



常用製程熱交換器設計技術 II

(直接接觸熱交換、再生式熱交換應用 與熱回收)

王啟川, PhD, 國立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: 03-571212 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。
- 感謝簡國祥博士提供部分教材
- 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：

- 質量傳遞與直接接觸熱交換技術
- 空氣特性與空氣線圖
- 冷卻水塔
- 熱管原理與熱管熱交換器
- 再生式熱交換器簡介與填料特性
- 再生式熱交換器熱流設計與案例



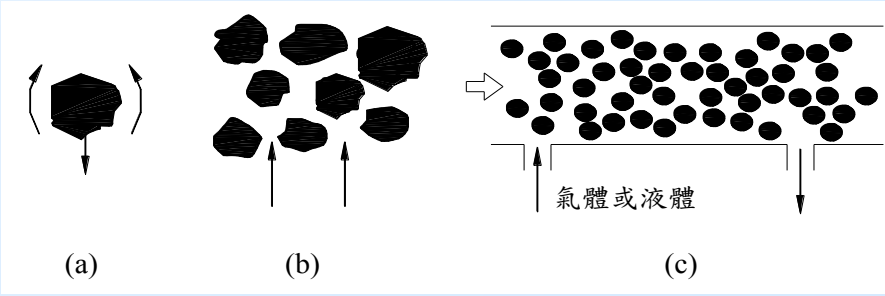
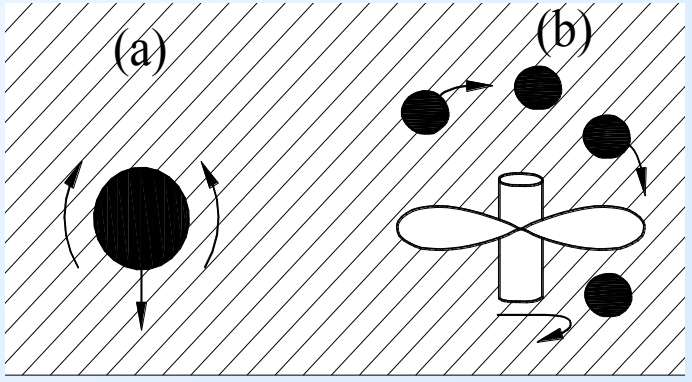
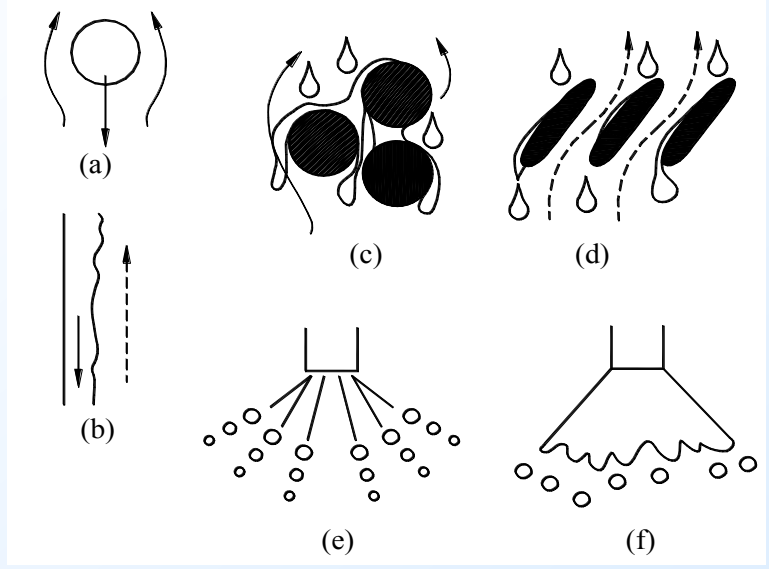
直接接觸熱交換

- 直接熱傳遞乃冷熱流體藉由直接接觸來交換熱量
 - 可藉由流體的直接接觸傳達大量的熱量，其單位體積與單位溫差的有效熱傳量，一般而言都遠比回復式熱交換器來的大。
 - 結構與製作較為簡單，可大幅降低起始投資成本。
 - 可避免使用或減少傳統金屬的使用量，降低腐蝕、結垢與沉積沉澱的問題。
 - 同時可應用在高溫流體中，由於不使用金屬，熱應力的影響也相對減少。
 - 製作成本較低，而且直接接觸效率高，流體流動壓降也較小，運轉成本也較低。



直接接觸熱交換

- 氣體與液體的直接接觸
- 無法混合的液體與液體的直接接觸
- 固體與流體或固體與氣體的直接接觸

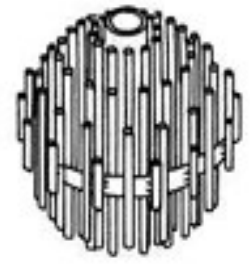




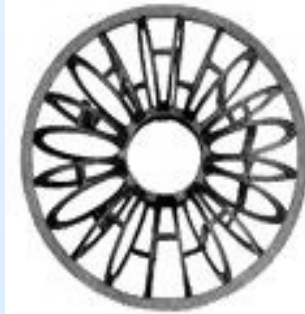
常見的各種填料 (www.ces.clemson.edu)



(a) Raschig ring (b) Berl Saddlen (c) Pall Ring (d) Intalox Saddle



(e) Interpak (f) Super Saddle (g) Toppak (h) Hedgehog

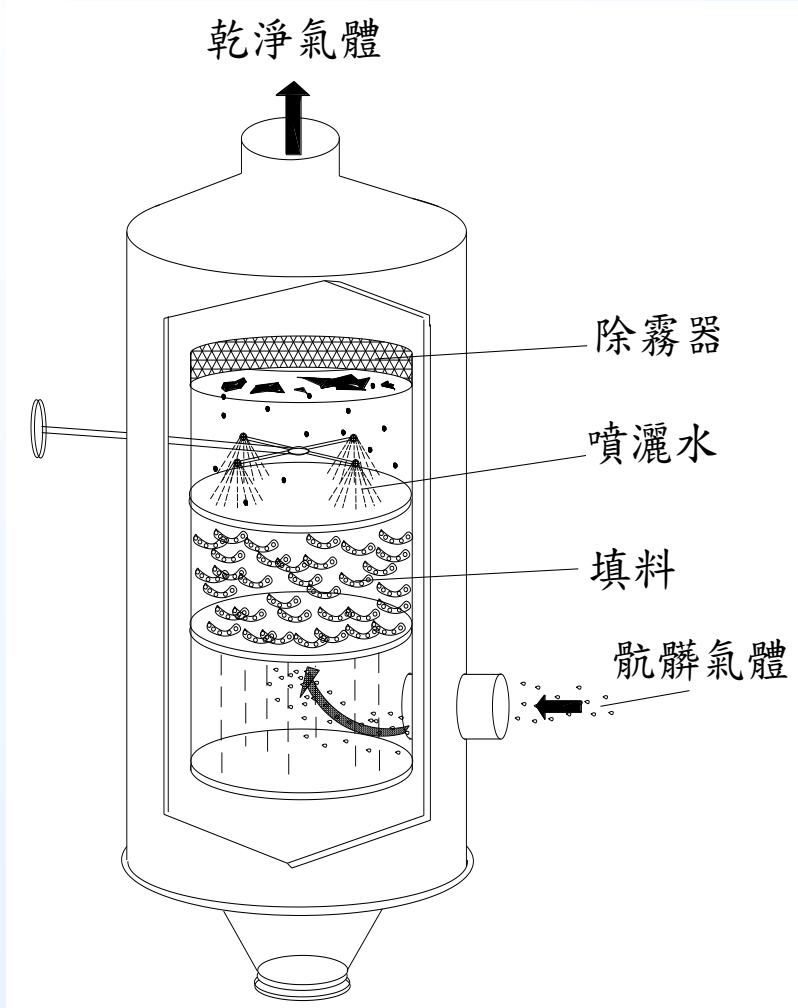


(i) Tellerette

填料的主要功用：
增加接觸時間
與面積，通常
並不扮演熱傳
角色！

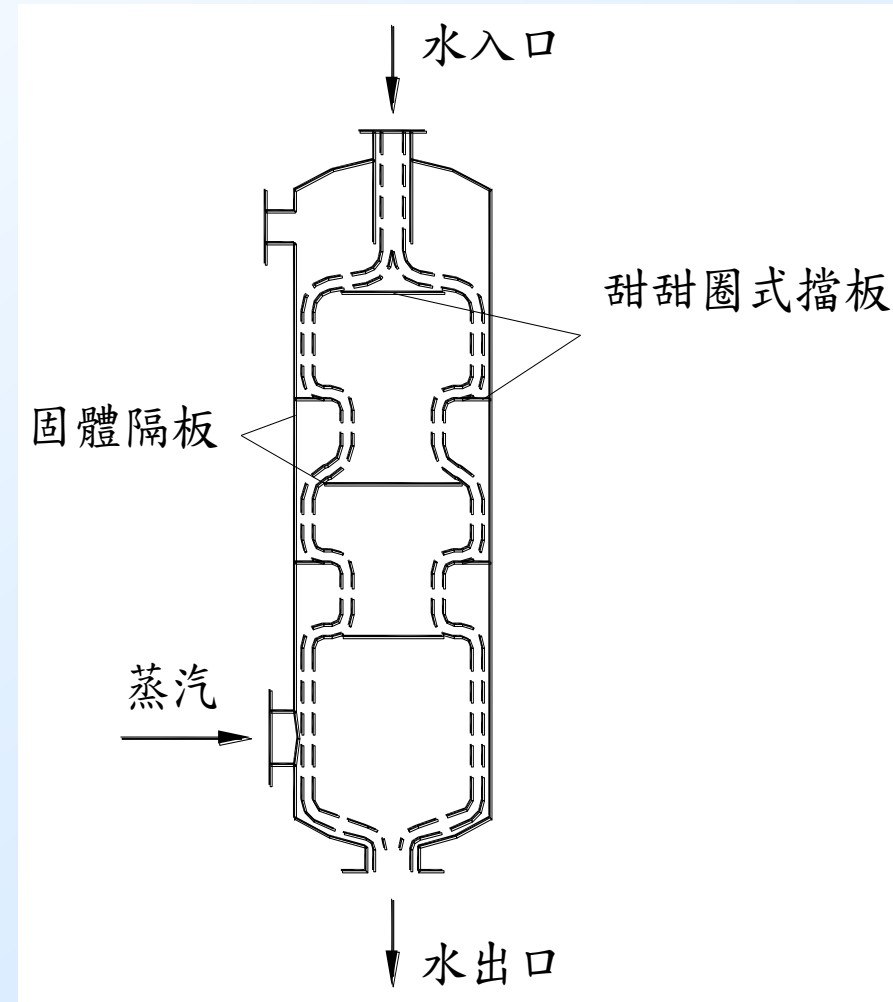


噴霧/氣泡塔槽 (Spray/Bubble columns)



