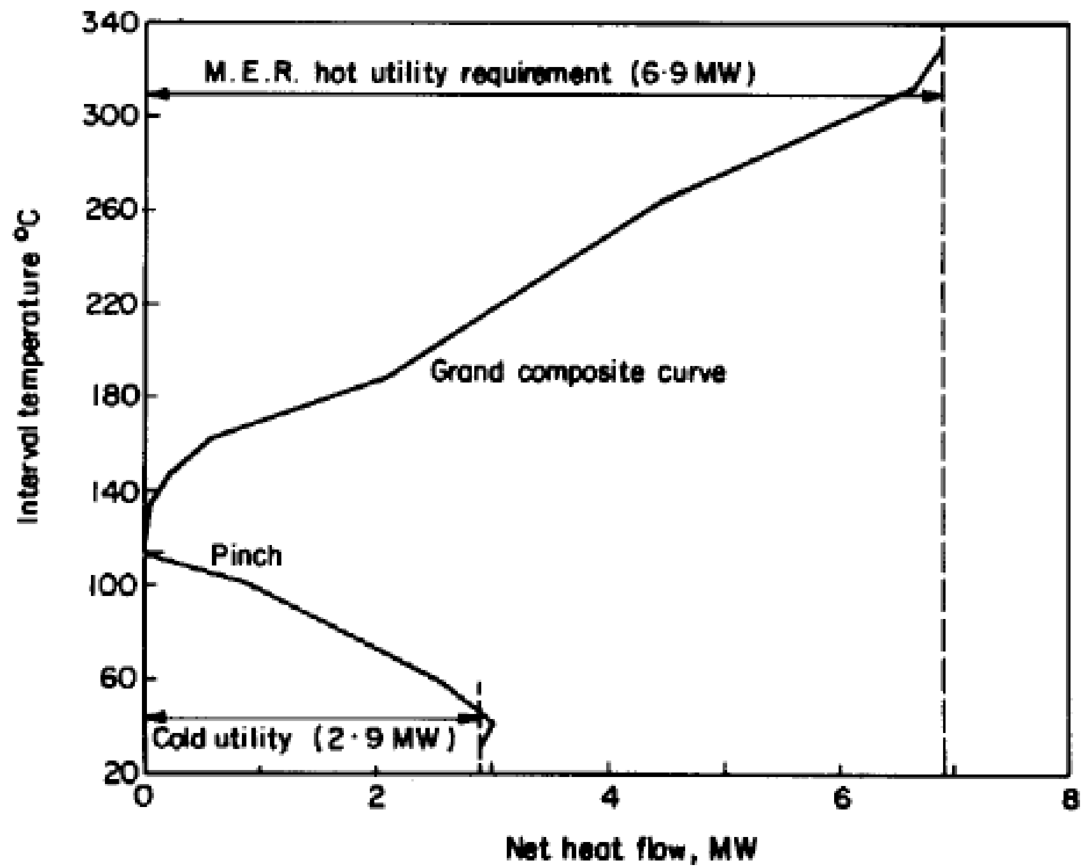
Maximum infeasibility (largest negative value) in the infeasible heat cascade is 6.9 MW at an interval temperature of 113°C. This is therefore the minimum amount of hot utility which must be added to the cascade, *i.e.*, the Minimum Energy Requirement (M.E.R.) target. The cold utility target is 4.0 MW less at 2.9 MW. The pinch temperature of 113°C interval temperature corresponds to a hot stream temperature of 123°C and a cold stream temperature of 103°C.

The Grand Composite Curve

(1) It is a plot of the enthalpy difference (horizontal axis) between the hot and cold composite curves with the necessary temperature correction, ΔT_{min}, included.

(2) It shows the net heat flow through the process, from hot utilities and hot streams to cold utilities and cold streams, at any interval temperature.

(3) It shows the amount of heat required by or released from the process at any given temperature and thus how much hot or cold utility can be used at different levels.



Sketch 6.3 Grand composite curve

Interpretation of the Graphs

- the relative importance of heat exchange and utility heating/cooling,
- the variation of driving forces with temperature,
- the basis of the calculation of heat exchanger area
- ΔT_{min}

- The grand composite curve primarily reveals:
- the pinch temperature,
- other areas of low heat flow where design will be tightly constrained,
- utility use at different temperature levels,
- the best placement of separation and reaction systems.

Grand composite curve (Conti..)

- The point of zero heat flow in the grand composite curve is the heat recovery pinch.
- The open “Jaws” at the top and bottom are $Q_{h,min}$ & $Q_{c,min}$, respectively.

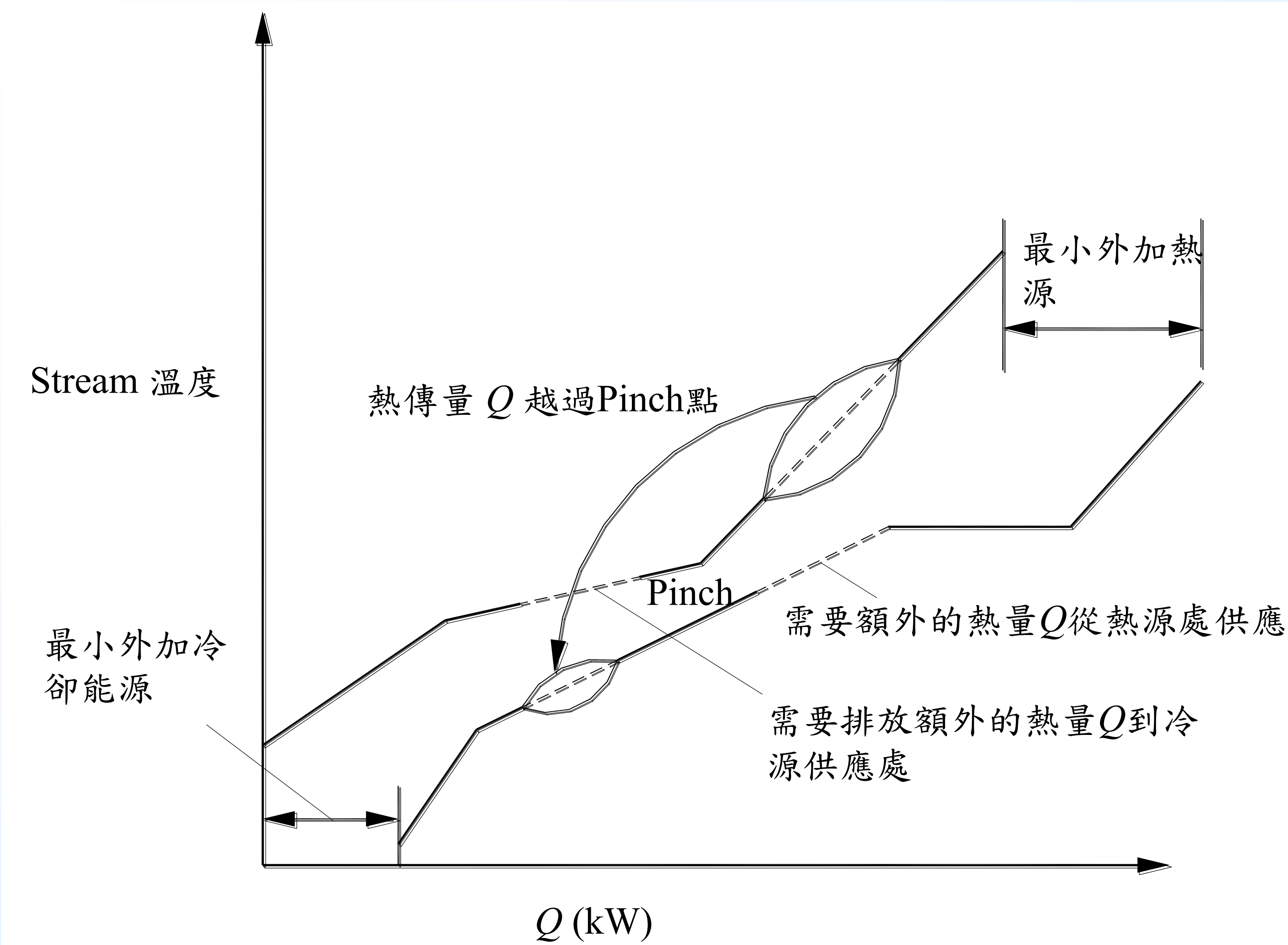
Pinch 技術提供了三大黃金原則 (three golden rules)