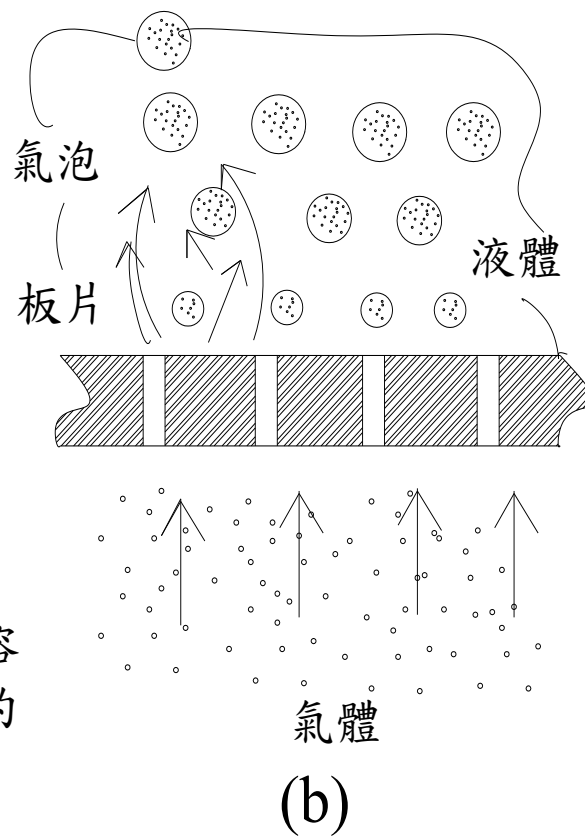
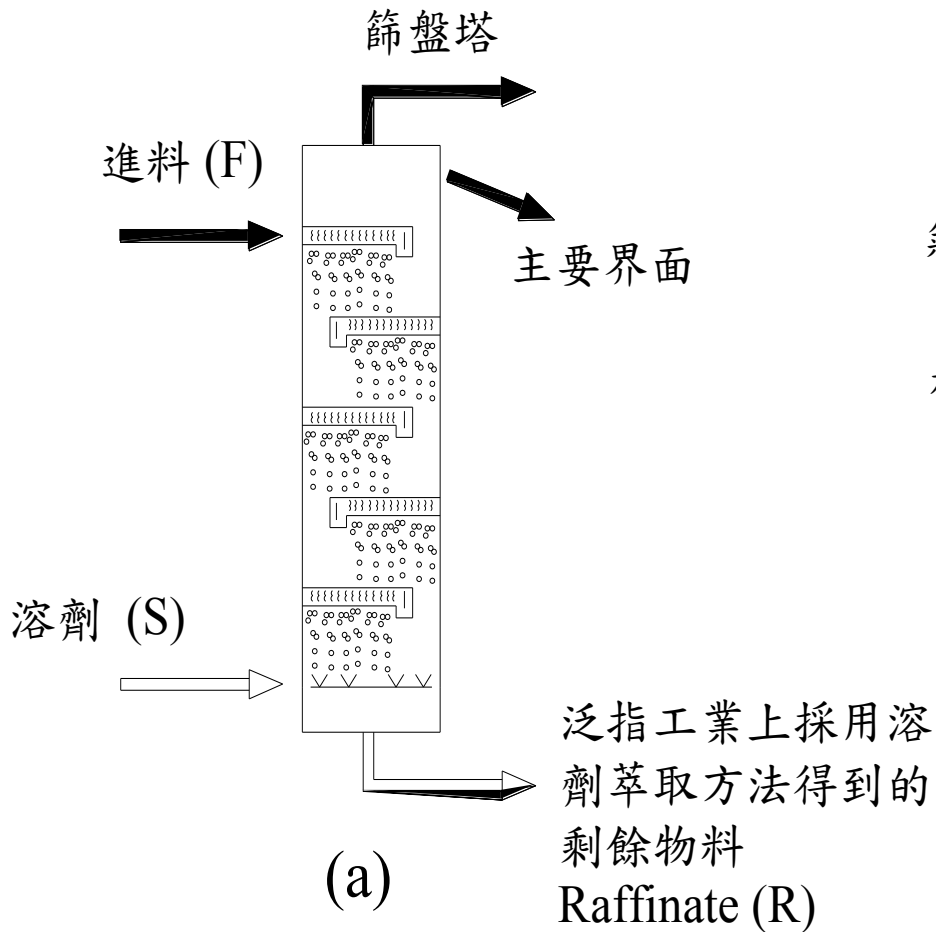


擋板式塔槽



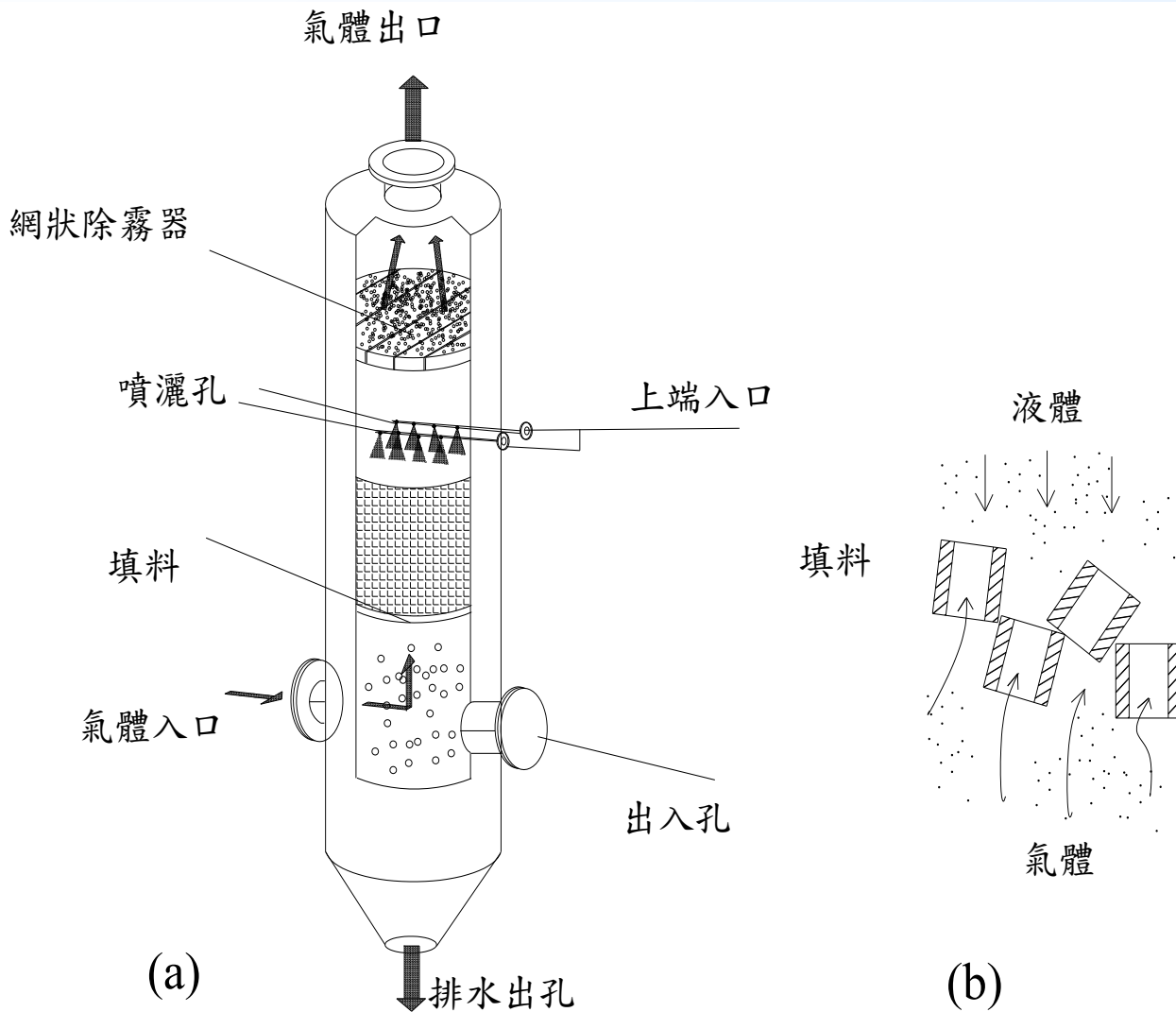


典型篩盤塔塔槽



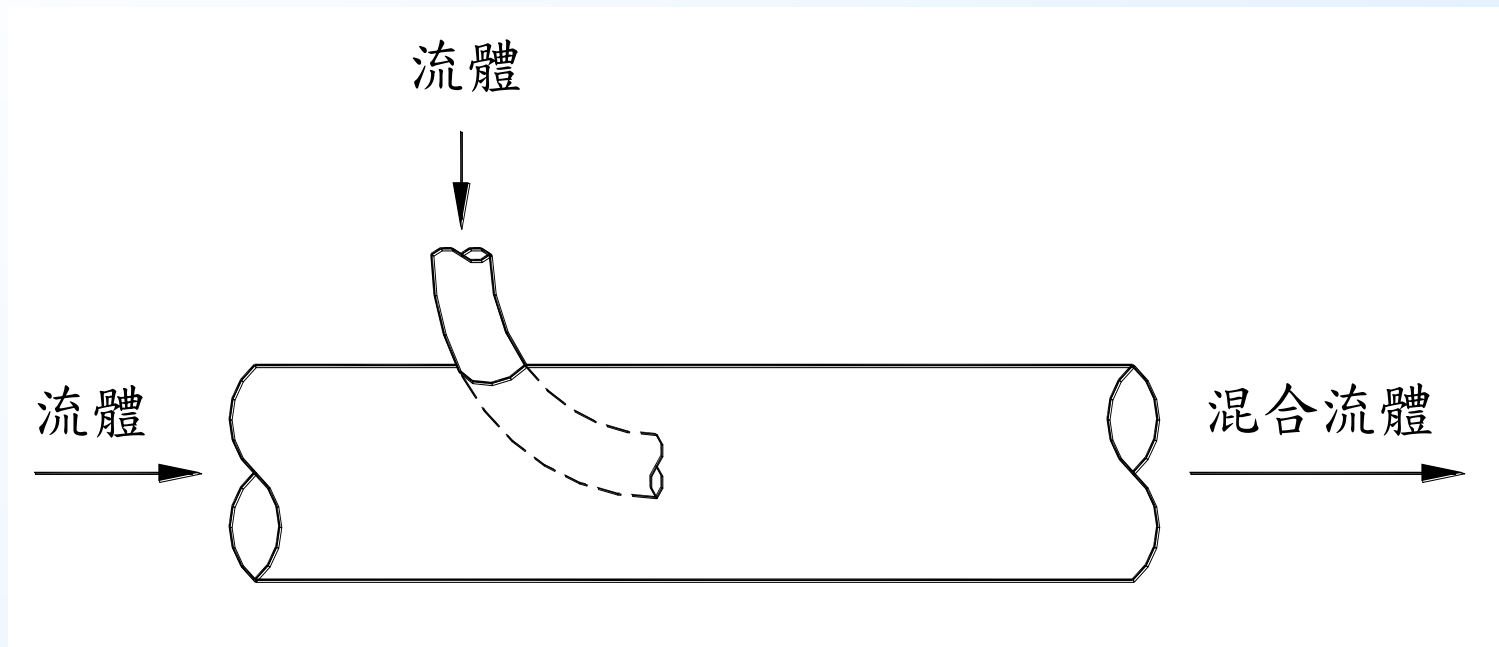


填充塔 (Packed columns)



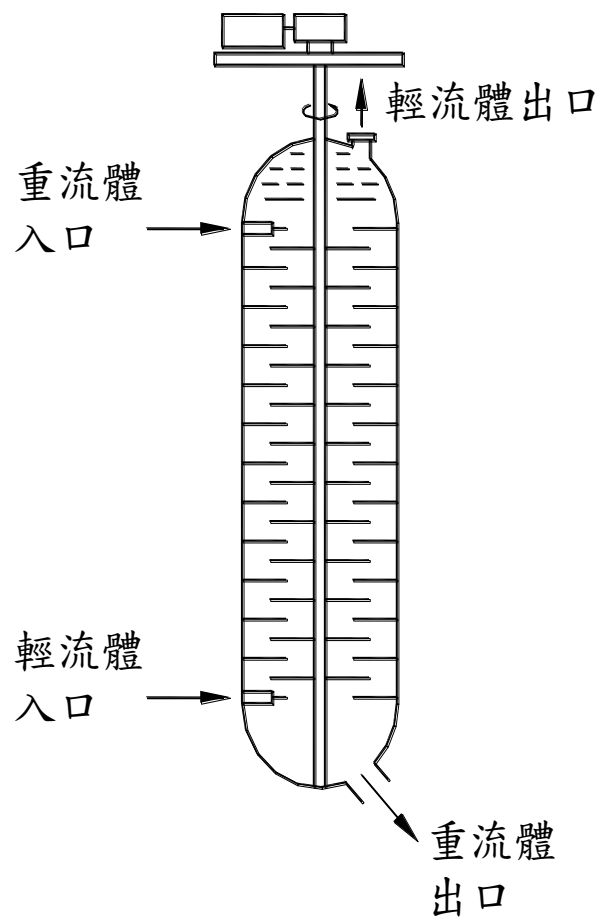


管路接觸 (Pipeline contactors)

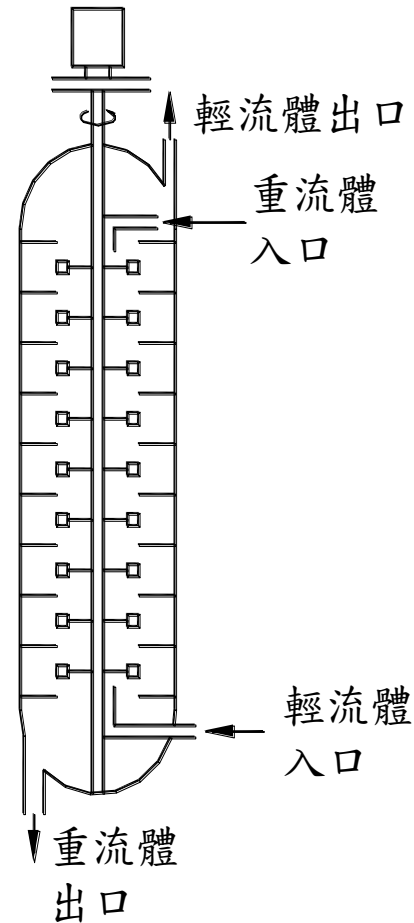




機械擾動接觸 (Mechanically agitated contactors)