(1) 熱交換器的網路設計連結，不可以穿越 Pinch 點。

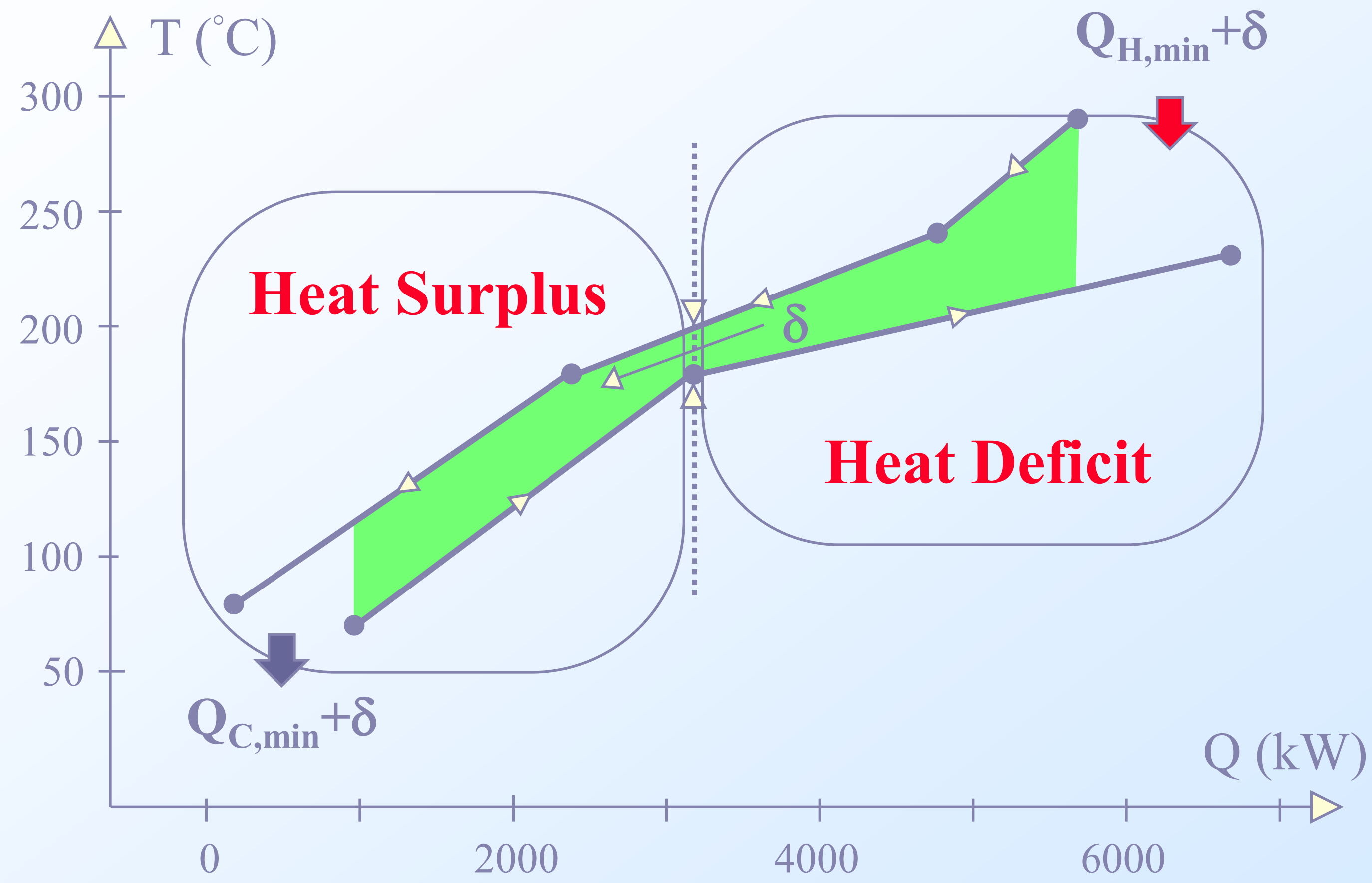
(2) 在Pinch的右上半部設計上，不要使用外加的冷卻源。

(3) 在Pinch的左下半部的設計上，不要使用外加的加熱源。

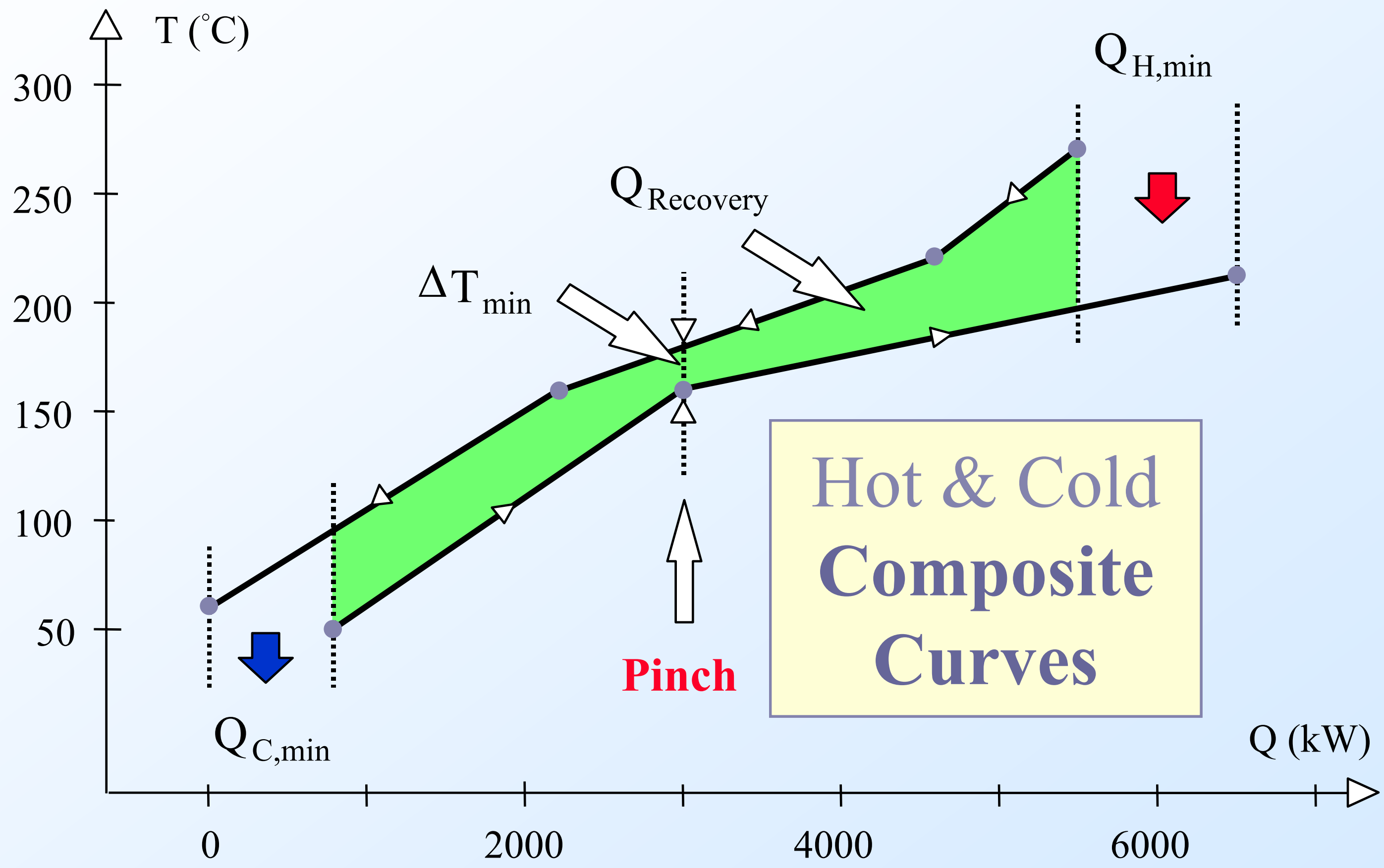
熱傳設計穿越Pinch的缺失



A Fundamental Decomposition



The Heat Recovery Pinch

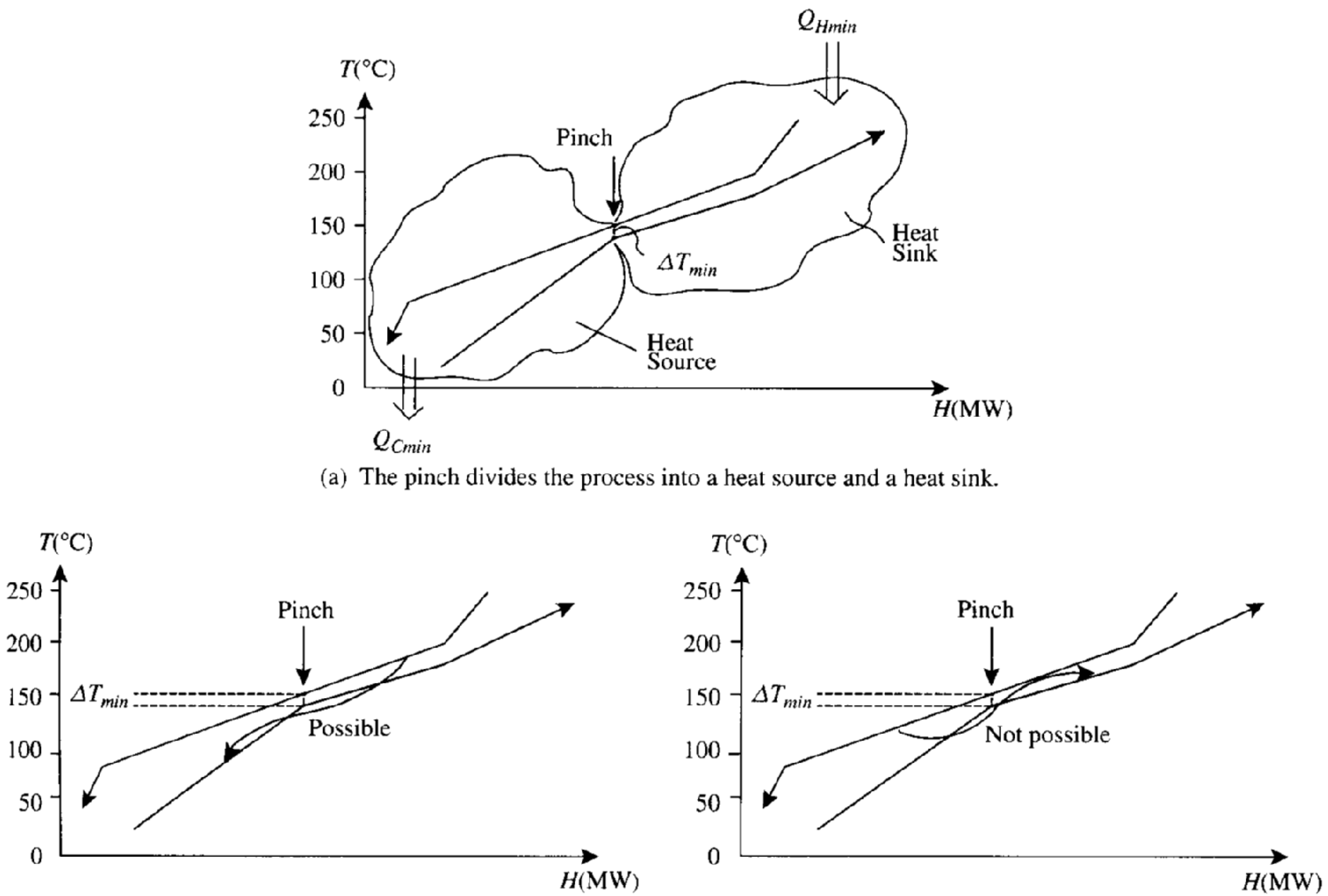


Hot & Cold
Composite Curves

Pinch (Conti..)

- Divide the process into
  - Above the pinch
    - Acts as a heat sink
  - Below the pinch
    - Acts as a heat source
- Transfer heat above or below the pinch
  - Feasible & infeasible
  - No heat exchangers should be designed with a temperature difference smaller than pinch.

Pinch (Conti..)



(a) The pinch divides the process into a heat source and a heat sink.

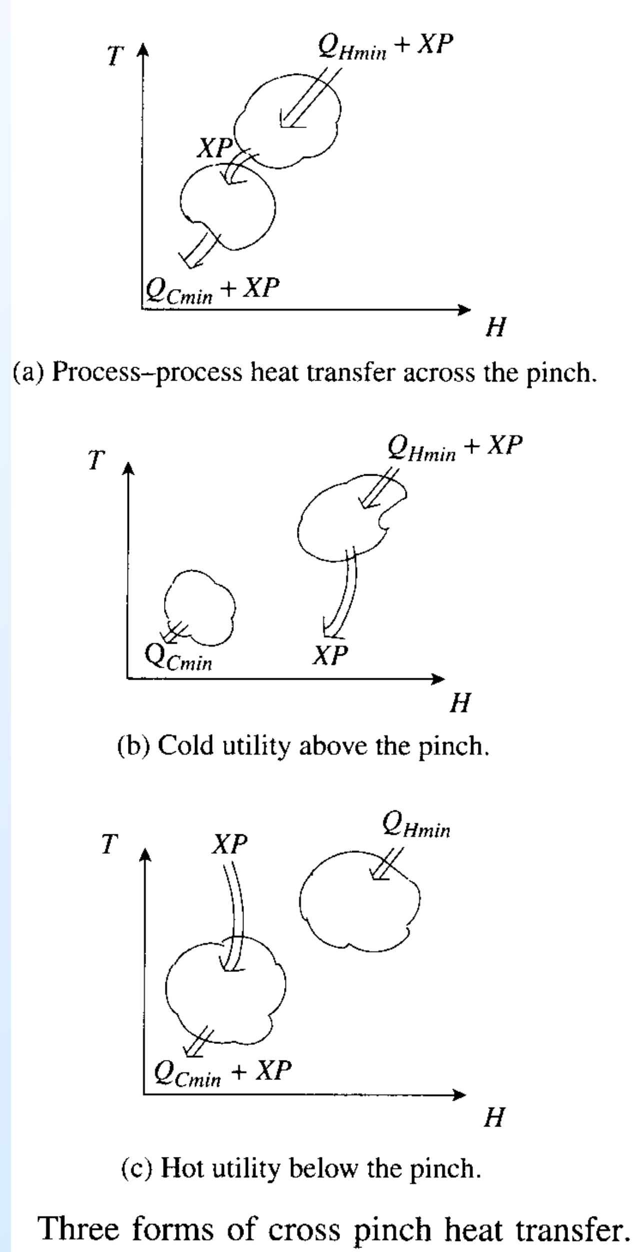
(b) Heat transfer from above the pinch to below the pinch is possible.

(c) Heat transfer from below to above the pinch is not possible without violating ΔT_{min}.

The composite curves set the energy target and the location of the pinch.

Interpretation of 3 golden rules

- If an amount of heat XP is transferred from the system above the pinch to below the pinch, it will create a deficit of heat XP above the pinch and a surplus of heat below the pinch. → An extra XP above the pinch and an extra XP below the pinch is needed.
- If cooling utility (amount = XP) is used to cool hot stream above the pinch, it will create an imbalance in the system above the pinch. To satisfy the imbalance, an import of (Q_{Hmin}+XP) heat from hor utility is required. Overall, $Q_{c,min}$+XP cold utility is used.
- Similar condition occurs when placing a hot utility below pinch.



(a) Process–process heat transfer across the pinch.

(b) Cold utility above the pinch.

(c) Hot utility below the pinch.

Three forms of cross pinch heat transfer.

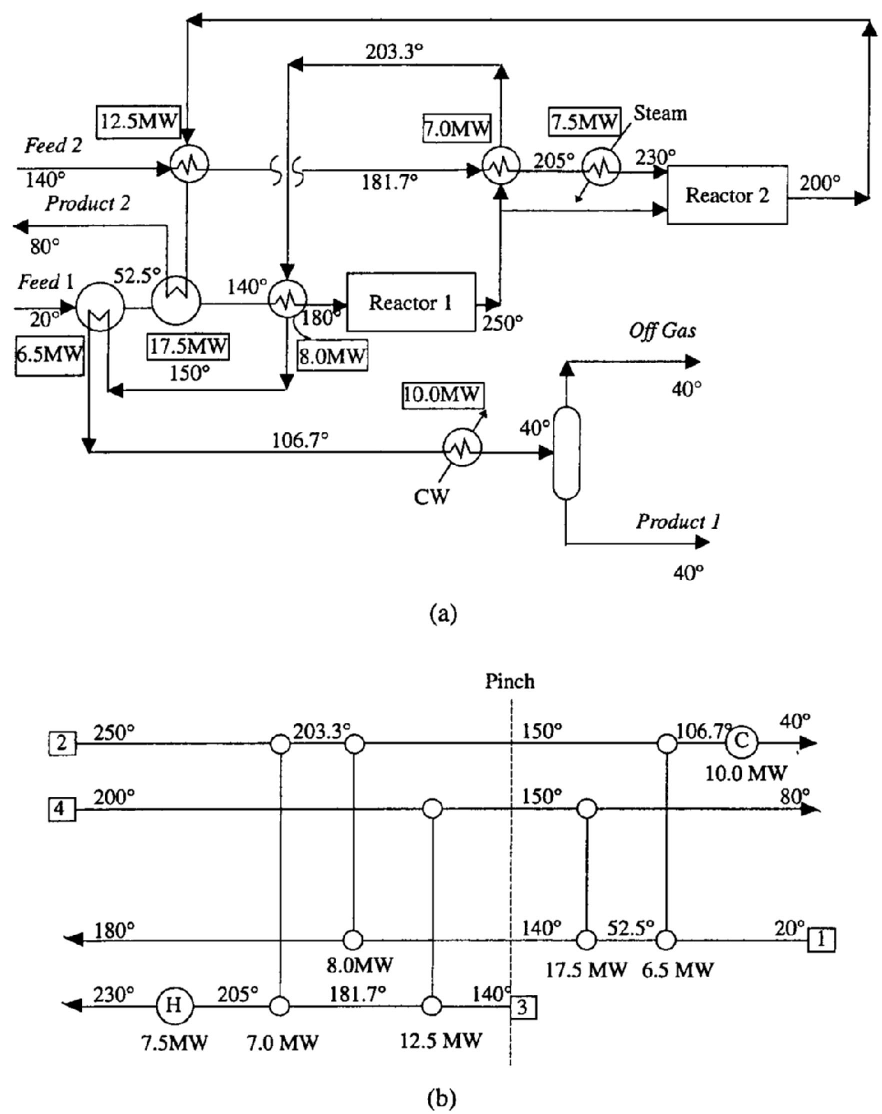
In essence

- To achieve the energy target set by composite curve, the designer should not transfer heat across pinch by
  - Process to process heat transfer
  - Inappropriate use of utility
- These rules are sufficient/necessary to meet the energy target when the smallest temperature of the individual heat exchanger are all below $\Delta T_{min.}$

Grid Diagram

- Heat exchanger arrangement to meet the target.
- General rule
  - Placing hot stream on top from left to right.
  - Placing cold stream at the bottom half from right to left.
  - Indicating the position of pinch and design the corresponding heat exchanger from pinch to both directions.
- Placing (H) as an indicator of heating utility.
- Placing (C) as an indicator of cooling utility.
- Placing a link of circle between hot and cold stream to indicate a match of heat exchanger.

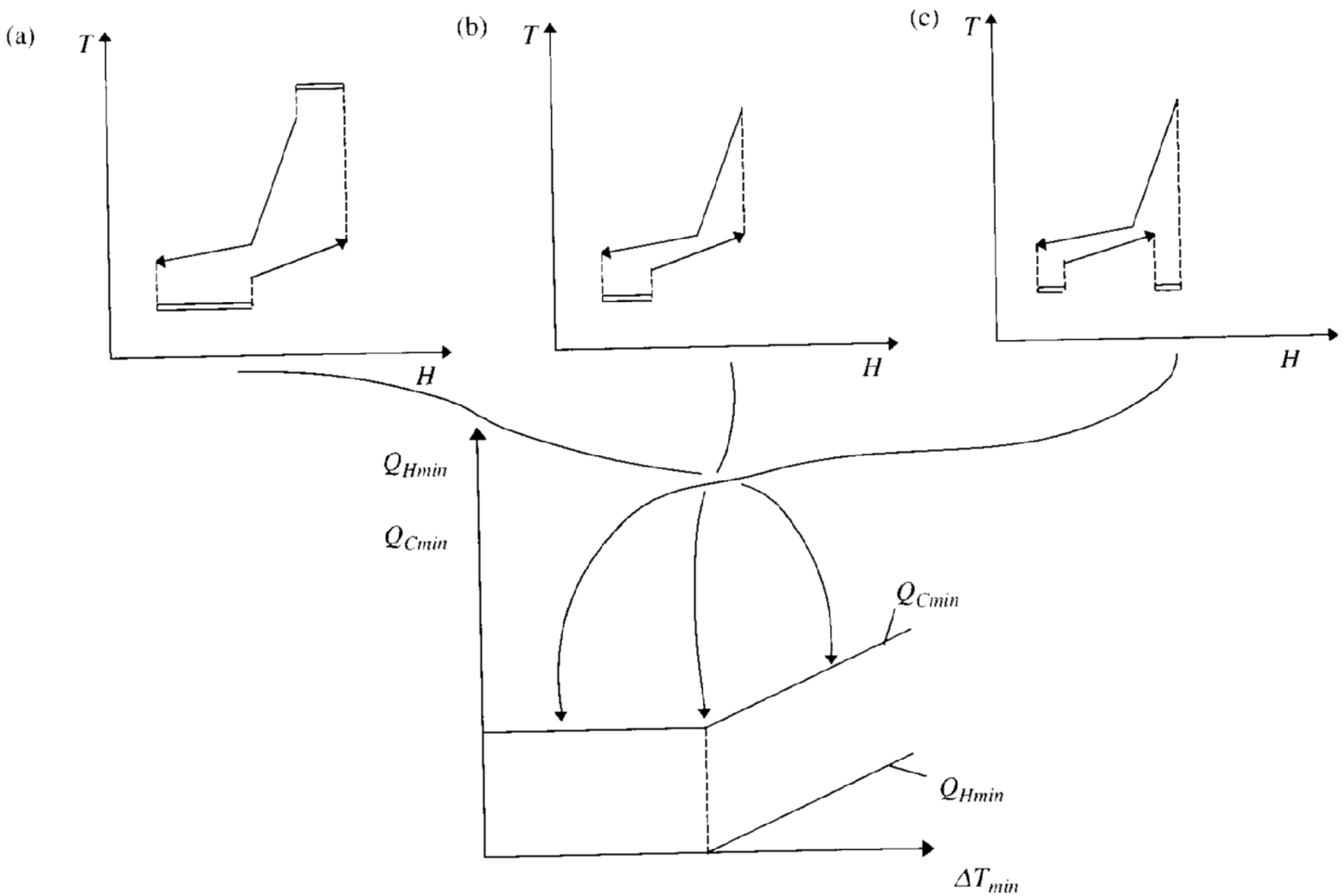
A Typical Result



Threshold problem

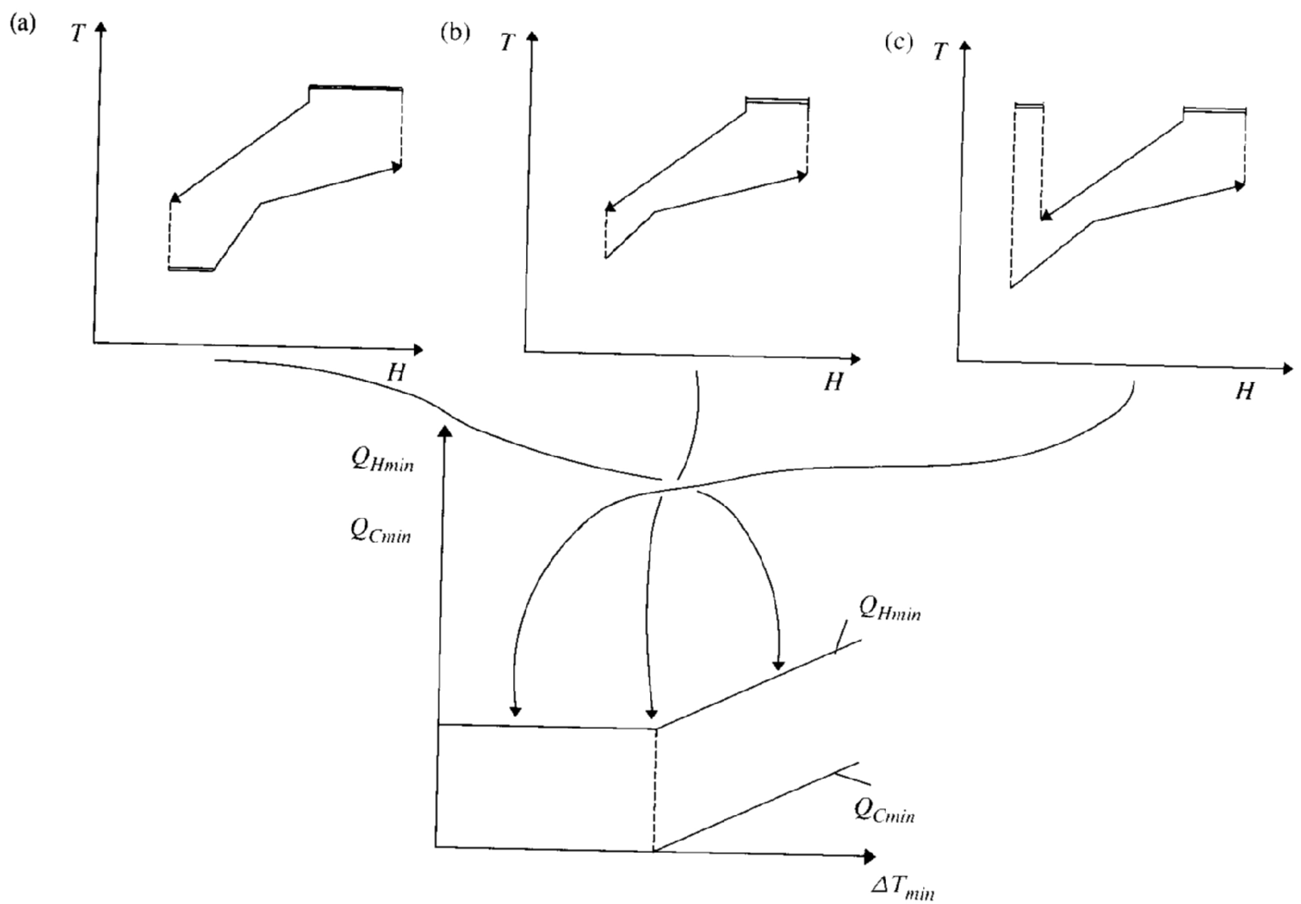
- Not all problems have a pinch to divide the process into two parts (i.e. above and below the pinch). E.g. in the following (b), the composite curve is in alignment with the hot end, indicating no longer need for the hot utility. Moving closer together in the (c) opens up a demand at the cold end but opens up a demand for cold utility at the hot end → corresponding with adecrease at the cold end. In short, as the curves moved closer, beyond the setting in the following (b), the utility demand is a constant.
- In some threshold problems, the hot utility requirement disappear. In others, the cold utility disappears.

Threshold problem (Conti..)



As ΔT_{min} is varied, some problems require only cold utility below a threshold value.

Threshold problem (Conti..)

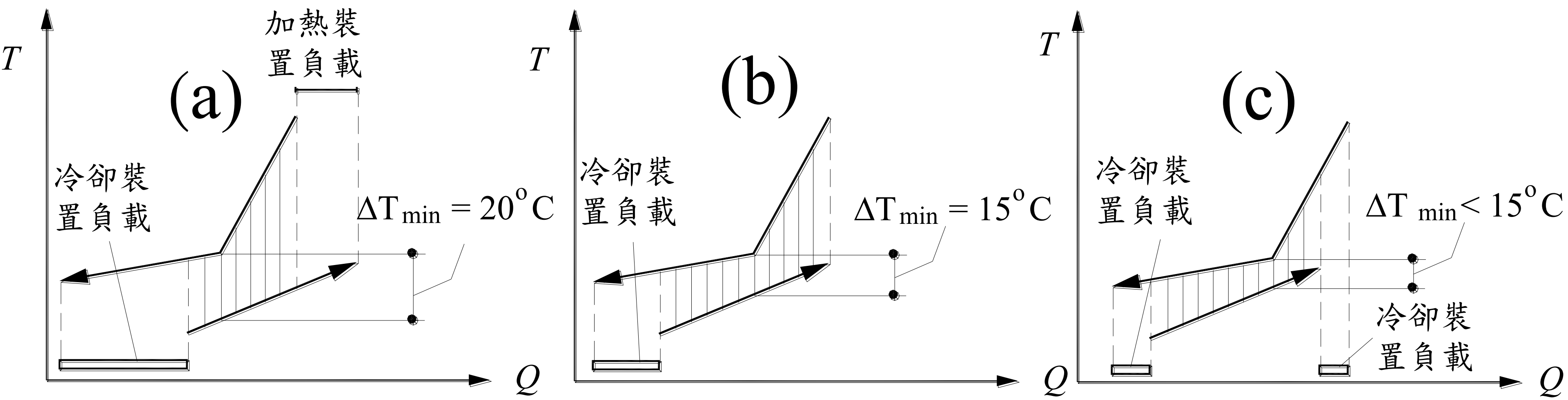


In some threshold problems, only hot utility is required below the threshold value of ΔT_{min}.

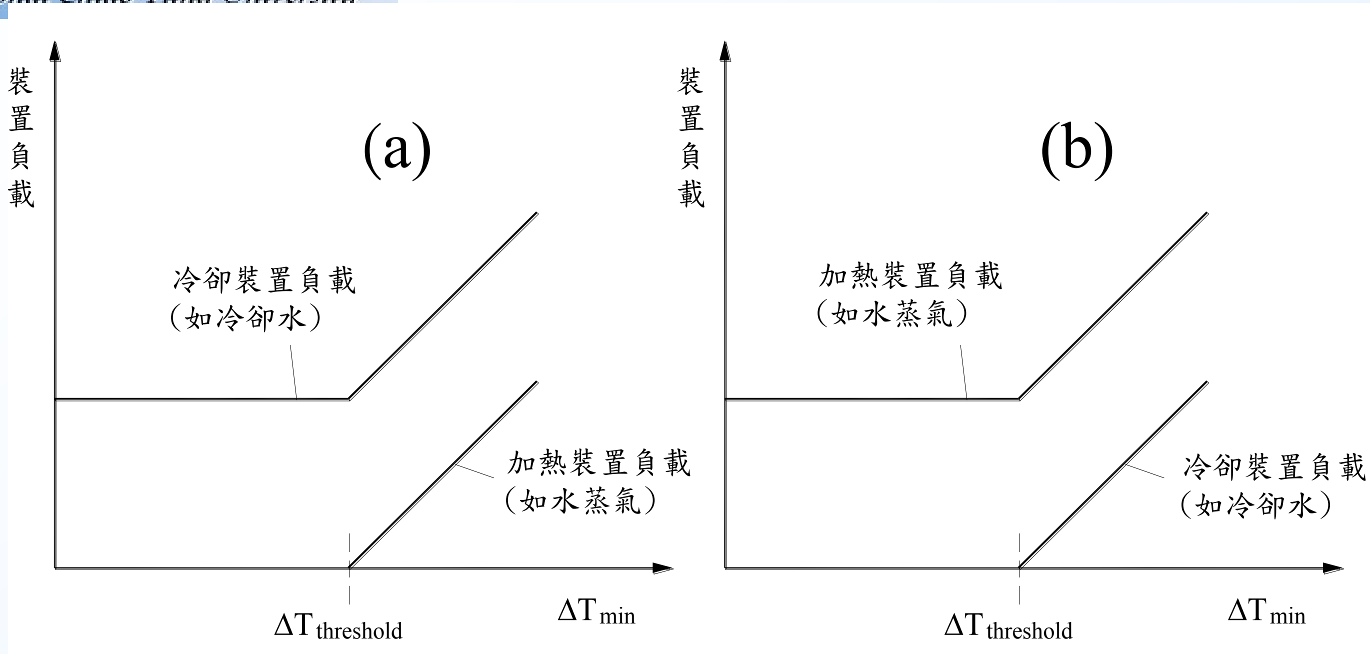
Threshold Problem (Conti..)

- Below the threshold ΔT_{min}, energy cost are constant since utility demand is constant.
- The flat profiles of energy cost below ΔT_{min}, indicating optimum can never achieved when temeprature is below $\Delta T_{min,\ threshold}$.

圖12-20 Threshold問題



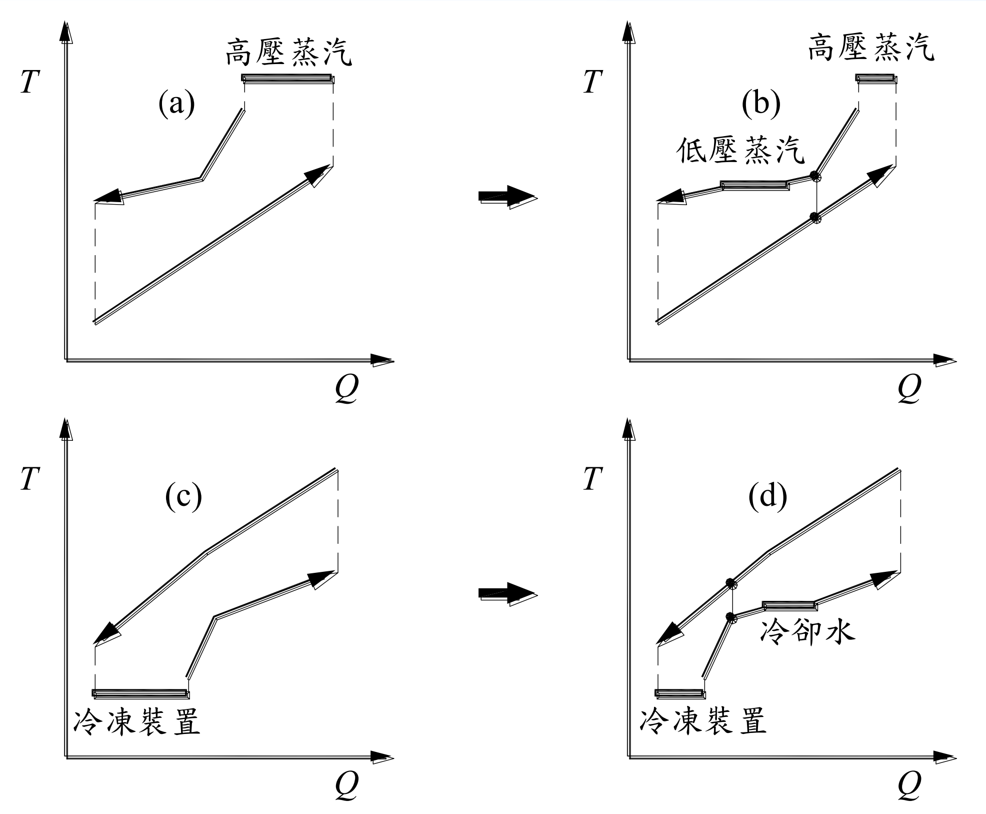
如圖12-20(a)的複合負載曲線，假設其原先的$\Delta T_{min} = 20$℃，*原有的設計各需要冷源與熱源；若我們將此一複合負載曲線的冷流向左平移，顯然的*ΔT_{min}會逐漸縮小，而冷側所需要的熱源也跟著縮小；*假設當*$\Delta T_{min} = 15$℃*時，冷流端點與熱流端點到達同一位置(即冷流端無須藉用加熱裝置熱源)，此時就到了一個臨界設計點(threshold design)，*$\Delta T_{threshold}$*當冷流*持續向左移動越過臨界點，雖然冷側不需要熱源，卻造成熱流側的兩端同時需要冷源來冷卻，而且此一冷源的冷卻量則維持固定*(如圖12-21)*，*不會因為*ΔT_{min}縮小而持續降低，也就是說無法持續降低冷源的使用量；但是因為ΔT_{min}縮小後反而會造成熱交換器變大，更何況冷源的冷卻量並不會持續縮小，因此設計在此臨界點後並無實質的幫忙與意義。