ROTATING DISK CONTACTOR



"LIGHTNIN" TYPE CONTACTOR



直接接觸熱交換的熱流分析 (wo mass transfer)

$$dA = adV = aAdz \quad (16-1)$$

$$q = h_G(T_G - T_i) = h_L(T_i - T_L) = U(T_G - T_L) \quad (16-2)$$

$$\frac{1}{U} = \frac{1}{h_G} + \frac{1}{h_L} \quad (16-3)$$

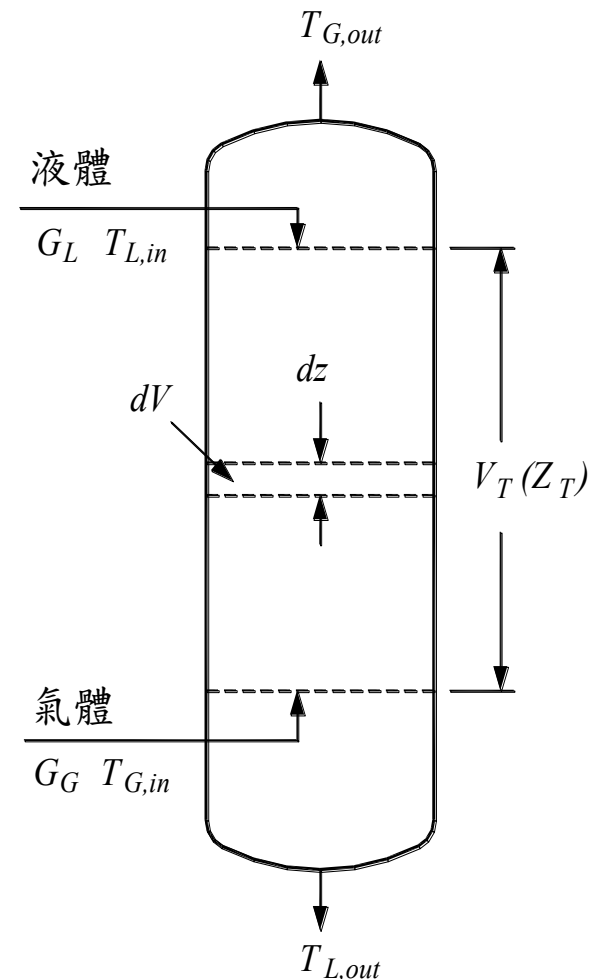
$$dQ = qdA = qaAdz = Ua(T_G - T_L)Adz \quad (16-4)$$

總熱傳量的獲得，可由積分式16-4如下：

$$Q = \int_0^{Z_T} dQ = \int_0^{Z_T} Ua(T_G - T_L)Adz = Ua \int_0^{Z_T} (T_G - T_L)Adz \quad \{\text{假設}U\text{為定值}\} \quad (16-5)$$

由平均值定理，上式可改寫如下：

$$Q = Ua \int_0^{Z_T} (T_G - T_L)Adz = Ua\Delta T_m \int_0^{Z_T} Adz = Ua\Delta T_m V_T \quad \{\text{假設截面積}A\text{為定值}\}$$





熱傳經驗方程式

一般研究歸納的經驗式如下：

$$h_L a = C_1 G_G^{m_1} G_L^{n_1} \quad (16-7)$$

$$h_G a = C_2 G_G^{m_2} G_L^{n_2} \quad (16-8)$$

其中 G_L 與 G_G 分別代表液體與氣體的質量通率， C_1 、 C_2 、 m_1 、 m_2 、 n_1 、 n_2 為測試資料經迴歸分析後的經驗常數，這些經驗常數與填料、塔槽大小、高度有關，而且與所使用的單位有關，讀者在使用上要特別注意；然而在實際應用上，由於界面面積甚難決定，因此氣液個別相的熱傳係數很難決定，因此也有許多研究直接將數據都整理成單位體積之總熱傳係數：

$$Ua = CG_G^m G_L^n \quad (16-9)$$

PS: 有因次單位的經驗式!!



例：一填充塔($D = 0.305$ m)利用水來冷卻熱空氣，熱空氣的入口溫度為350 K流量為0.06 kg/s，水的流量為0.15 kg/s而入口溫度為290K，下表為各種填料的特性資料；試問利用不同的填料來設計塔槽的高度 Z_T 以符合空氣出口溫度為295 K的需求，假設空氣的 $c_p = 1007$ J/kg·K而水的 $c_p = 4180$ J/kg·K。

填料	尺寸 (m)	熱傳係數	C	m	n
Rasching rings	0.025	h_{La}	26,680	0.51	0.63
	0.025	h_{Ga}	6,947	1.10	0.02
Rasching rings	0.038	h_{La}	46,060	0.48	0.75
	0.038	h_{Ga}	6,130	1.45	0.16
Intalox saddles	0.025	h_{La}	36,960	0.20	0.84
	0.025	h_{Ga}	7,228	1.01	0.25
Intalox saddles	0.038	h_{La}	42,570	0.20	0.69
	0.038	h_{Ga}	6,174	1.38	0.10
Pall rings	0.025	h_{La}	33,460	0.45	0.87
	0.025	h_{Ga}	5,065	1.12	0.33
Pall rings	0.038	h_{La}	32,910	0.31	0.80
	0.038	h_{Ga}	5,310	1.28	0.26
HyPak rings	0.030	h_{La}	43,670	0.15	0.76
	0.030	h_{Ga}	8,150	0.99	0.18



首先，承襲以往計算熱交換的步驟，先計算幾何尺寸與一些入出口參數，再算出個別的熱傳係數 ha 與單位體積總熱傳係數 Ua ；再由 $Q = Ua\Delta T_m V_T$ ，算出體積 V_T ，最後由 $Z_T = V_T/A$ 算出塔槽高度。詳細流程如下：

(1) 計算塔槽截面積 $A = \pi D^2/4 = \pi \times 0.305^2/4 = 0.0731 \text{ m}^2$

(2) $G_L = \frac{\dot{m}_L}{A} = \frac{0.15}{0.0731} = 2.053 \text{ kg/m}^2 \cdot \text{s}$

(3) $G_G = \frac{\dot{m}_G}{A} = \frac{0.06}{0.0731} = 0.821 \text{ kg/m}^2 \cdot \text{s}$

(4) 塔槽之總散熱量與冷卻水之出口度：

$$Q = \dot{m}_G c_{p,G} (T_{G,in} - T_{G,out}) = 0.06 \times 1007 \times (350 - 295) = 3323.1 \text{ W}$$

$$\text{又 } Q = \dot{m}_L c_{p,L} (T_{L,out} - T_{L,in})$$

$$\therefore T_{L,out} = \frac{Q}{\dot{m}_L c_{p,L}} + T_{L,in} = \frac{3323.1}{0.15 \times 4180} + 290 = 295.3 \text{ K}$$



(5) 由於是逆向流動安排，塔槽之有效平均溫差 $\Delta T_m =$ 對數平均溫差

$$\begin{aligned}\Delta T_m = LMTD &= \frac{(T_{G,in} - T_{L,out}) - (T_{G,out} - T_{L,in})}{\ln\left(\frac{T_{G,in} - T_{L,out}}{T_{G,out} - T_{L,in}}\right)} \\ &= \frac{(350 - 295.3) - (295 - 290)}{\ln\left(\frac{350 - 295.3}{295 - 290}\right)} = 20.8^\circ\text{C}\end{aligned}$$

(6) 以 Raschig rings 而言，

$$h_L a = C_1 G_G^{m_1} G_L^{n_1} = 26680 \times 0.821^{0.51} \times 2.053^{0.63} = 37964$$

$$h_G a = C_2 G_G^{m_2} G_L^{n_2} = 6947 \times 0.821^{1.1} \times 2.053^{0.02} = 5675$$

由 $h_G a$ 與 $h_L a$ 的值可知氣側的值遠小於液體側的值，因此，此一熱質傳過程中，氣側為決定性側。因此如果要快速估算，可忽略液體側的阻抗。

$$(7) \quad Ua = \left(\frac{1}{\frac{1}{h_G a} + \frac{1}{h_L a}} \right) = \frac{1}{\frac{1}{5675} + \frac{1}{37964}} = 4937 \text{ W/m}^3$$

(8) 由 $Q = Ua \Delta T_m V_T$

$$\therefore V = \frac{Q}{Ua \Delta T_m} = \frac{3323.1}{4937 \times 20.8} = 0.0324 \text{ m}^3$$



質傳與熱傳的類比

$$\frac{h}{h_M \rho c_p} = Le^{2/3} \quad (16-19)$$

其中 h 為熱傳係數， h_M 為質傳係數， Le 為 Lewis number ($Le = \alpha/D_v = Sc/Pr$)，對空氣-水蒸氣系統而言，此值 $= 0.6/0.71 = 0.845$ ，所以：

$$\frac{h}{h_M \rho c_p} = 0.845^{2/3} = 0.894 \quad (16-20)$$

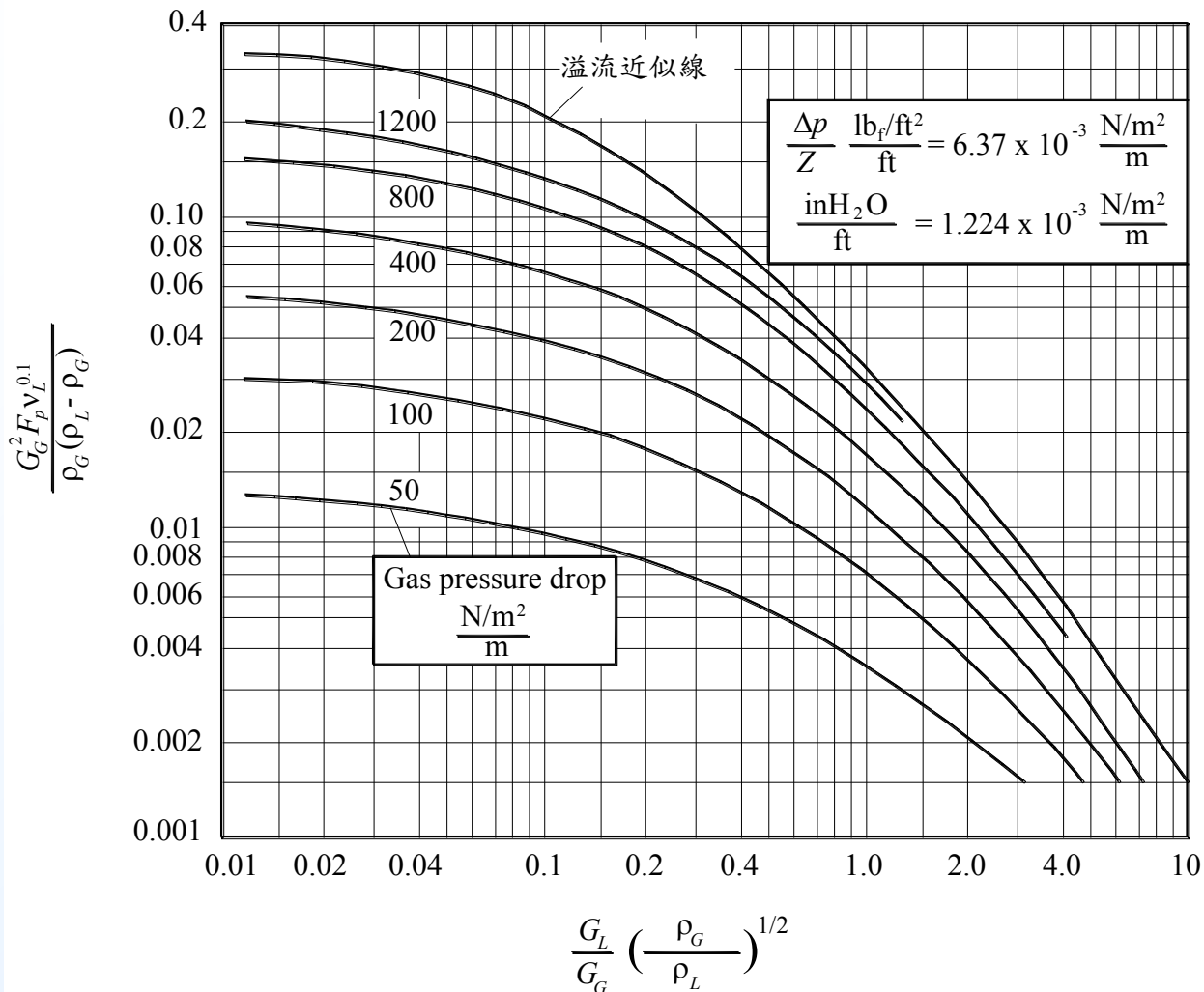
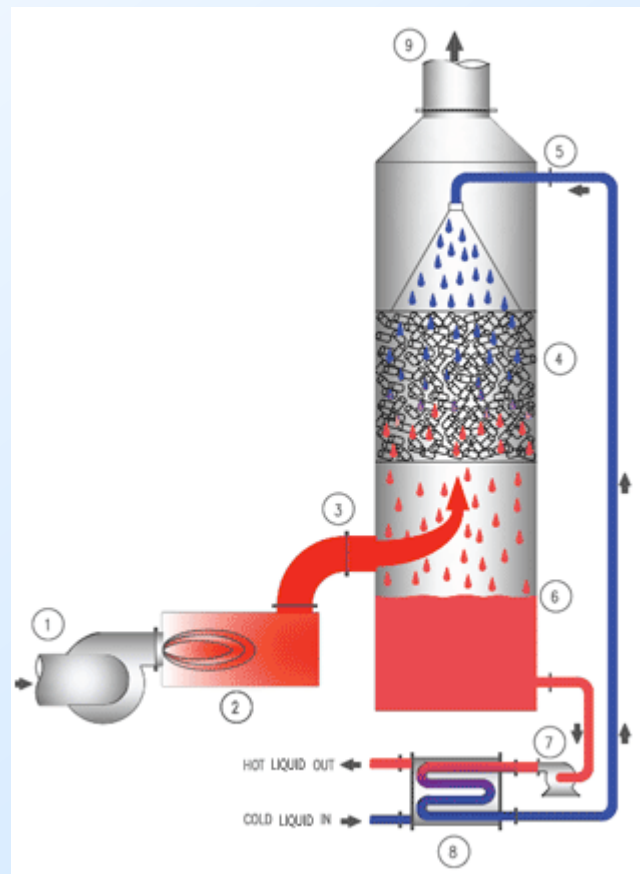