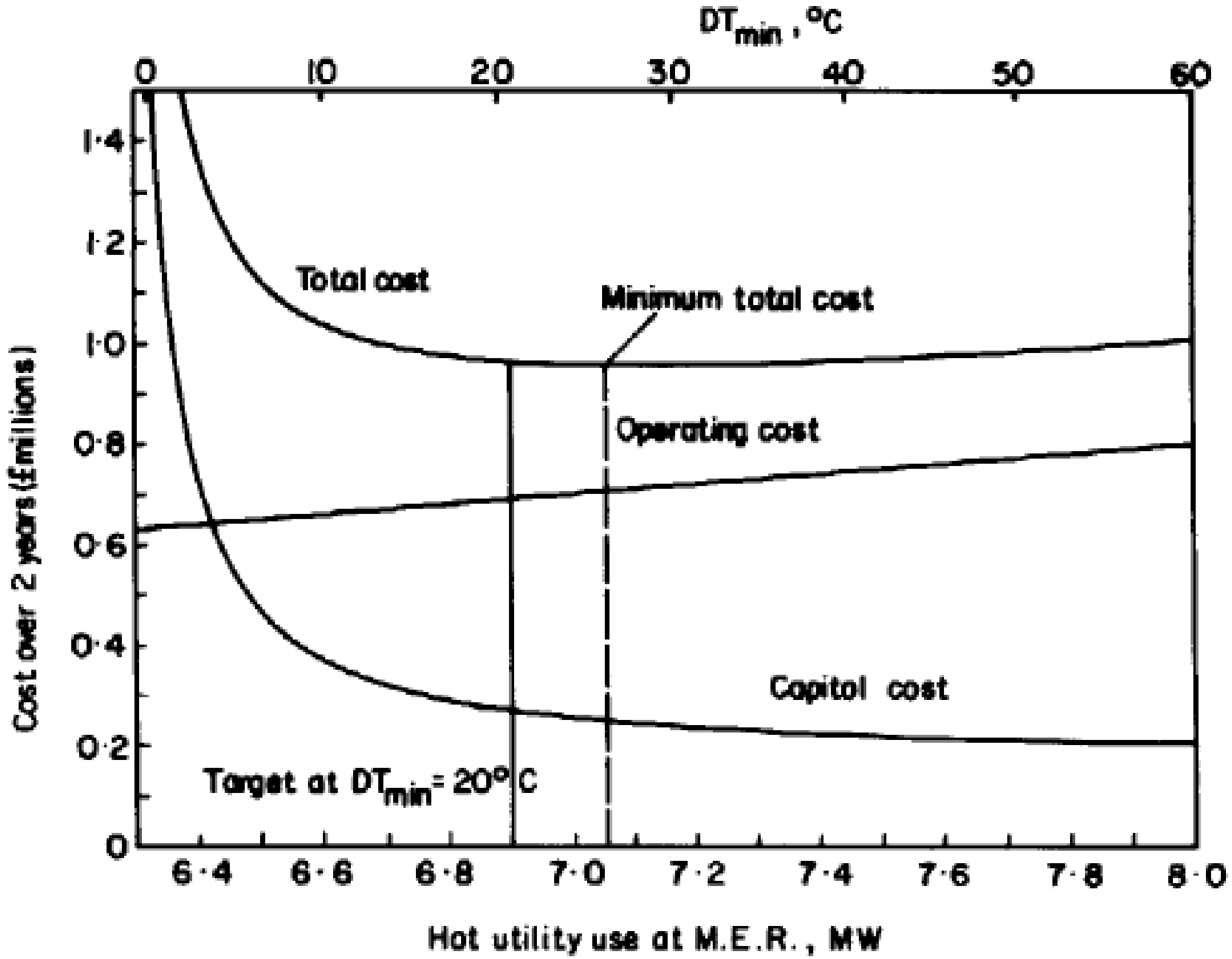
通常我們對ΔT_{min} 都有一個預估的經驗值，即$\Delta T_{min,exp}$；如果$\Delta T_{threshold} >> \Delta T_{min,exp}$，在這個條件下，冷流與熱流*負載曲線岔的相當的開，如前所說明，使用*$\Delta T_{min,exp}$ *設計將無實質的幫助，因此設計上可以*$\Delta T_{threshold}$當做設計點，其設計步驟可說明如下：

- *由於在臨界點溫度下，其中一側無須使用熱源或冷源，因此該側設計必須要恰好滿足冷熱交換，不能供應多餘的冷源或熱源。*
- *另一側需要冷源或熱源，將冷源或熱源置於開始處(即冷熱流端點起始處)。*
- *接下來剩下的設計相當直接也非常簡單，只需將合適的加以撮合，通常有很多答案均可滿足這一側的設計，要注意設計的操作性是否良好與安排是否恰當等的考量。第二種情形是*$\Delta T_{threshold} \approx \Delta T_{min,exp}$，此時在快速設計的需求上，可由$\Delta T_{threshold} = \Delta T_{min,exp}$進行第一手設計。

多Pinch的觀念

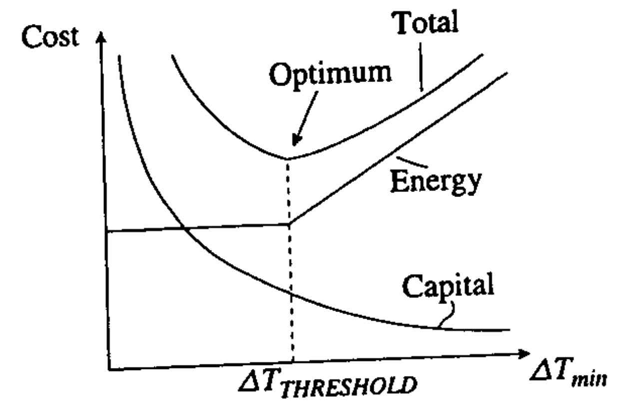


- 外加冷卻設備與外加加熱設備的選用上除了要注意減少使用量外，另外要特別注意盡量避免使用高溫加熱用的鍋爐蒸氣或低溫的冷凍系統，這是因為其能源成本相對較高；以圖12-22Pinch設計來說明，圖12-22(a)中的原設計使用了高壓的蒸氣來加熱使得冷流能夠達到設計溫度，為了減少高壓蒸氣使用的運轉成本，設計上可加入一個低壓的蒸氣鍋爐，同時減少高壓鍋爐的容量(見圖12-22(b))，此時雖然增加了一個加熱裝置，但如果考慮長期的運轉成本，還是相當值得的；同樣的，圖12-22(c)的原設計使用了大量的低溫冷凍來冷卻熱流，利用圖12-22(d)的改善設計，引入常溫的冷卻裝置(如一般的冷卻水)，同樣的可以達到減少低溫冷凍系統的裝置容量

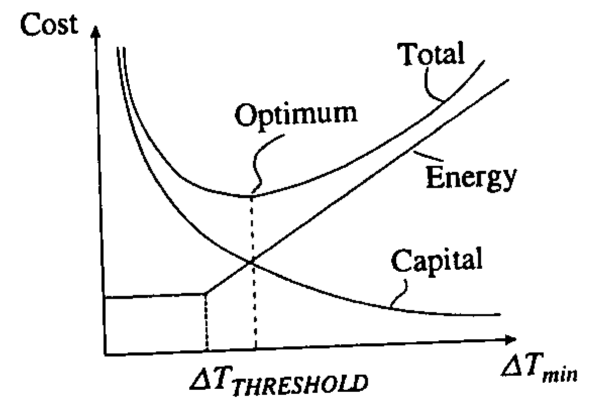


Sketch 9.4 Cost – energy graph

Threshold Problem (Conti..)



(a) The capital - energy trade-off can lead to an optimum at threshold ΔT_{min}



(b) The capital - energy trade-off can lead to an optimum above $\Delta T_{THRESHOLD}$

The data collection phase usually has the following stages in a study of an existing plant

- Obtain, or draw up, a plant flowsheet that shows all the main process flows requiring heating or cooling. Details of ancillary equipment such as pumps and valves are not required.
- Collect data on feed rates, production throughput, flowrates, reflux and recycle ratios, fluid densities, and form a mass balance. This will show all the mass flows in the plant.
- Collect data on temperatures, fuel use, heat exchanger ratings and performance, and specific heats.
- Use this to form a heat balance, analogous to the mass balance but showing temperatures and heat flows.
- Use the heat and mass balances to extract the stream data. These are the data in the form required for the Process Integration analysis and consist of a series of streams requiring heating or cooling.
- For each of these, temperature-enthalpy data are required.

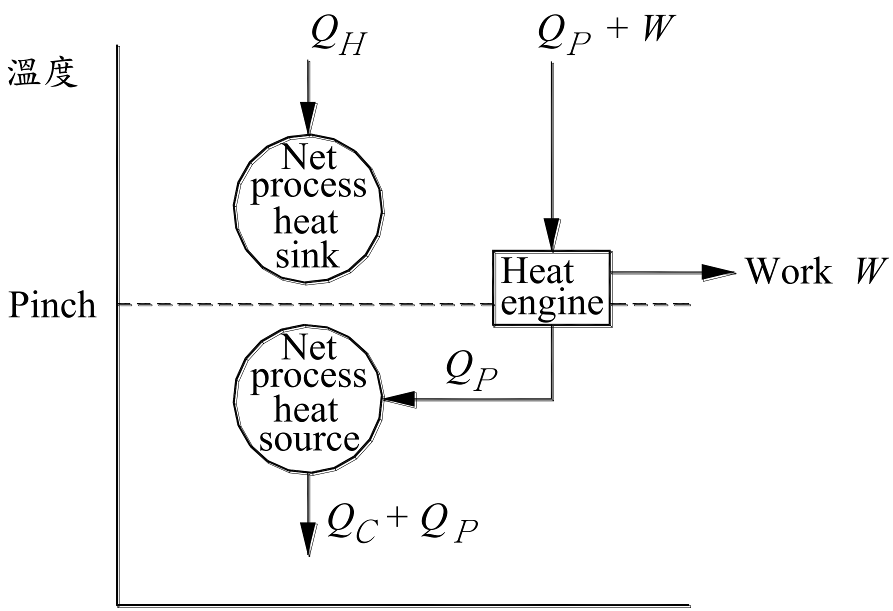
Utility selection

- The most common hot utility → steam
- High temperature heating duty → Furnace flue gas or hot oil circuit.
- Cold utility → refrigeration, cooling water, air-cooling, furnace air preheating, boiler feedwater preheater, steam generation.
- Do not use cooling water above pinch.

熱交換器、外加冷卻設備與外加加熱設備的選用原則

- 儘量使用便宜的低階加熱設備，避免使用昂貴的高階能源(例如電能)。
- 儘量使用常溫的冷卻水與較高溫的冷凍設備來冷卻，避免使用昂貴的低溫冷凍設備(例如冷凍系統)。
- 在Pinch點下時，嘗試增加熱水或水蒸氣的產生量，這些熱水或水蒸氣可以用於鍋爐飼水器預熱或用於空間加熱。
- 檢查是否有其他的設備可以提昇整體設備性能，例如汽電共生設備

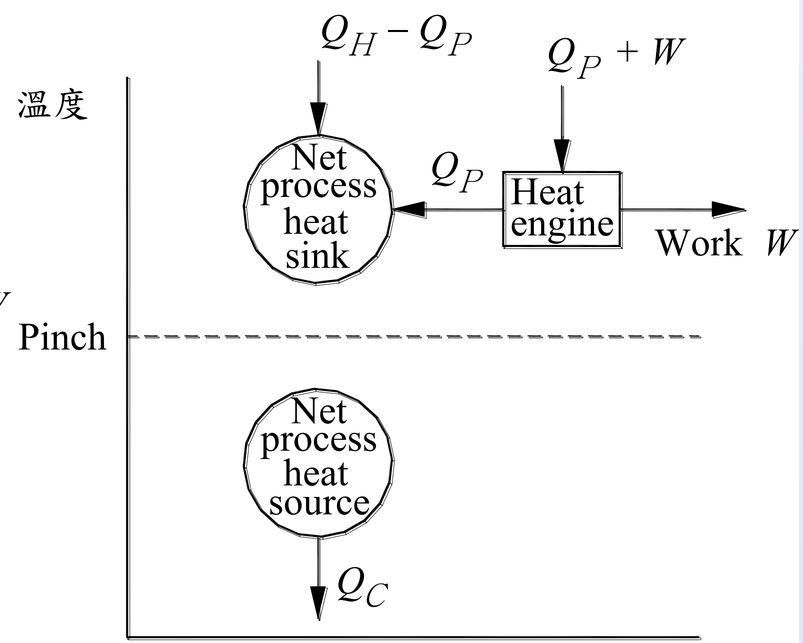
熱引擎於Pinch上的正確使用方法，其中(a)為錯誤的使用方法(b)為正確的設計



整體能源需求 = $Q_H + (Q_P + W)$

整體冷卻總需求 = $Q_C + Q_P$

此設計無法節省能源

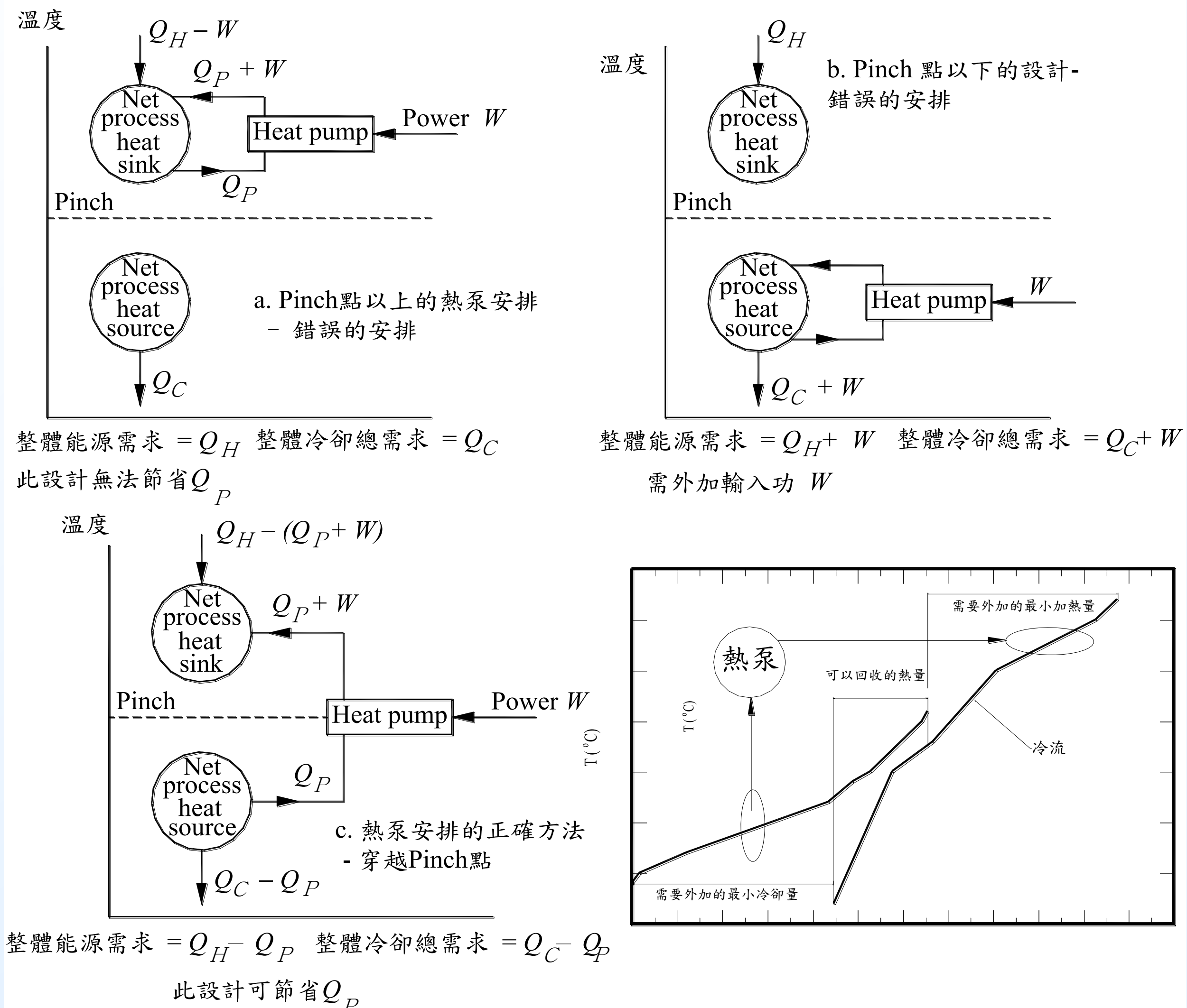


整體能源需求 = $Q_H + W$

整體冷卻總需求 = Q_C

此設計可節省 Q_P

熱泵於Pinch上的正確使用方法，其中(a)與(b)為錯誤的使用方法(c)為正確的使用方法



Principal components of capital costs

- Number of units (matches hot and cold streams)
- Heat exchanger area
- Number of shells
- Materials of construction
- Type of heat exchanger
- Pressure rating

From the graph theory it can be proved:

- $N_{units} = (N_{stream\ number} - 1)$
- i.e.,

  $N_{units} = (N_{above\ pinch} - 1) + (N_{below\ pinch} - 1)$

NOTE: the number of stream includes heating/cooling utility

Heat Exchanger Target

- $A_{network,k} = \Delta h_k/(U\Delta T_{m,k})$

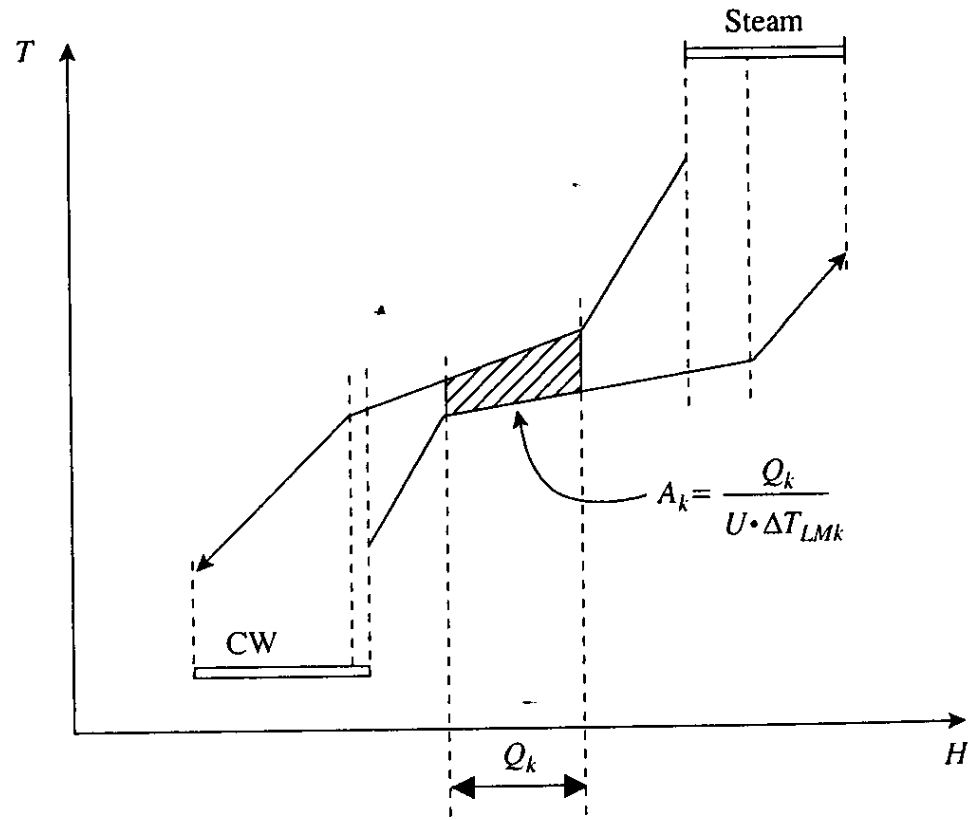
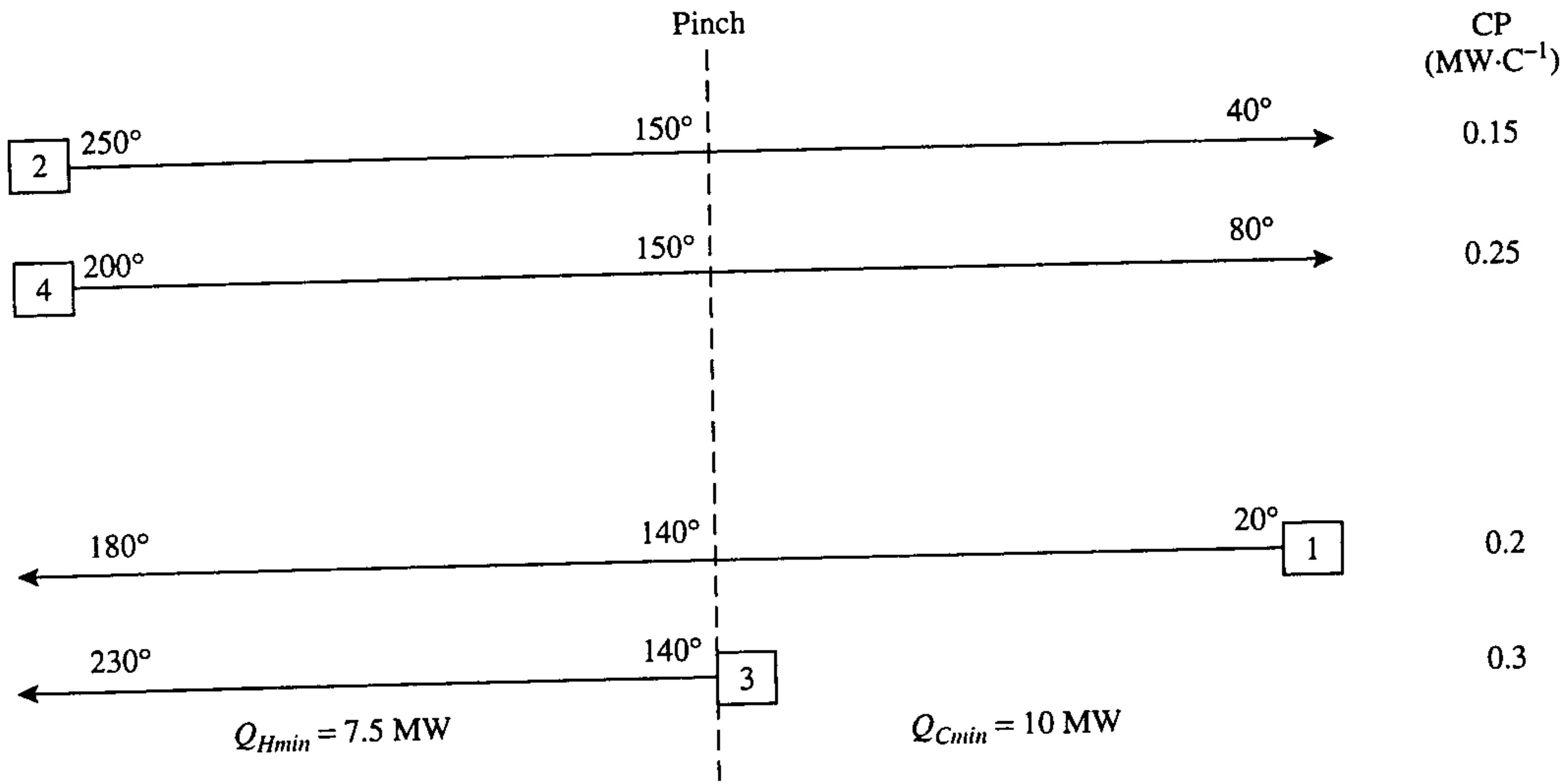


Figure 17.3 To determine the network area the balanced composite curves are divided into enthalpy intervals.

Summary - Network Design The pinch Design Method

1. Start at the pinch
2. The CP inequality for individual match
3. Provide CP Table
4. Tick-off Heuristic

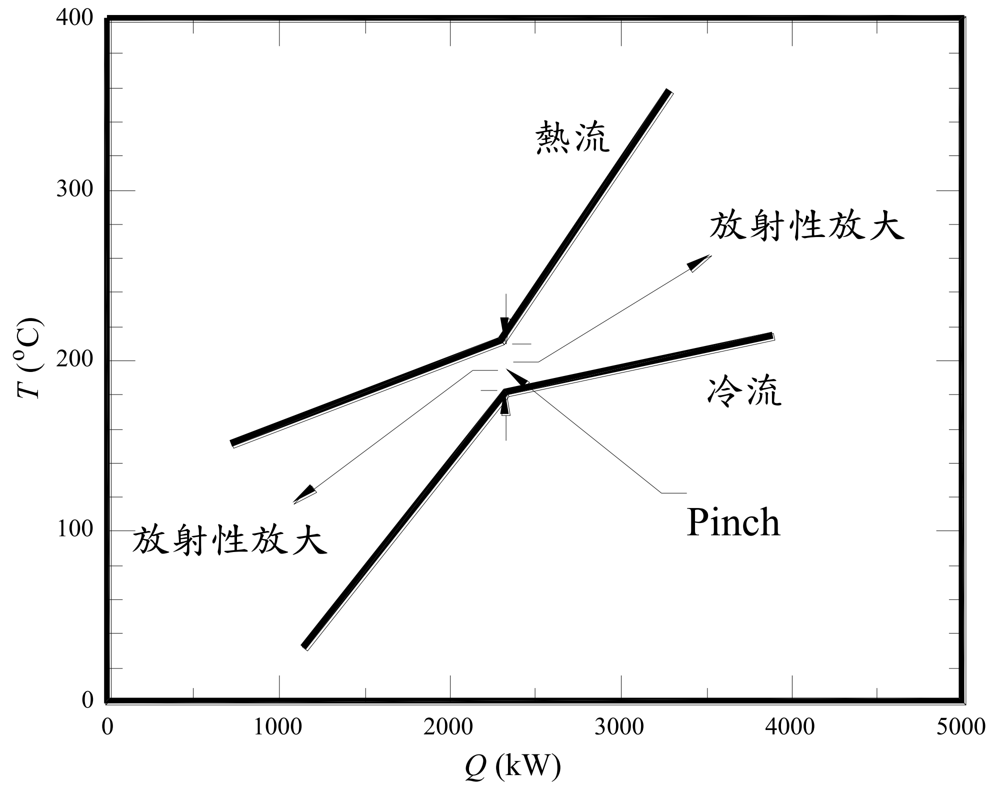
1. Start at the Pinch (10°C ΔT)

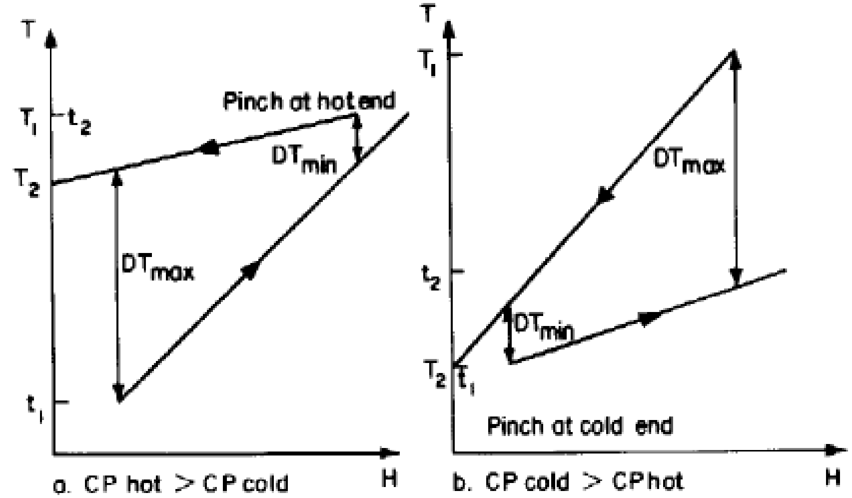


The grid diagram for the data from the Table 16.2.

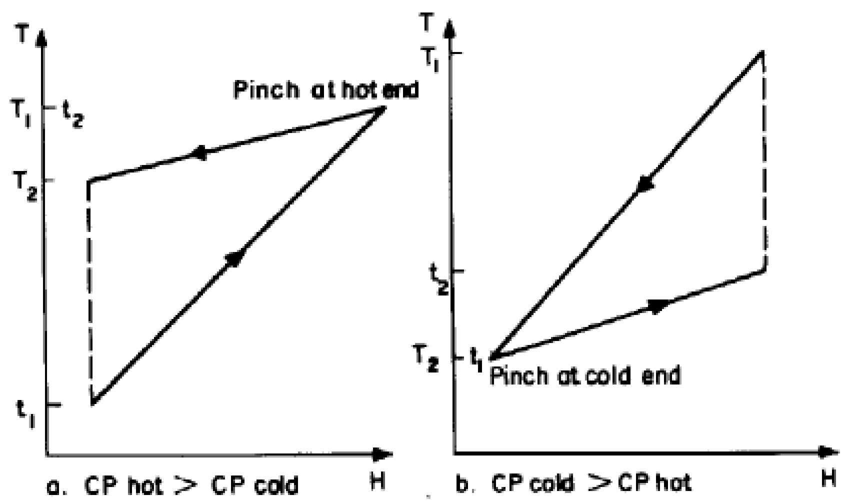
2. CP Inequality

- $CP_H \leq CP_c$ (above pinch)
- $CP_H \geq CP_c$ (below pinch)