圖 16-12 溢流、壓降與 $\frac{G_L}{G_G} \sqrt{\frac{\rho_G}{\rho_L}}$ 、 $\frac{G_G^2 F_p v_L^{0.1}}{\rho_G (\rho_L - \rho_G)}$ 間的關係圖 (資料來源: Treybal, R.E., 1980, Mass-transfer operations)



直接接觸熱交換的熱流分析(有質傳效應)

在許多應用中，兩接觸的流體會夾雜複雜的質傳效應，例如冷凝、蒸發或沸騰等現象發生時，氣體與液體間除了顯熱交換(Q_s)外，因為質傳所帶出的潛熱熱傳(Q_l)會讓塔槽內的現象更為複雜





有質傳效應的情況可分四類

- 冷卻伴隨除濕的情況(但未完全冷凝)

$$\frac{1}{Ua} = \frac{1}{\gamma h_G a} \frac{Q_s}{Q} + \frac{1}{h_L a}$$

$$\phi = \frac{\dot{r}}{h} = \frac{\dot{r}}{h_G a V}, \quad \gamma = \frac{\phi}{e^\phi - 1} \quad (\text{Ackermann correction factor})$$

- 冷卻伴隨除濕的情況(完全冷凝)

在此應用下，總熱傳量與冷凝熱傳量相同，故

$$Ua = h_L a$$

- 氣體冷卻伴隨增濕的情況。

$$Ua = \gamma h_G a$$

- 氣體加熱並伴隨增濕的情況。



塔槽熱傳設計方程式

- 參考熱交換設計一書第16-5
- 資料非常欠缺且老舊
- 由於這些資料多使用因次化的單位，而且經常公制、英制交錯的使用，讀者在使用時一定要用對單位



空氣特性與空氣線圖



- M_a : 乾空氣的質量
- M_w : 水蒸氣的質量
- M : 濕空氣的總質量 = $M_a + M_w$
- P : 濕空氣的總壓力 = $P_a + P_w$
- P_a : 乾空氣的壓力
- P_w : 水蒸氣的壓力
- x_a : 乾空氣的莫耳分率 (mole fraction of air) = P_a/P
- x_w : 水蒸氣的莫耳分率 (mole fraction of water) = P_w/P
- 比濕 (humidity ratio, moisture content, mixing ratio) :
 - $W = M_w/M_a = 0.62198x_w/x_a$
- 絕對溼度 (absolute humidity, water vapor density) : $d_v = M_w/V$
- 濕空氣密度 : $\rho = (M_a + M_w)/V = (1/v)(1 + W)$
- 其中 v 為濕空氣的比容 (m^3/kg dry air)
- 飽和比濕 (saturation humidity ratio, $W_s(T, P)$) 為在一給定的溫度及壓力下，濕空氣的飽和比濕



濕空氣的飽和度(degree of saturation, μ)，定義如下：

$$\mu = \frac{W}{W_s} \Big|_{T,P} \quad (6-5)$$

相對濕度(relative humidity, ϕ)，定義如下：

$$\phi = \frac{x_w}{x_{ws}} \Big|_{T,P} \quad (6-6)$$

由於

$$P = (P_a + P_w) \quad , \quad n = (n_a + n_w)$$

$$x_a = P_a/P$$

$$x_w = P_w/P$$

由比濕的定義可得到：

$$W = 0.62198 \frac{x_w}{x_a} = 0.62198 \frac{P_w}{P - P_w} \quad (6-7)$$



$$\phi = \frac{x_w}{x_{ws}} \Big|_{T,P} = \frac{\frac{P_w}{P}}{\frac{P_{ws}}{P}} = \frac{P / \left(\frac{0.62198}{W} + 1 \right)}{P / \left(\frac{0.62198}{W_s} + 1 \right)} = \frac{1 + \frac{0.62198}{W_s}}{1 + \frac{0.62198}{W}} = \mu \frac{0.62198 + W_s}{0.62198 + W}$$

$$\Rightarrow \mu = \frac{\phi}{1 + (1 - \phi)W_s / 0.62198} \quad (6-9)$$

露點溫度(dew point temperature, T_{dew})的定義為空氣在同一壓力及比濕條件下時，相對飽和濕空氣的乾球溫度：

$$W(P, T_{dew}) = W_s \quad (6-10)$$

ASHRAE (2005)提供下列經驗方程式來快速計算露點溫度 T_{dew} 如下：

$$P_{ws}(T_{dew}) = P_w = (P \times W) / (0.62198 + W) \quad (6-11)$$

若露點溫度在0到93°C間：

$$T_{dew} = 6.54 + 14.526\alpha + 0.7398\alpha^2 + 0.09486\alpha^3 + 0.4569P_w^{0.1984} \quad (6-12)$$

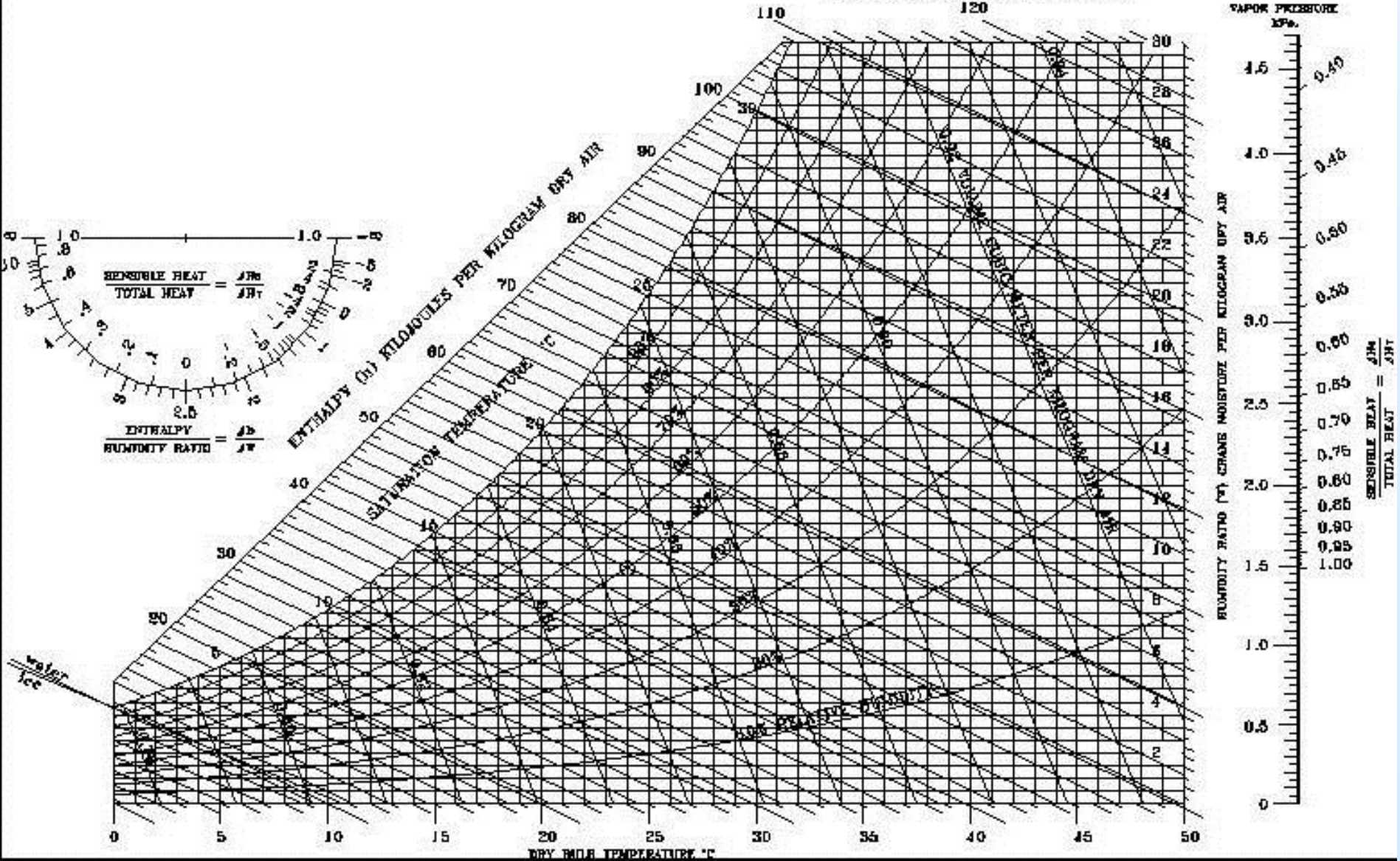
若溫度低於0°C

$$T_{dew} = 6.09 + 12.608\alpha + 0.4959\alpha^2 \quad (6-13)$$

其中 $\alpha = \ln(P_w)$ ， P_w 的壓力單位為kPa， T_{dew} 溫度單位為°C，注意這些都是有因次的方程式。



NORMAL TEMPERATURE SEA LEVEL
BAROMETRIC PRESSURE 101.326 kPa.