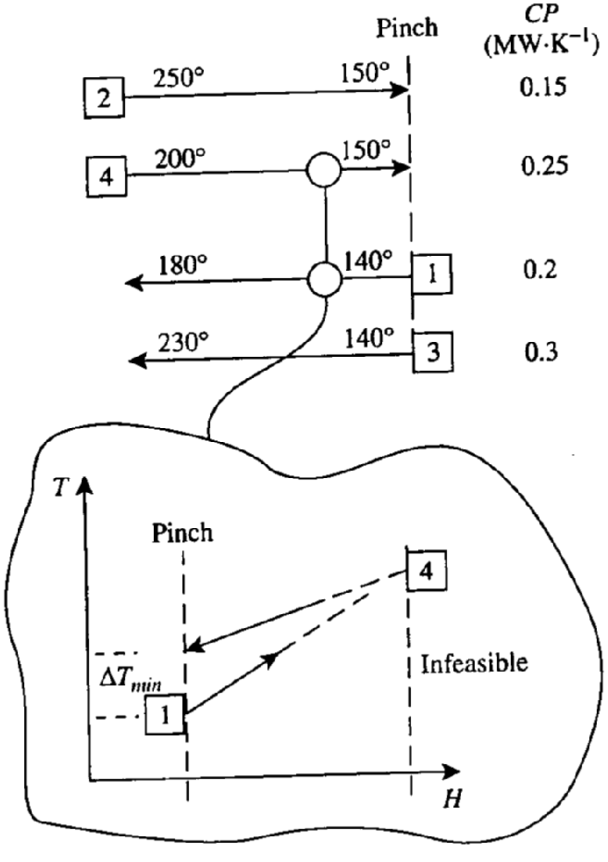


Sketch 4.6 Maximum energy recovery with a real DT_{min}

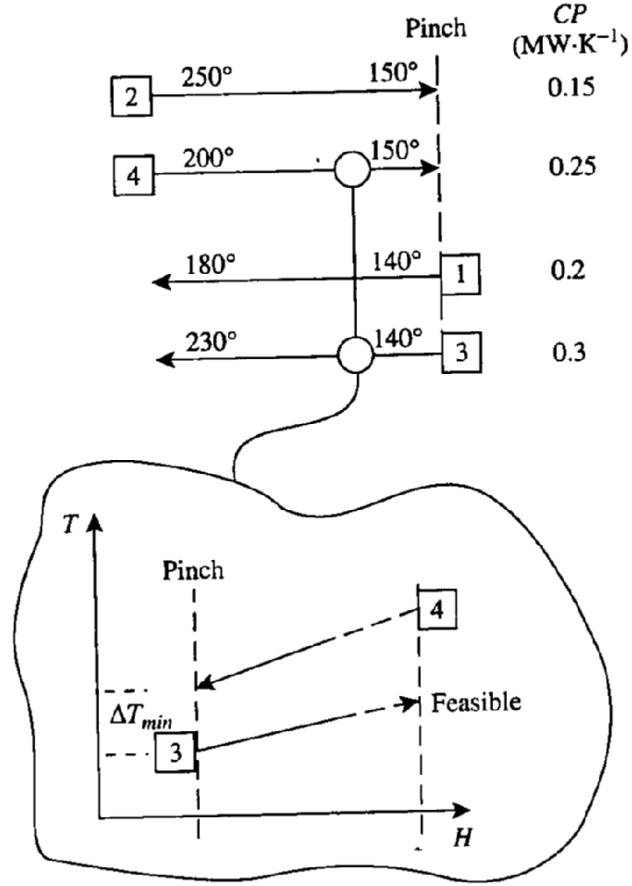


Sketch 4.5 Limiting heat transfer cases showing maximum energy recovery

3. CP Table
Criteria for pinch match above the pinch

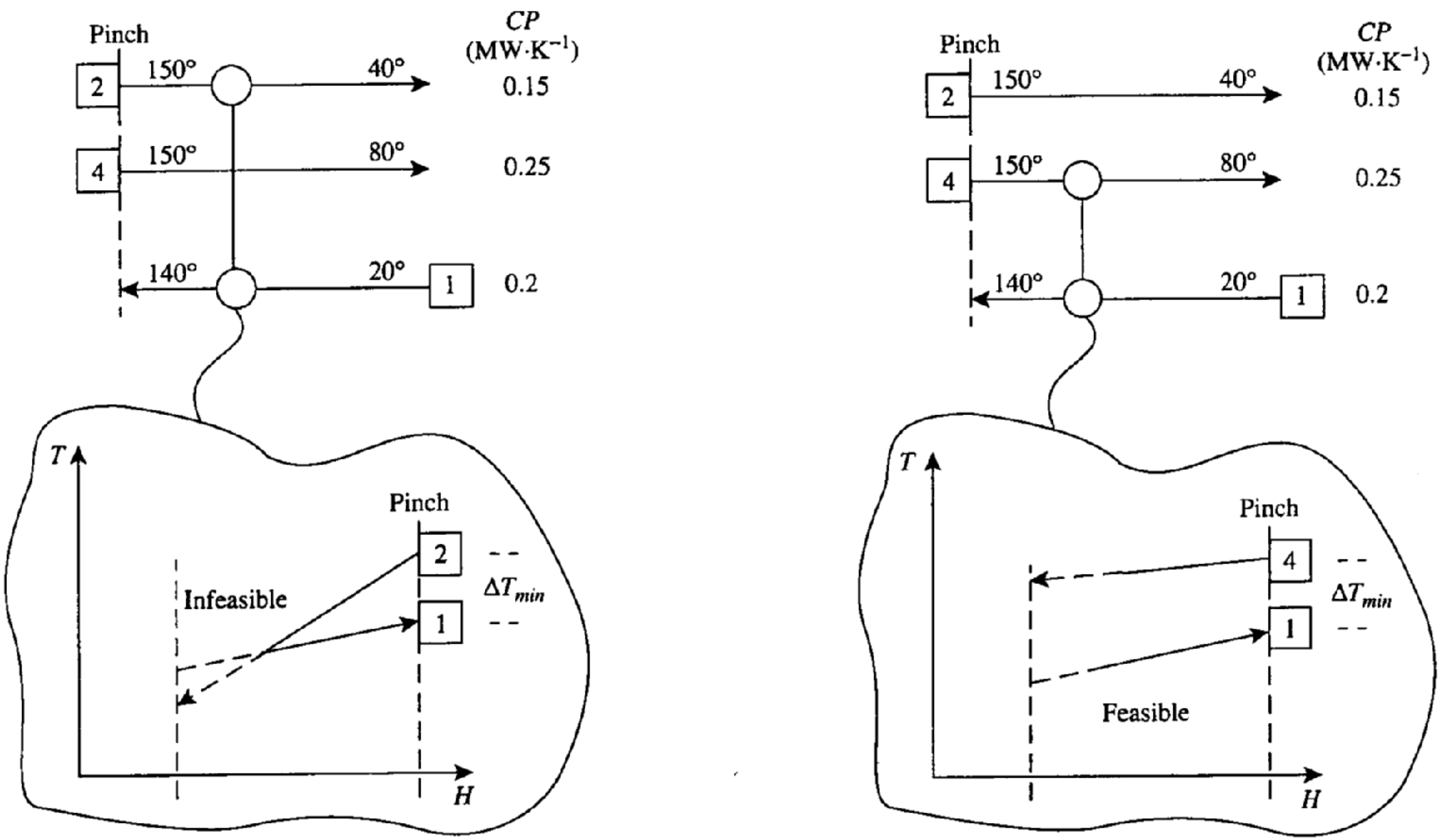


(a) Match is infeasible.



(b) Match is feasible.

Criteria for pinch match below the pinch



(a) Match is infeasible.

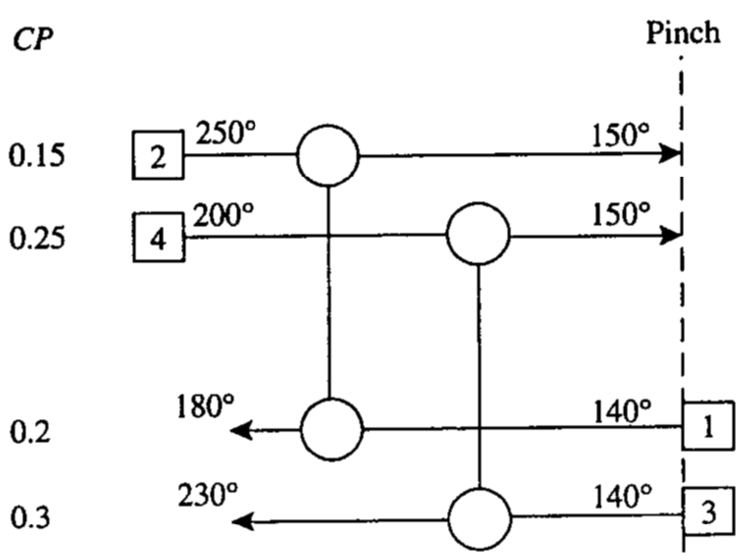
(b) Match is feasible.

CP Table (Conti..)

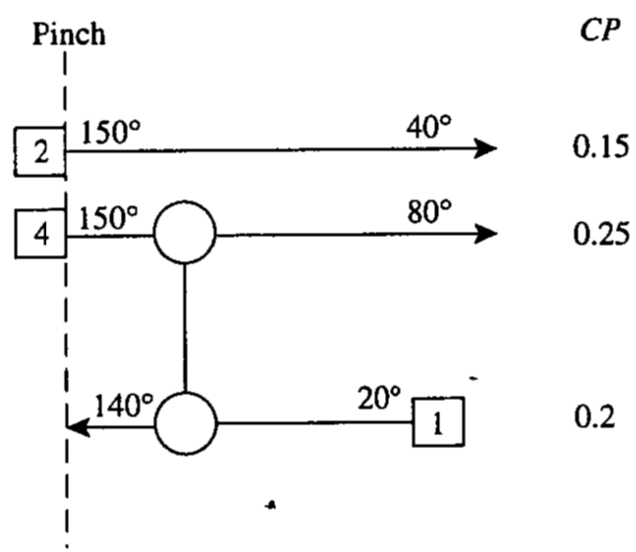
CP's in descending order

$CP_H \le CP_C$	
0.25	0.3
0.15	0.2

$CP_H \ge CP_C$	
0.25	0.2
0.15	



(a) Above pinch.



(b) Below pinch.

The CP table for the designs above and below the pinch for the problem from Table 16.2.

4. Tick-off Heuristic

- Once the matches around the pinch have been chosen ton satisfy the minimum energy, one important criterion is keeping the number of units minimum.
- To tick off a stream, individual units are made as LARGE as possible.
- Again, cooling water is NEVER used above pinch.

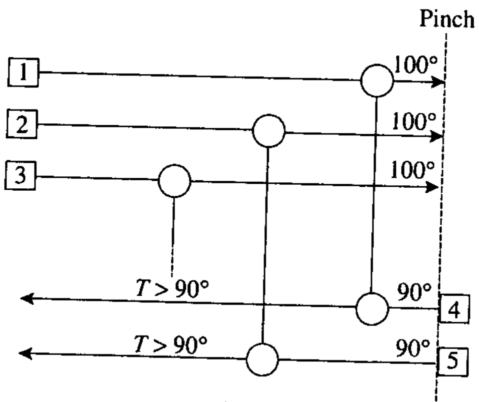
Stream Splitting

- Occasionally, appropriate match seems unlikely. For instance, if the number of hot stream is greater than the cold stream.
- Hence, an additional stream number must be fulfilled with splitting to meet:

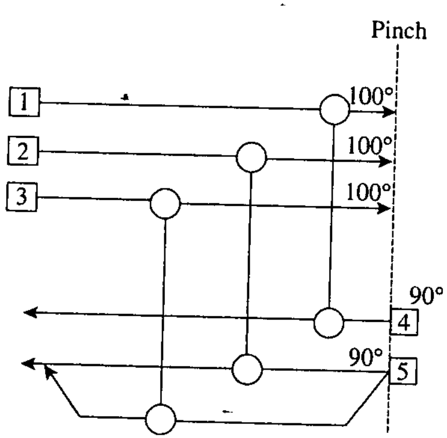
$S_H \leq S_c$ (above pinch)

$S_H \geq S_c$ (below pinch)

Stream splitting (Conti..)



(a)



(b)

Figure 18.12 If the number of hot streams at the pinch, above the pinch, is greater than the number of cold streams, then stream splitting of the cold streams is required.

Stream splitting ($CP_H > CP_c$, above pinch) or ($CP_H < CP_c$, below pinch)

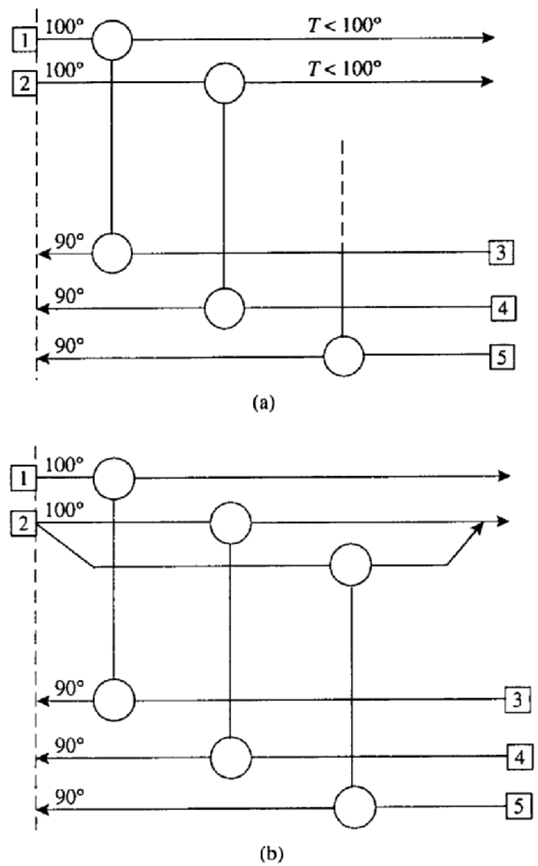


Figure 18.13 If the number of cold streams below the pinch, at the pinch, is greater than the pinch number of hot streams, then stream splitting of the hot steam is required.

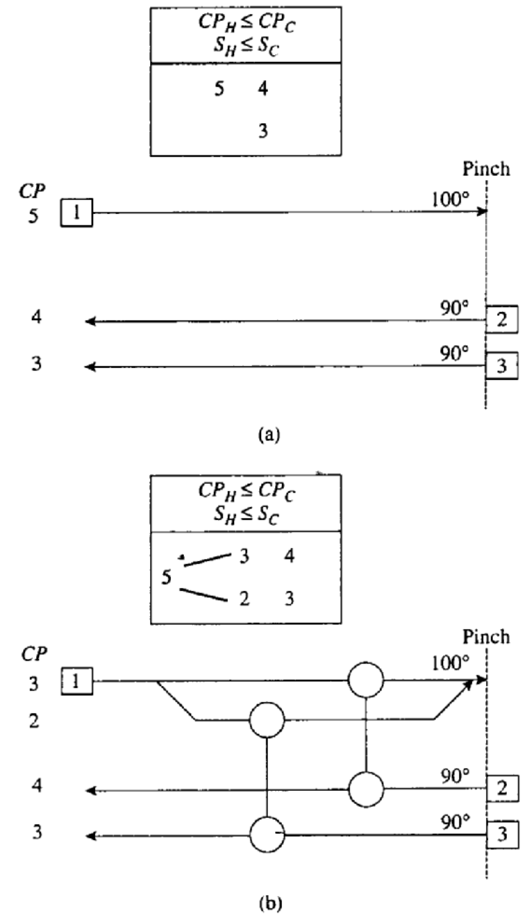


Figure 18.14 The CP in equality rules can necessitate stream splitting above the pinch.

Stream splitting (Conti..)

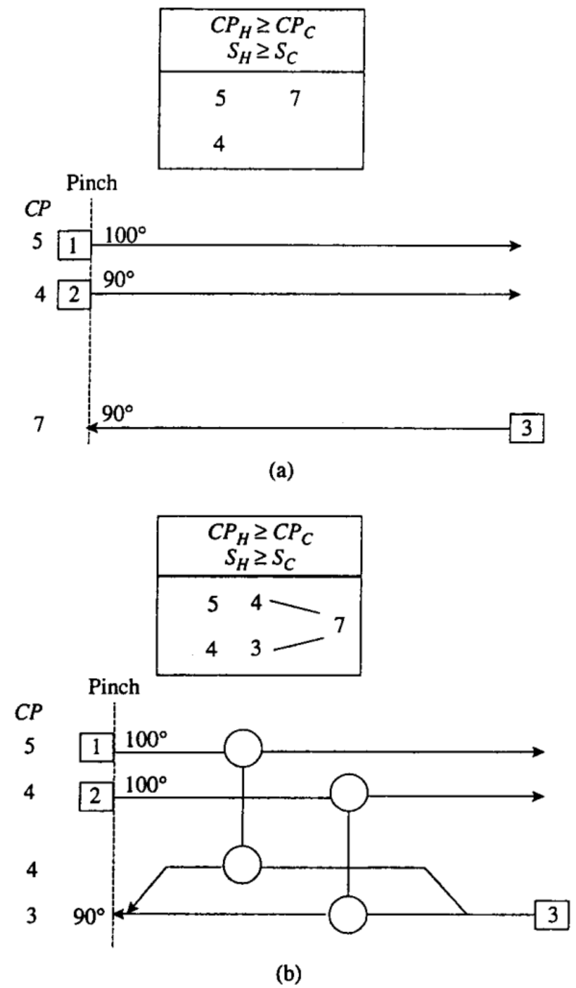
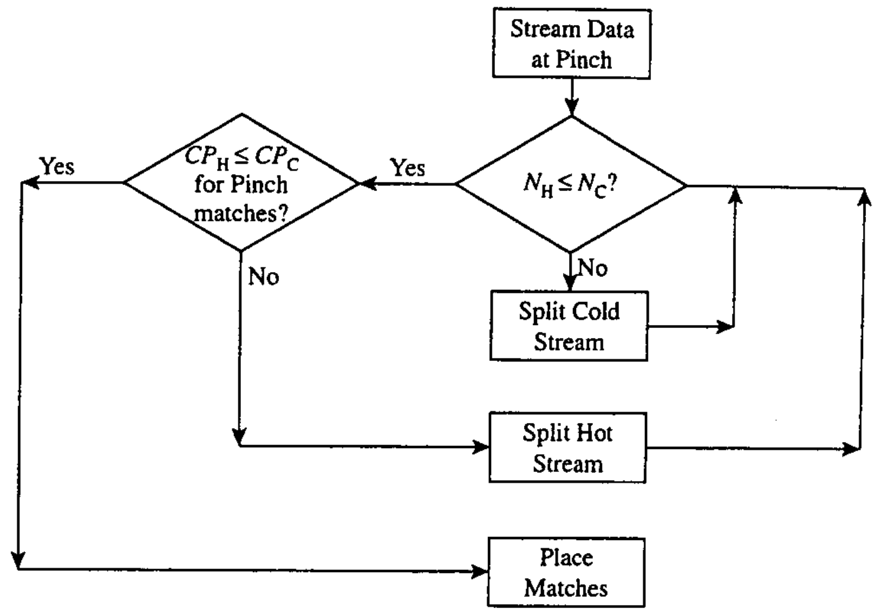
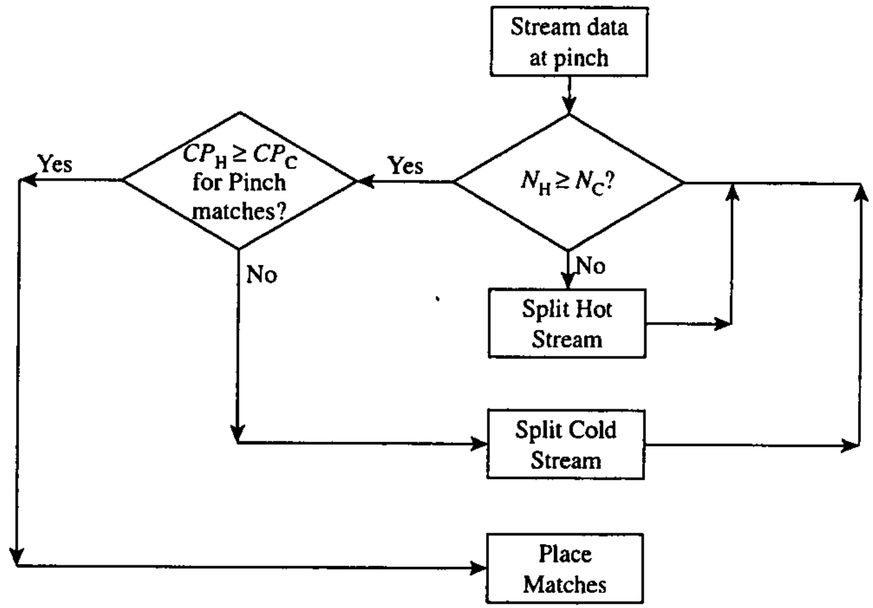


Figure 18.15 The CP in equality rules can necessitate stream splitting below the pinch.