主要空氣組成

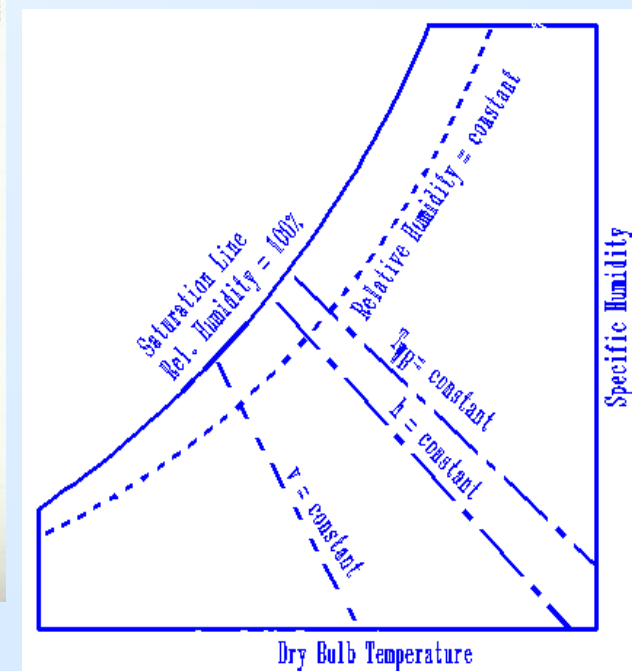
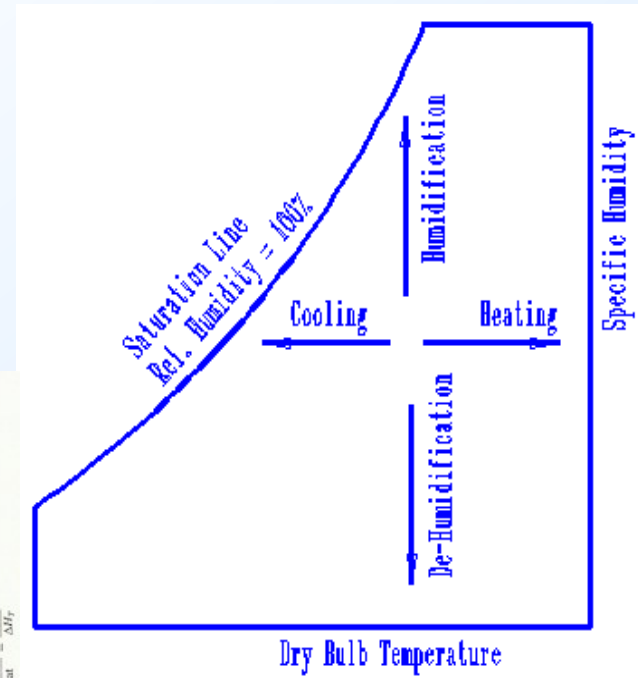
- 氮(Nitrogen) : 78.084%
- 氧(Oxygen) : 20.9476%
- 氬(Argon) : 0.934%
- 二氧化碳(Carbon dioxide) : 0.0314%
- 氖(Neon) : 0.01818%
- 氦(Helium) : 0.000524%



空氣線圖

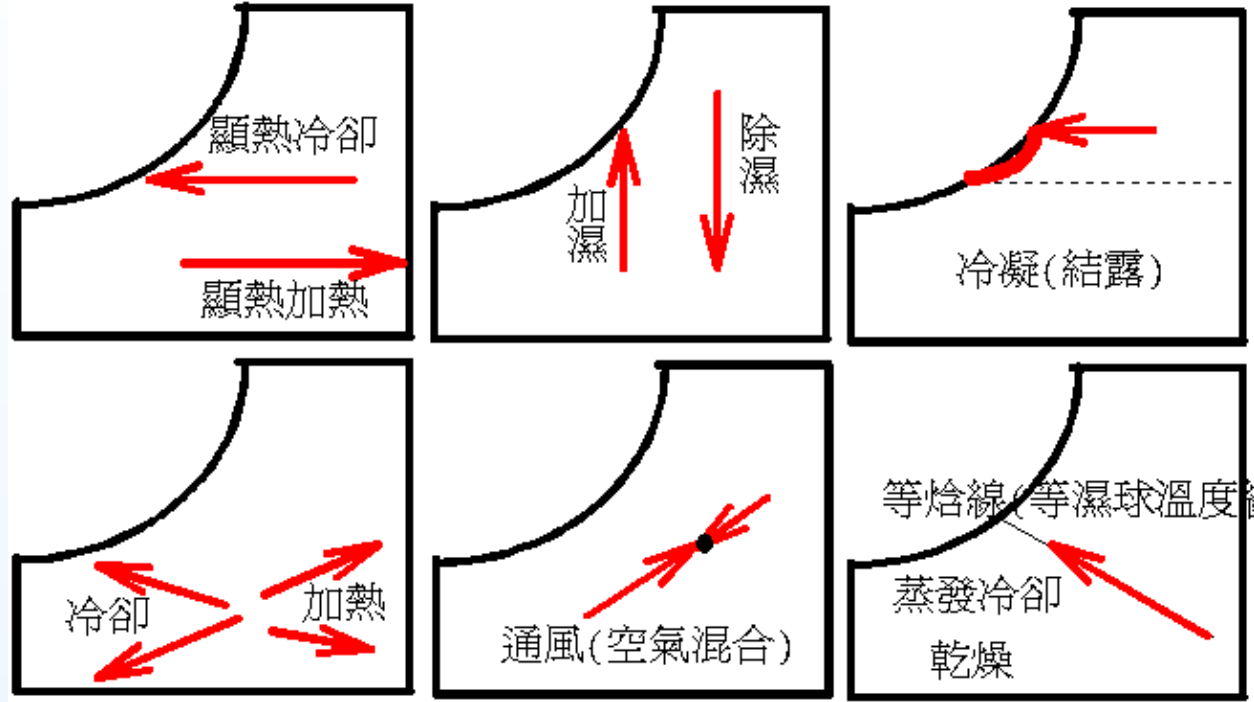
Psychrometric Chart for Air at 101.325 kPa

Source: ASHRAE Publications





Process for HVAC&R



顯熱(間接)加熱與冷卻
(Sensible Heating and
Sensible Cooling)

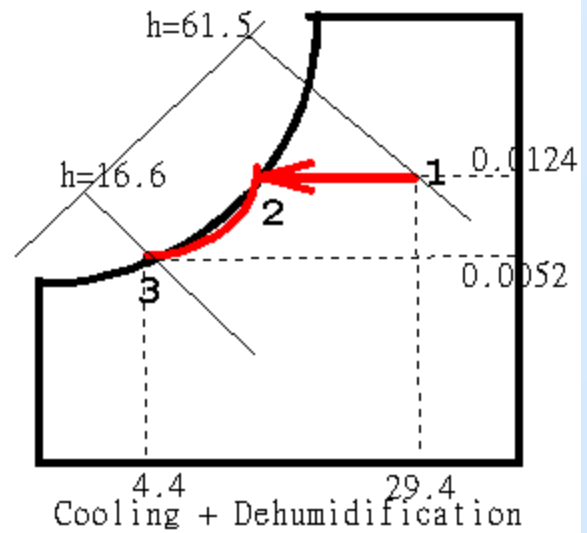
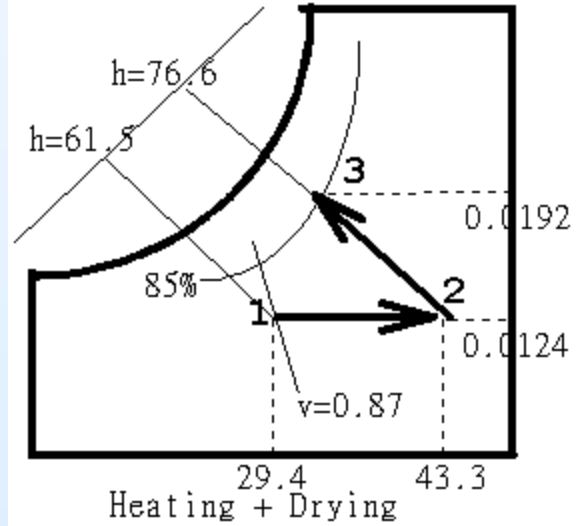
加濕與除濕 (Humidification ,
Dehumidification)

加熱與冷卻 (Heating ,
Cooling)

通風/空氣混合 (Ventilation ,
Air Mixing)

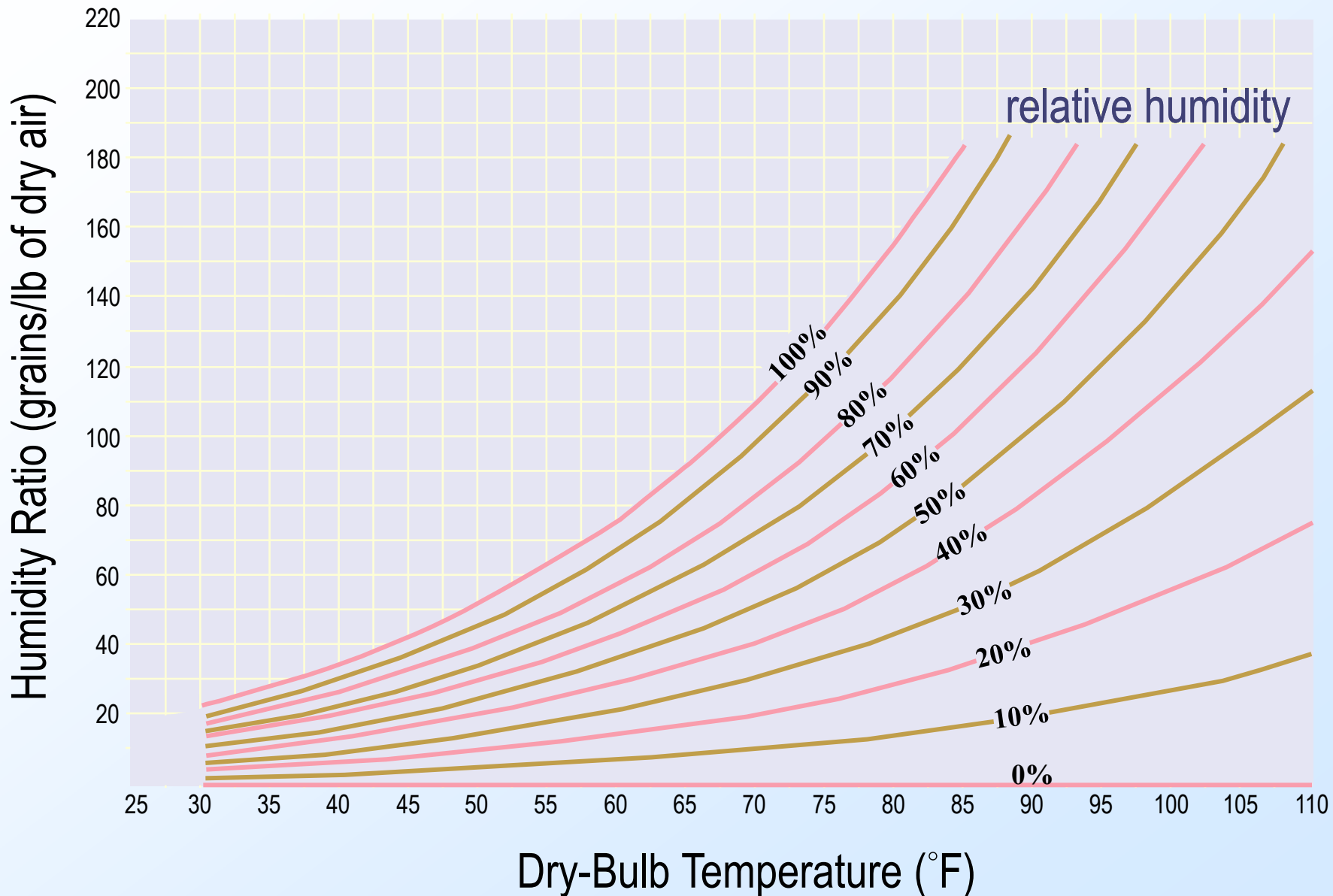
蒸發冷卻與乾燥 (Evaporative
Cooling , Drying)

上述之綜合 (Combinations of
the above)



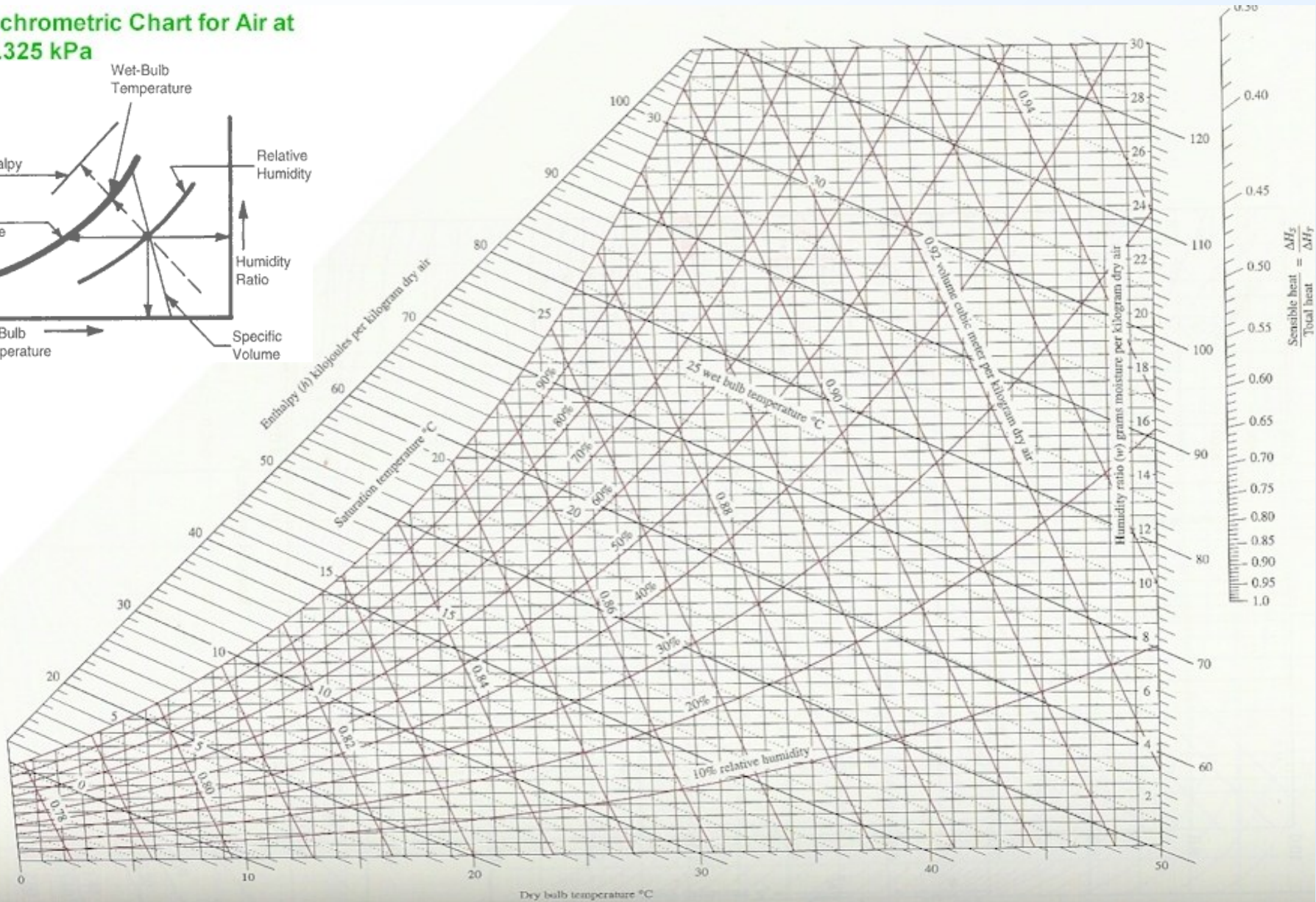
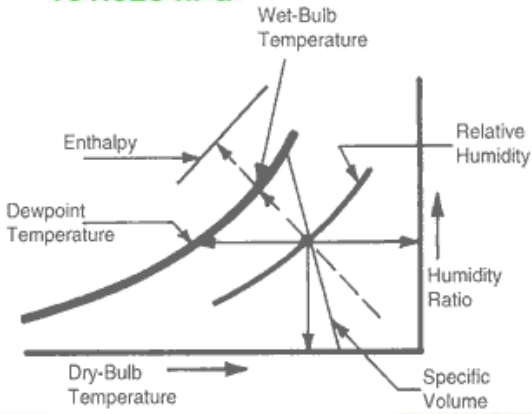