Stream splitting algorithm



(a) Above the pinch.



(b) Below the pinch.

- First step – construct the problem table, composite curve and the like.
- Design the heat exchanger network

Example Heat exchange stream data (Pinch = 30 °C)

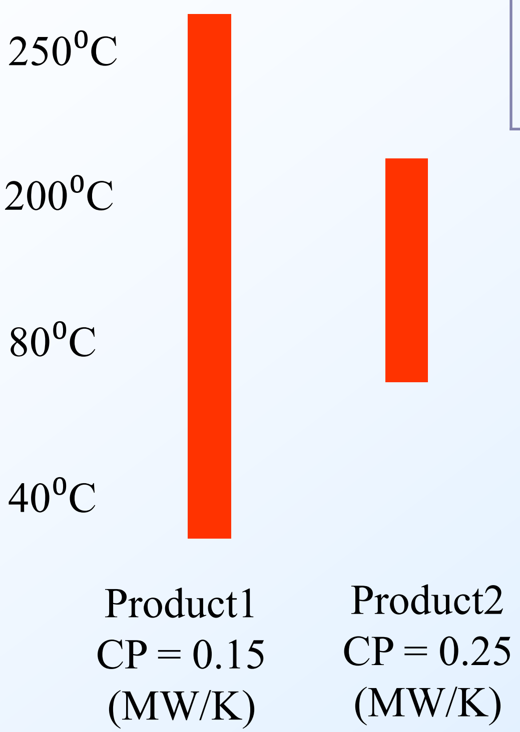
Stream	Type	Supply temp(°C)	Target temp(°C)	ΔH (MW)	CP(MW/K)
Reactor 1 feed	Cold	20	180	32	0.2
Reactor 1 product	Hot	250	40	-31.5	0.15
Reactor 2 feed	Cold	140	230	27	0.3
Reactor 2 product	Hot	200	80	-30	0.25

According to above the process data, try to solve this problem and plot the grand composite curve.

Example - Heat exchange stream data

Solve this problem

Hot Stream



Stream	Supply temp(°C)	Target temp(°C)	CP(MW/K)
Product 1	250	40	0.15
Product 2	200	80	0.25

$Q = CP\Delta T = 0.15*(250 - 200) = 7.5$ MW

$Q = CP\Delta T = (0.15+0.25)*(200 - 80) = 48$ MW

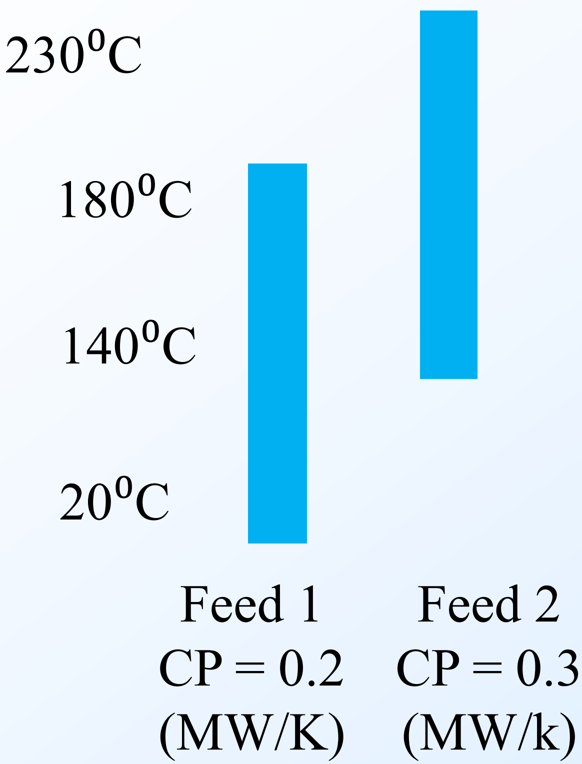
$Q = CP\Delta T = 0.15*(80 - 40) = 6$ MW

Example - Heat exchange stream data

Solve this problem

Cold Stream

Stream	Supply temp(^{o}C)	Target temp(^{o}C)	CP(MW/K)
Feed 1	20	180	0.2
Feed 2	140	230	0.3



$Q = CP\Delta T = 0.3*(230 - 180) = 15$ MW

$Q = CP\Delta T = (0.2+0.3)*(180 - 140) = 20$ MW

$Q = CP\Delta T = 0.2*(140-20) = 24$ MW

Example - Heat exchange stream data

Assumption: ΔTmin = 30 ^{0}C

THot stream - ΔTmin / 2

250^0C	235^0C
200^0C	185^0C
80^0C	65^0C
40^0C	25^0C

TCold stream + ΔTmin / 2

230^0C	245^0C
180^0C	195^0C
140^0C	155^0C
20^0C	35^0C

Temperature	CP, Hot steam – CP, Cold steam	Heat Increment (MW)
245^0C		
	-0.3	-0.3*(245-235)= -3 MW
235^0C		
	+0.15 -0.3	-0.15*(235-195)= -6 MW
195^0C		
	+0.15 -0.3 -0.2	-0.35*(195-185)= -3.5 MW
185^0C		
	+0.25 +0.15 -0.3 -0.2	-0.1*(185-155)= -3 MW
155^0C		
	+0.25 +0.15 -0.2	-0.2*(155-65)= +18 MW
65^0C		
	+0.15 -0.2	-0.05*(65-35)= -1.5 MW
35^0C		
	+0.15	+0.15*(35-25)= +1.5 MW
25^0C		

Example - Heat exchange stream data

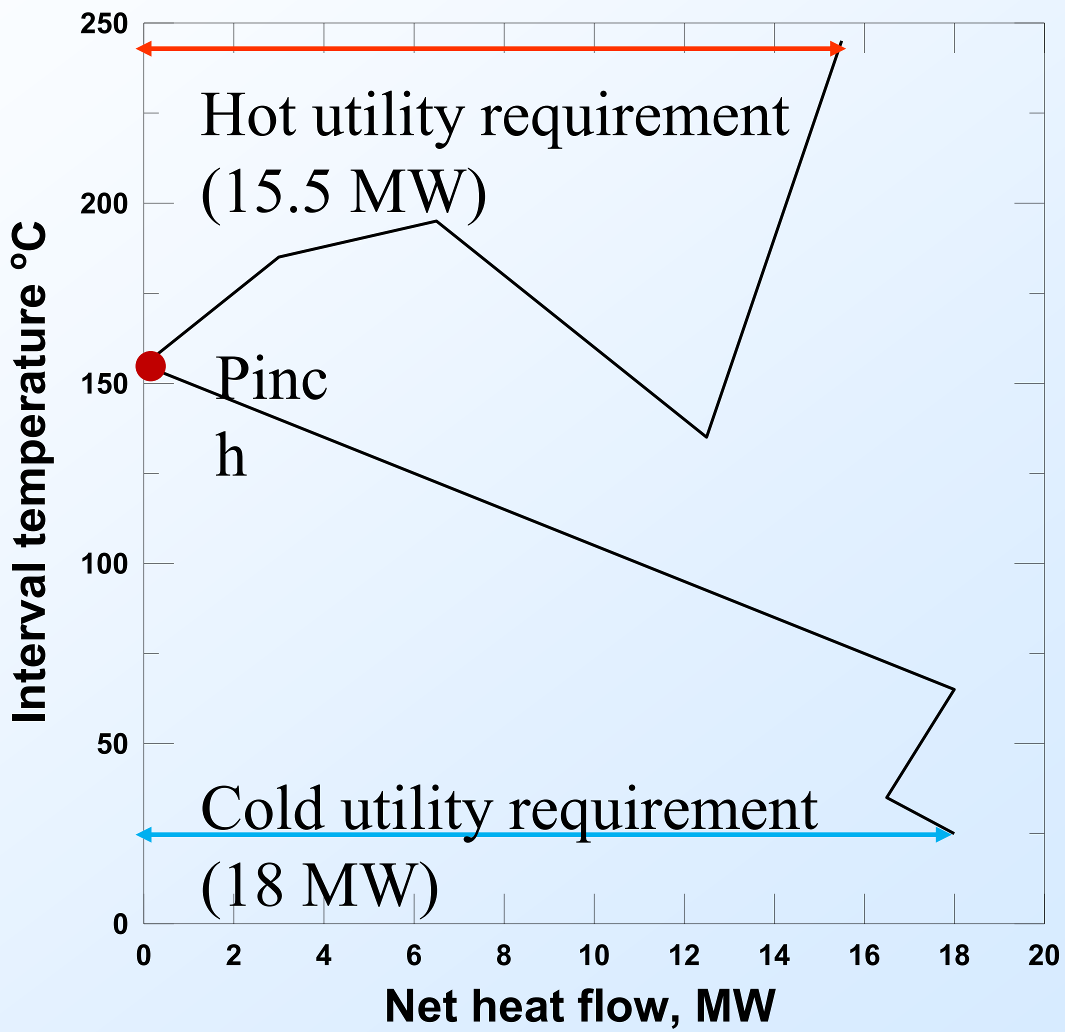
Continued

Total Heat Cascades

Temperature	Heat Increment (MW)	Infeasible	Final	
245^{0}C		0 MW	15.5 MW	
	-3 MW			
235^{0}C		-3 MW	12.5 MW	
	-6 MW			
195^{0}C		-9 MW	6.5 MW	
	-3.5 MW			
185^{0}C		-12.5 MW	3.5 MW	
	-3 MW			
155^{0}C		-15.5 MW	0 MW	**Pinch Temp.**
	+18 MW			
65^{0}C		+2.5 MW	18 MW	
	-1.5 MW			
35^{0}C		+1 MW	16.5 MW	
	+1.5 MW			
25^{0}C		+2.5 MW	18 MW	

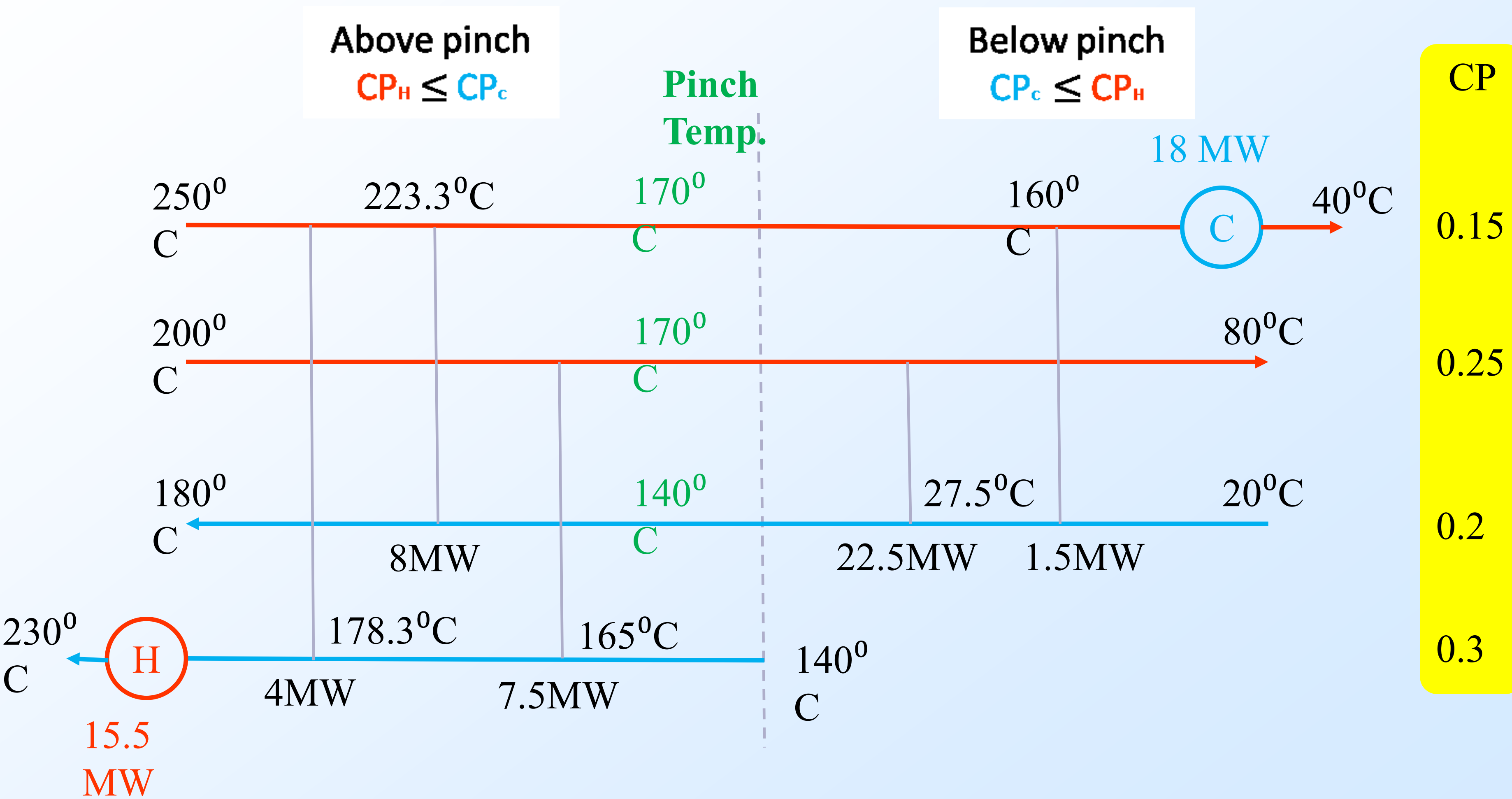
Example - Heat exchange stream data

Grand Composite Curve



Example - Heat exchange stream data

Network Design ⇒ $N_{units} = (N_{above\ pinch} - 1) + (N_{below\ pinch} - 1)$



Step 2, construct the HX network using Tick-off Heuristic)

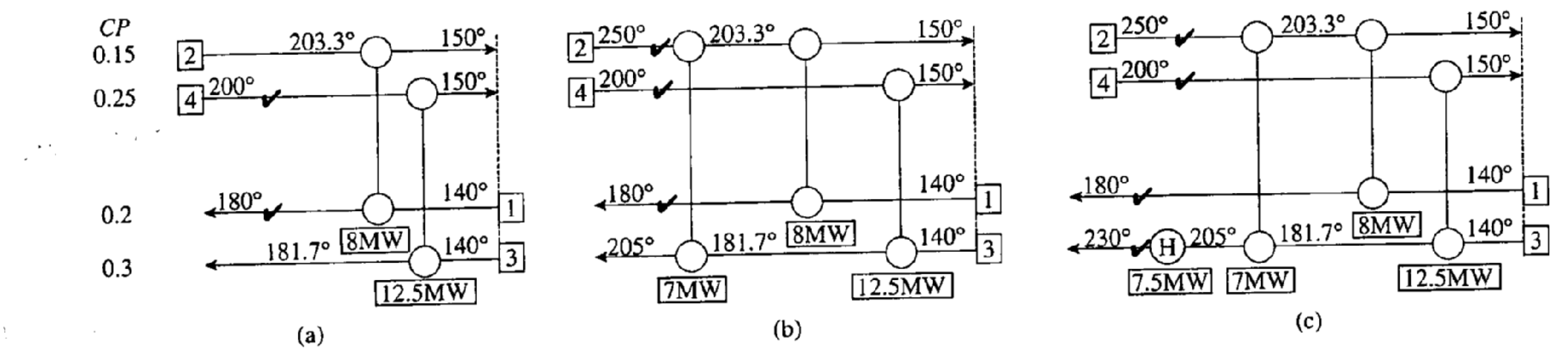


Figure 18.5 Sizing the units above the pinch using the tick-off heuristic.

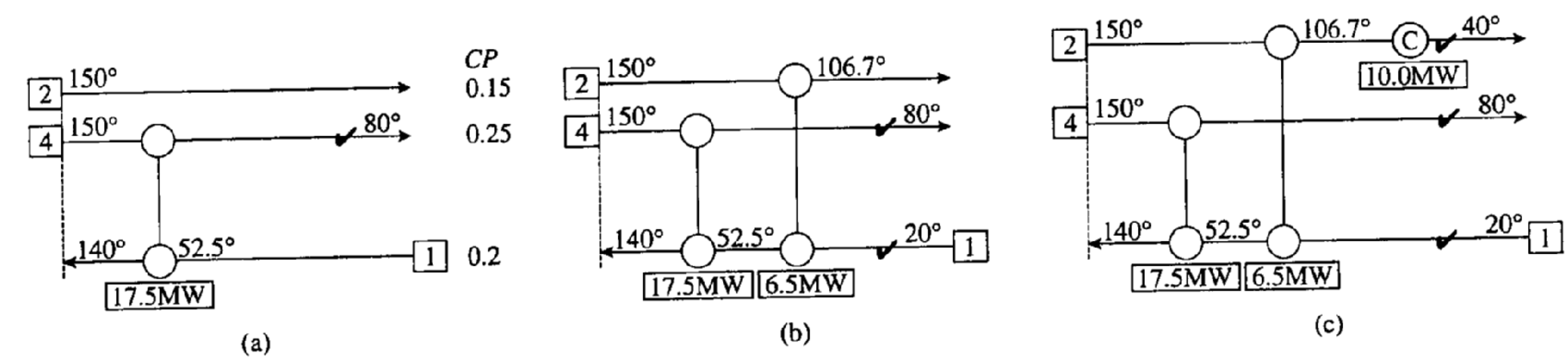


Figure 18.6 Sizing the units below the pinch using the tick-off heuristic.

Summary of Pinch Design Method

- Divide the problem at the pinch into separate problems.
- The design for separate problems is started at the pinch, moving away.
- Temperature feasibility requires CP to be satisfied.
- The loads on individual units are determined using the kick-off heuristic to minimize the number of units.
- Away from the pinch, more freedom in the choices.

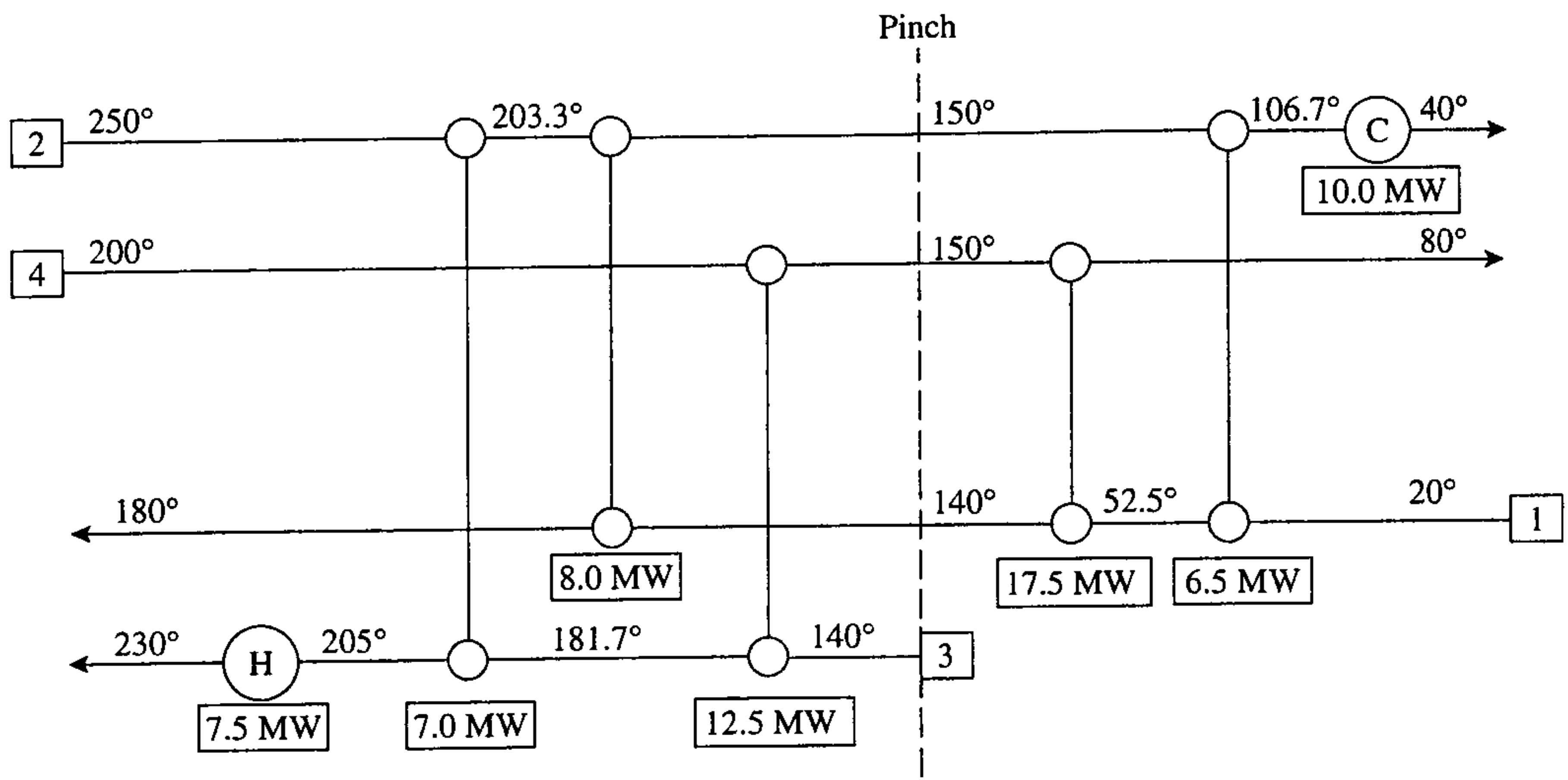
EXAMPLE

Example 18.1 The process stream data for a heat recovery network problem are given in Table 18.1.

Table 18.1 Stream data for Example 18.1.

Stream No.	Stream Type	Supply temperature (°C)	Target temperature (°C)	Heat capacity flowrate ($MW \cdot K^{-1}$)
1	Hot	400	60	0.3
2	Hot	210	40	0.5
3	Cold	20	160	0.4
4	Cold	100	300	0.6

Example (Conti..)



The completed design for the data from Table 16.2.

Example (Conti..)

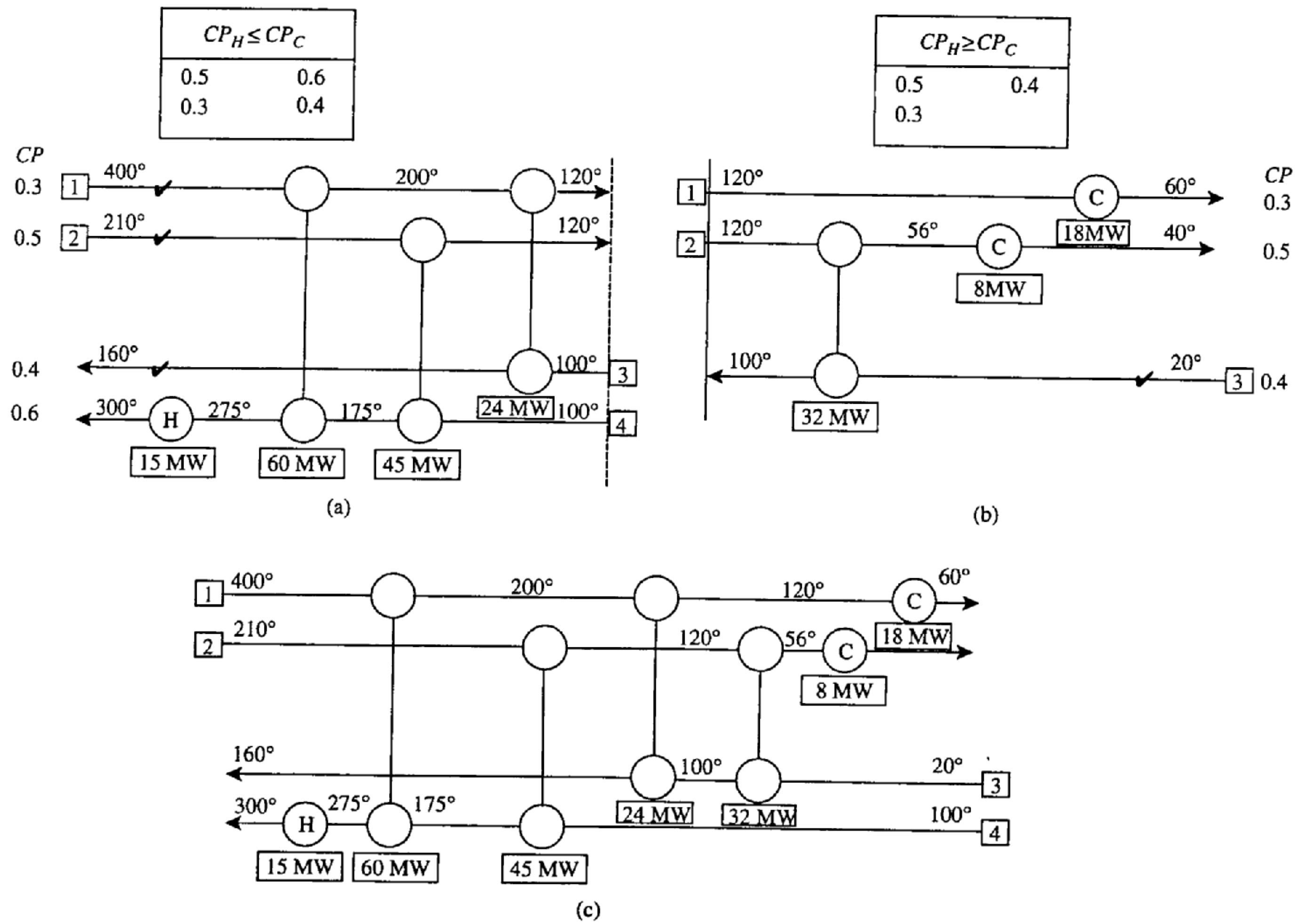


Figure 18.8 Maximum energy recovery design for Example 18.1.

Design for Threshold problem

- The design philosophy is to start the design it was “Constrained”.
- For threshold problem, it is at the “no-utility end”
- It acts like a “half” of a pinch problem.

Design for Threshold problem

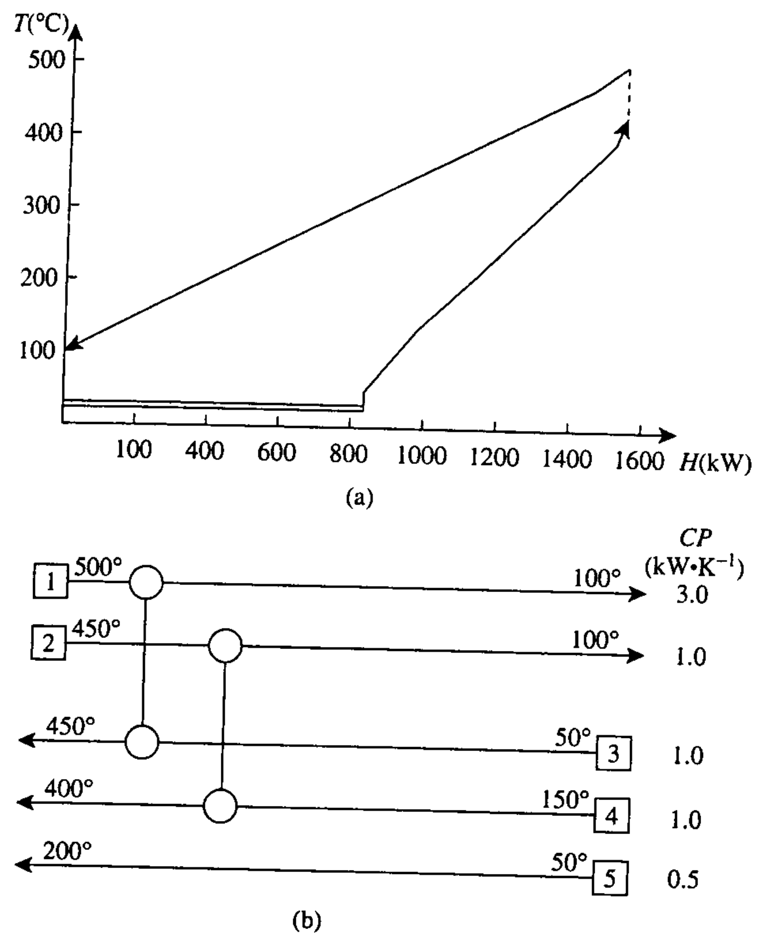


Figure 18.10 Even though threshold problems have large driving forces, there are still often essential matches to be made, especially at the no-utility end.

結語

- 製程上的熱交換器相當的多，因此如何有效安排熱源與冷源也就非常的重要，因為其中不乏可以回收的熱源與冷源、節約能源的目的就在於把回收做到最大而讓外加的冷源與熱源降到最少，Pinch Technology的目的就在這裡。

End of 5th Talk,

Questions?