Relative Humidity Curves



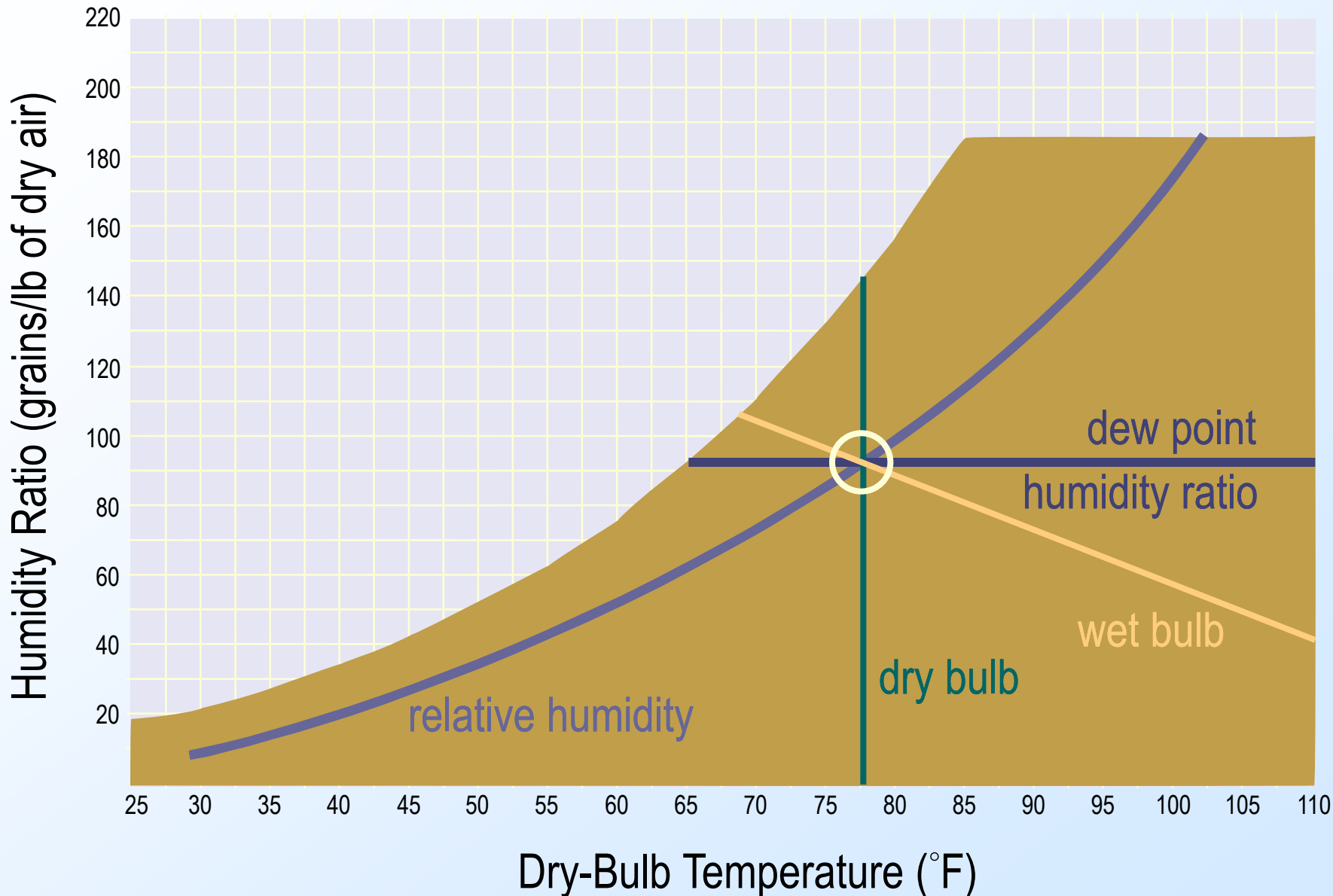


Psychrometric Chart for Air at 101.325 kPa





Properties





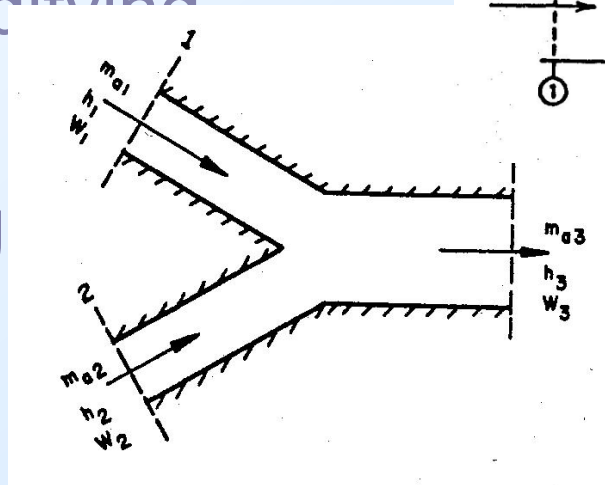
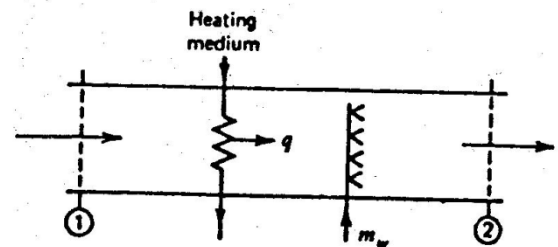
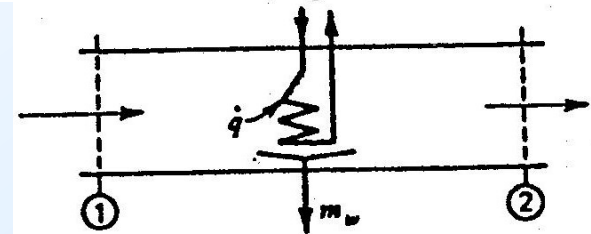
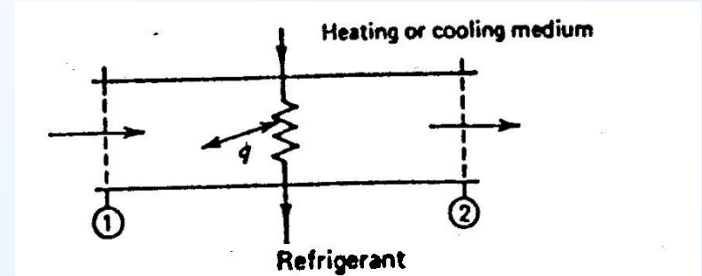
Typical Conditioning process..

● Heating & Cooling

● Cooling & Dehumidification

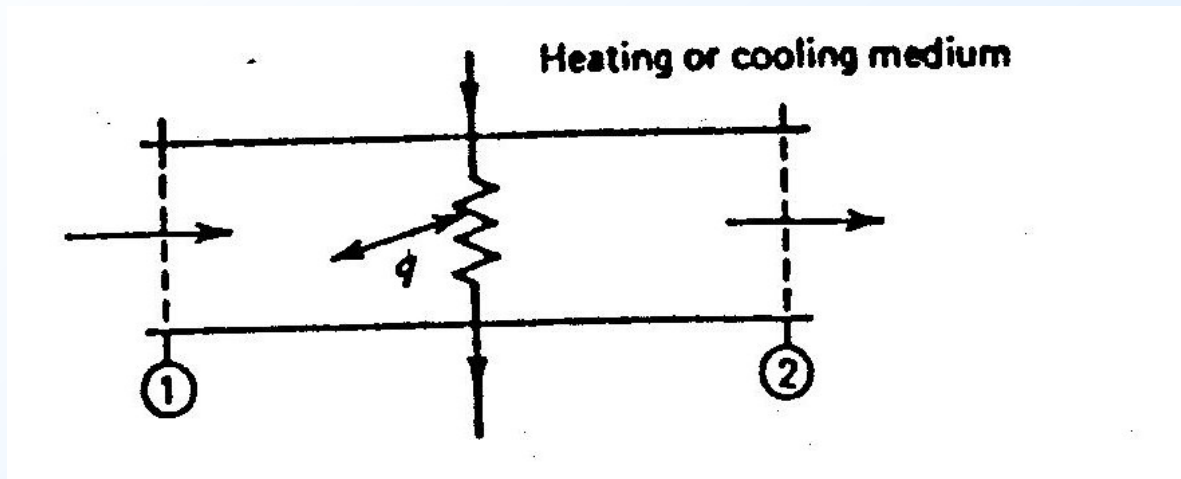
● Heating & Humidifying

● Adiabatic Mixing





Heating or Cooling

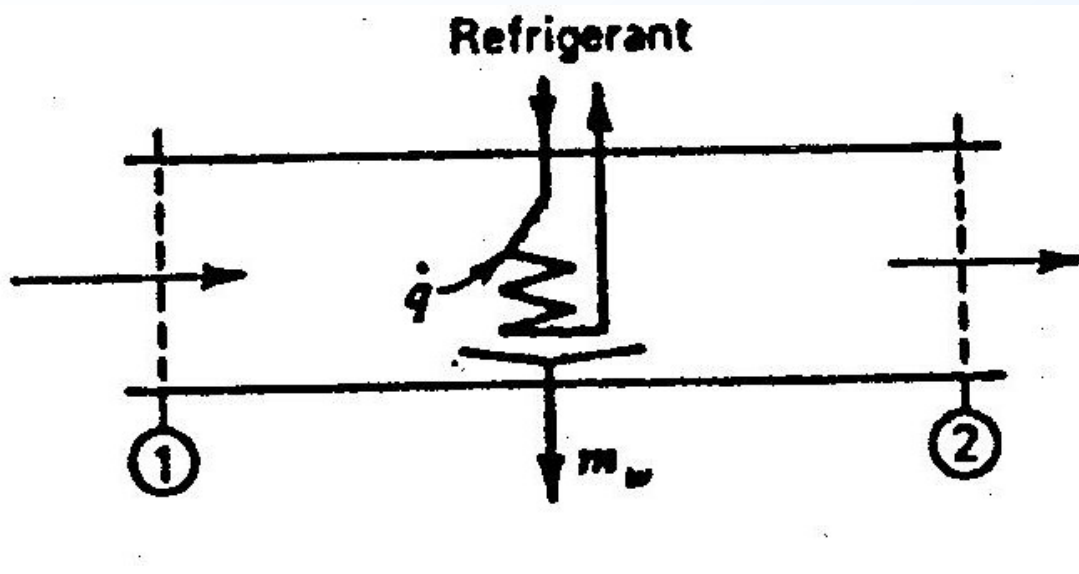


$$m_a h_1 - m_a h_2 + q = 0$$

$$W_2 = W_1$$



Cooling & Dehumidification



$$m_a h_1 = m_a h_2 + {}_1q_2 + m_w h_{w2}$$

$$m_a W_1 = m_a W_2 + m_w$$

Thus:

$$m_w = m_a (W_1 - W_2)$$

$${}_1q_2 = m_a [(h_1 - h_2) - (W_1 - W_2) h_{w2}]$$

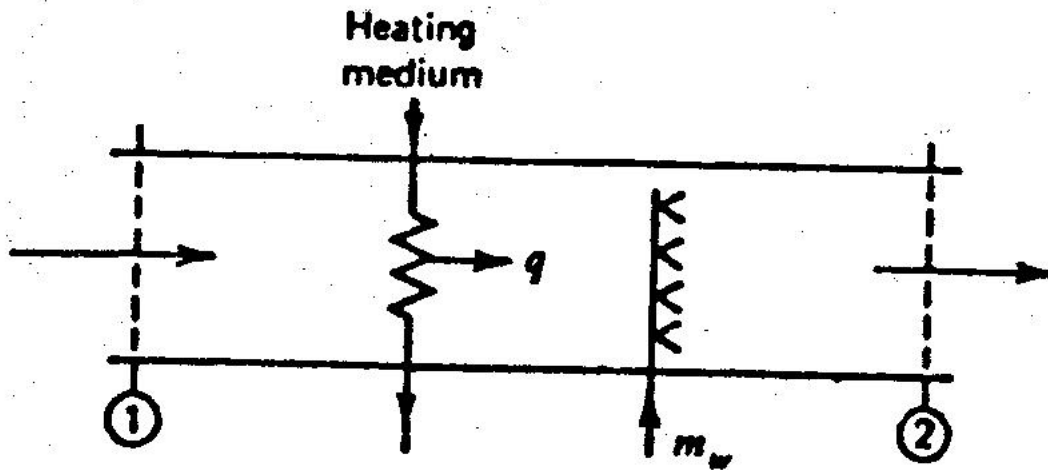


Question

- Air flow (5 CMM) 27 DB 50%RH enters air-conditioning, and departs at 15 DB, 100 RH , what is the cooling capacity of the air-conditioner? Sensible cooling load? Latent cooling load? Rate of condensate rate?



Heating & Humidifying Air



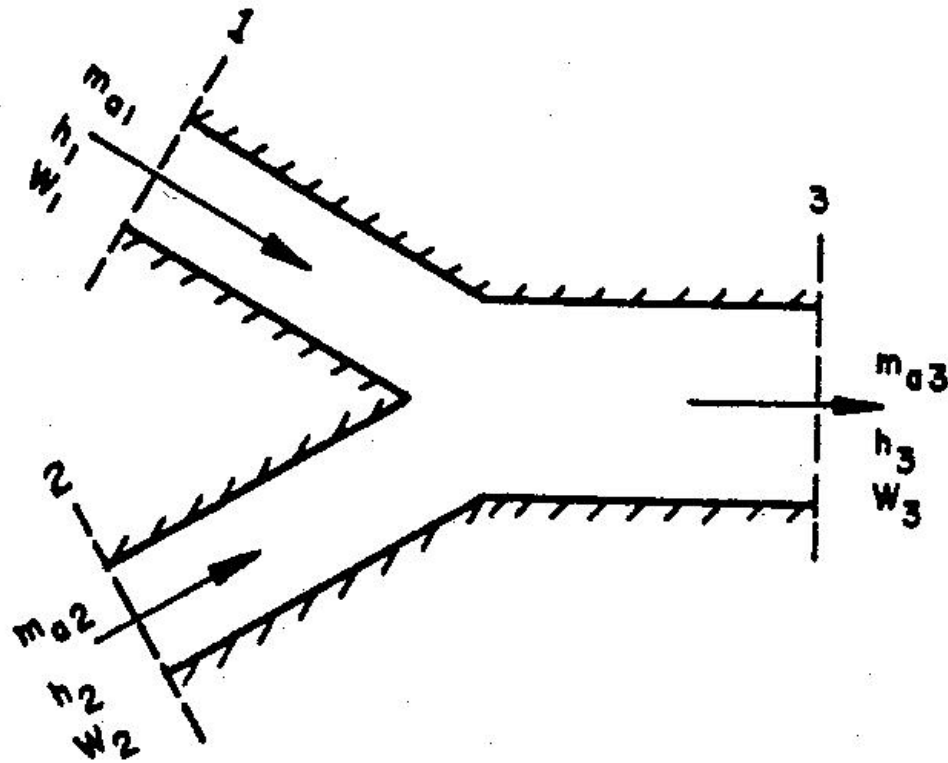
$$m_a h_1 + q + m_w h_w = m_a h_2$$

and a mass balance on the water gives:

$$m_a W_1 + m_w = m_a W_2$$



Adiabatic Mixing



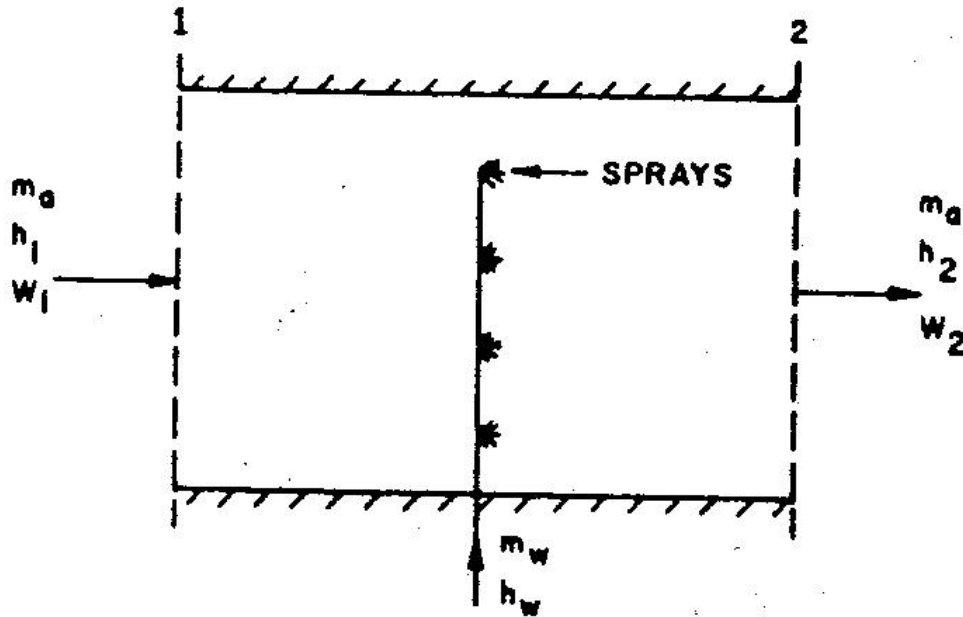
$$m_{a1}h_1 + m_{a2}h_2 = m_{a3}h_3$$

$$m_{a1} + m_{a2} = m_{a3}$$

$$m_{a1}W_1 + m_{a2}W_2 = m_{a3}W_3$$



Injection of Water Into Air (humidification)



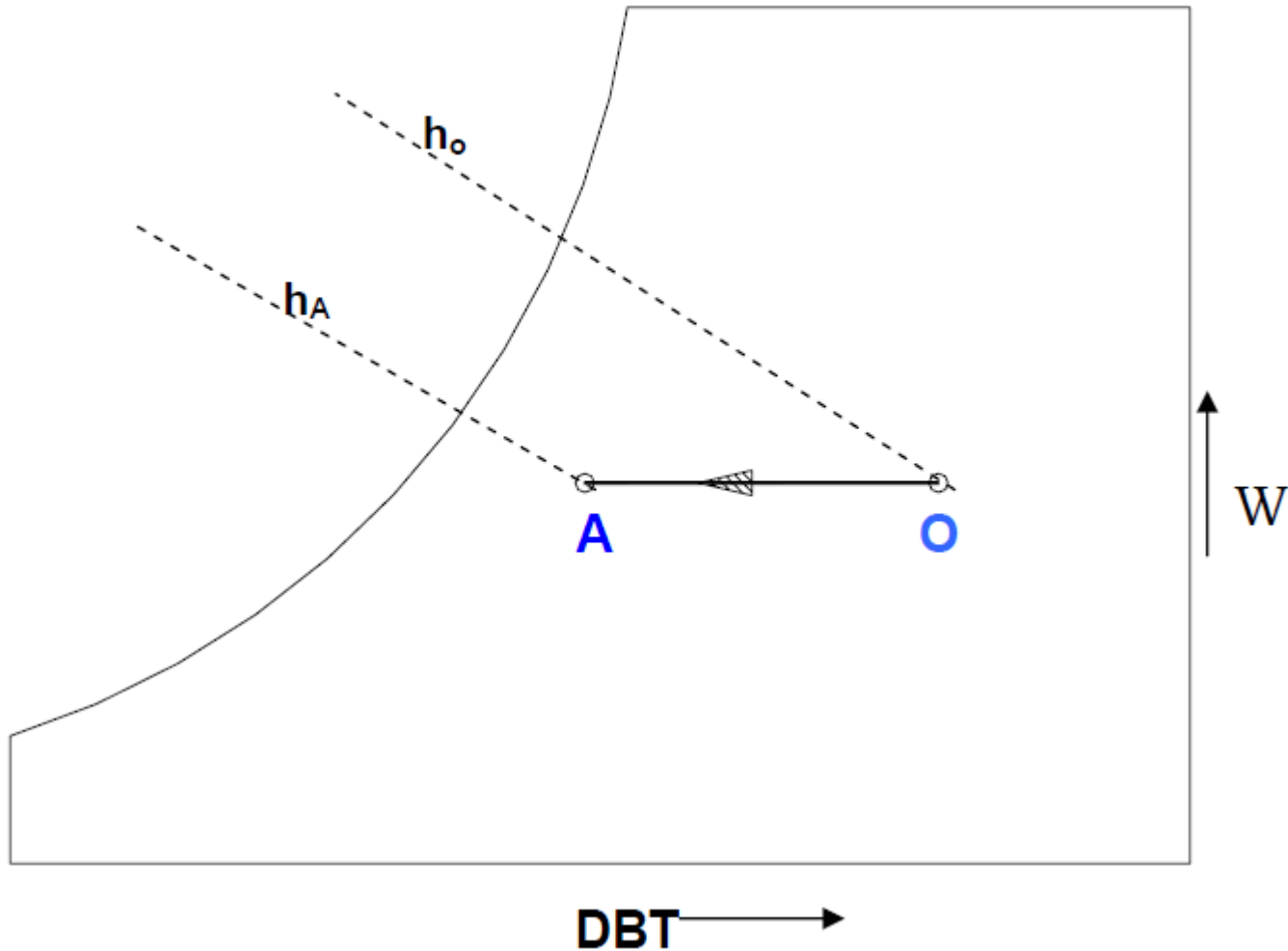
$$m_a h_1 + m_w h_w = m_a h_2$$

$$m_a W_1 + m_w W_w = m_a W_2$$



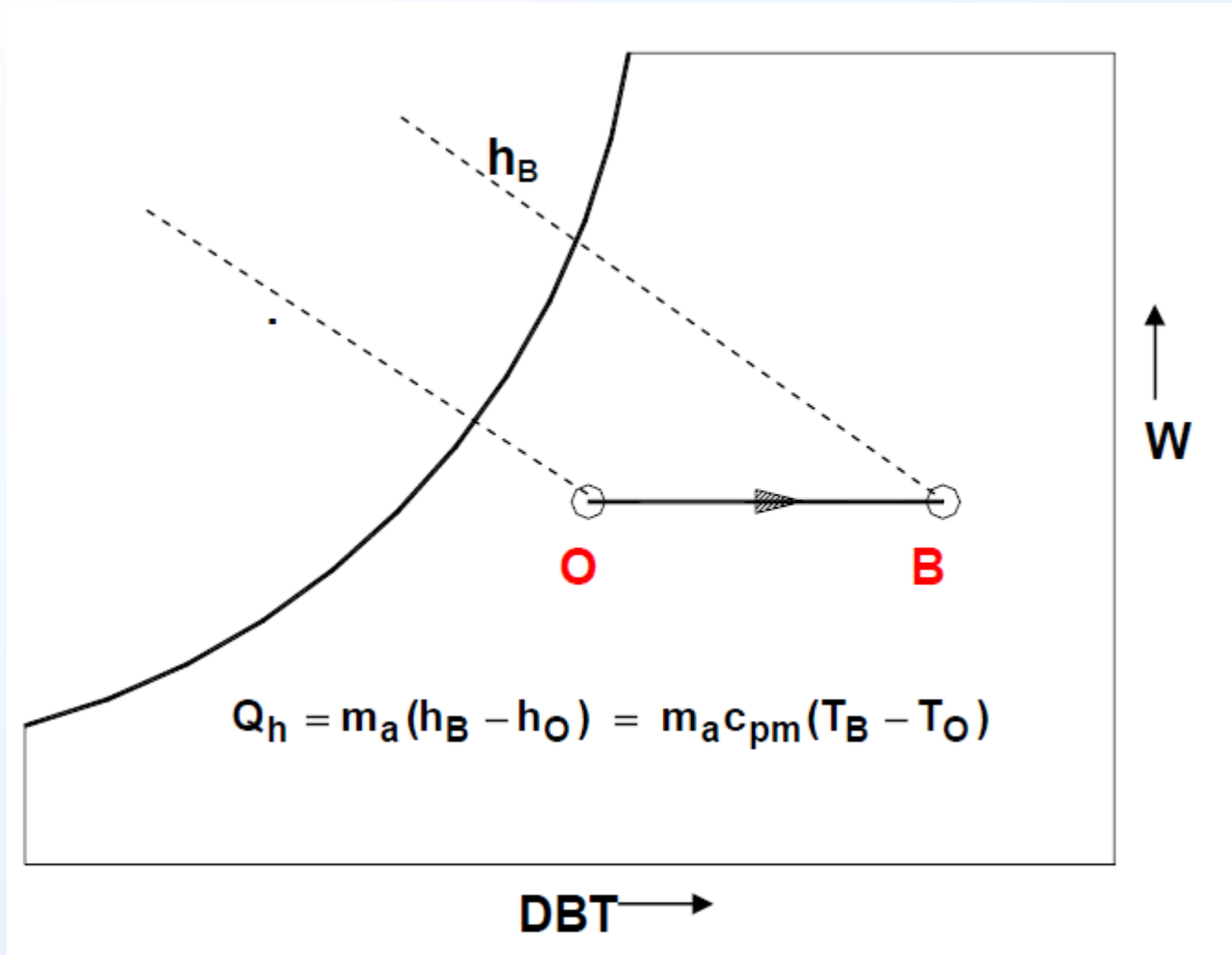
Sensible cooling

$$Q_c = m_a(h_o - h_A) = m_a c_{pm}(T_o - T_A)$$





Sensible heating





Cooling and dehumidification

By applying mass balance for the water:

$$m_a \cdot w_o = m_a \cdot w_c + m_w$$

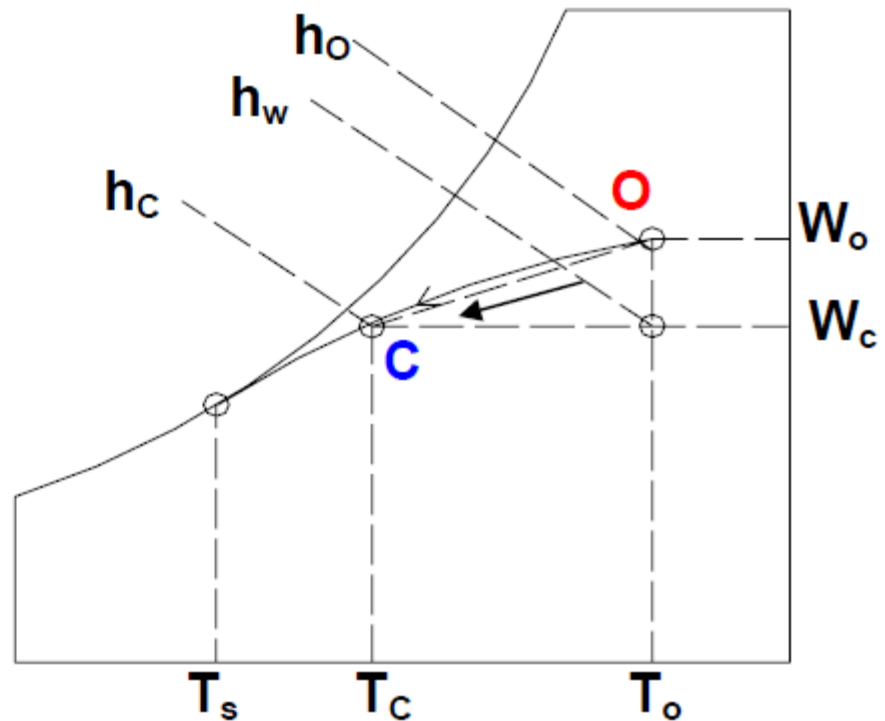
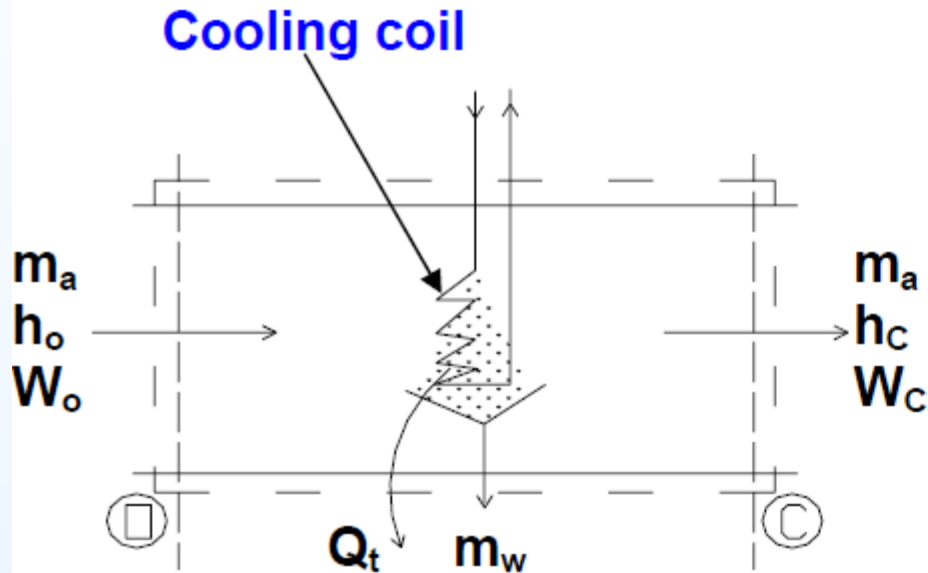
By applying energy balance:

$$m_a \cdot h_o = Q_t + m_w \cdot h_w + m_a \cdot h_c$$

$$Q_t = m_a (h_o - h_c) - m_a (w_o - w_c) h_w$$

- the 2nd term on the RHS of the above equation is normally small compared to the other terms, so it can be neglected. Hence,

$$Q_t = m_a (h_o - h_c)$$





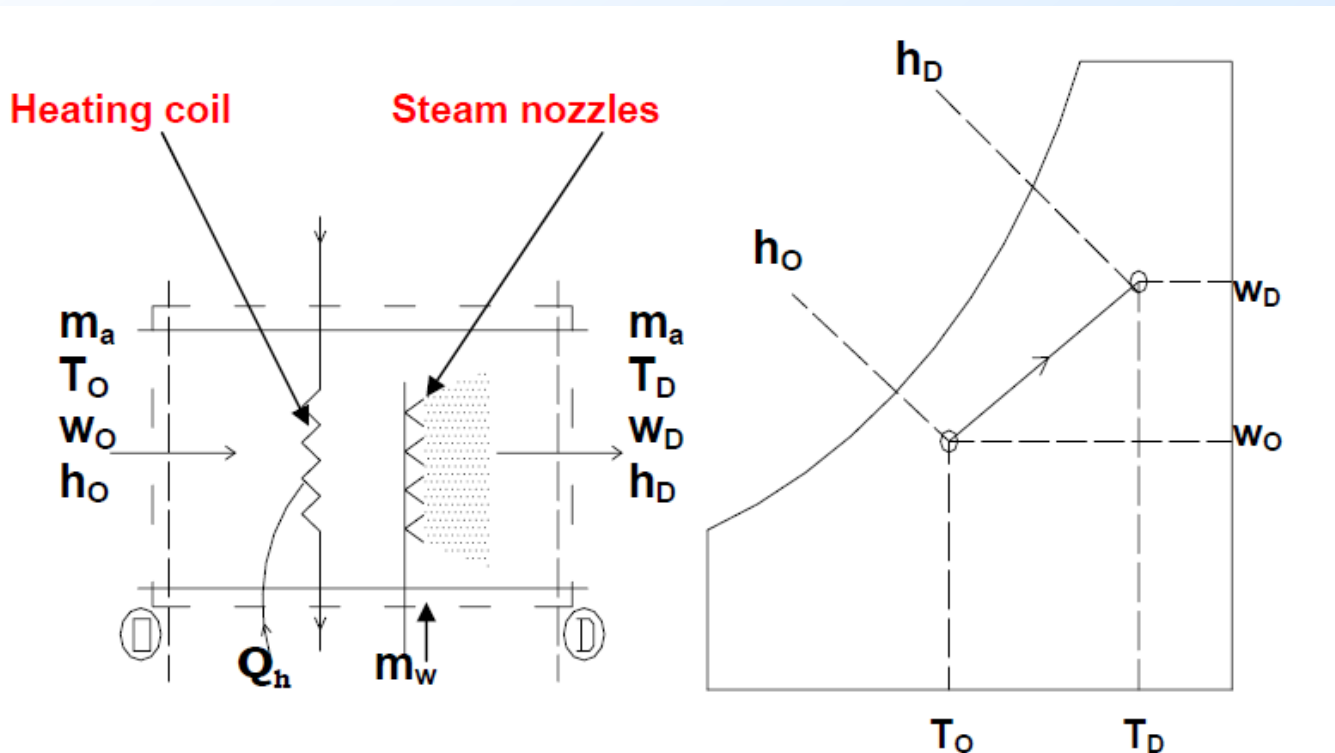
Heating and Humidification

- During winter it is essential to heat and humidify the room air for comfort.

$$m_w = m_a(w_D - w_O)$$

$$Q_h = m_a(h_D - h_O) - m_w h_w$$

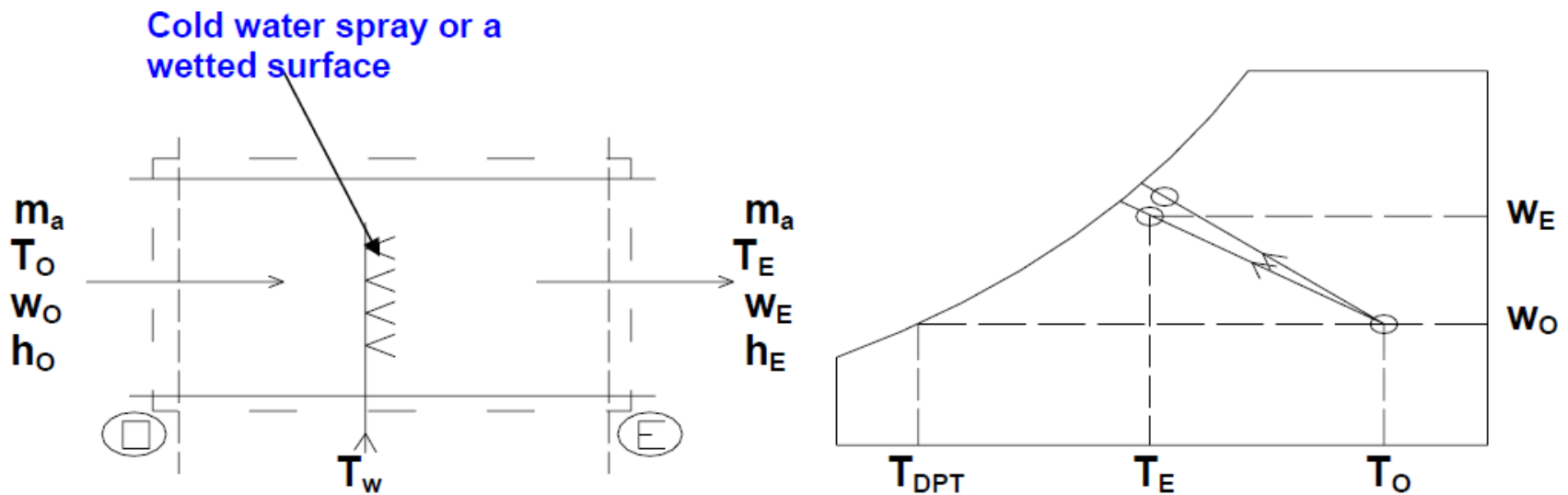
- where Q_h is the heat supplied through the heating coil and h_w is the enthalpy of steam





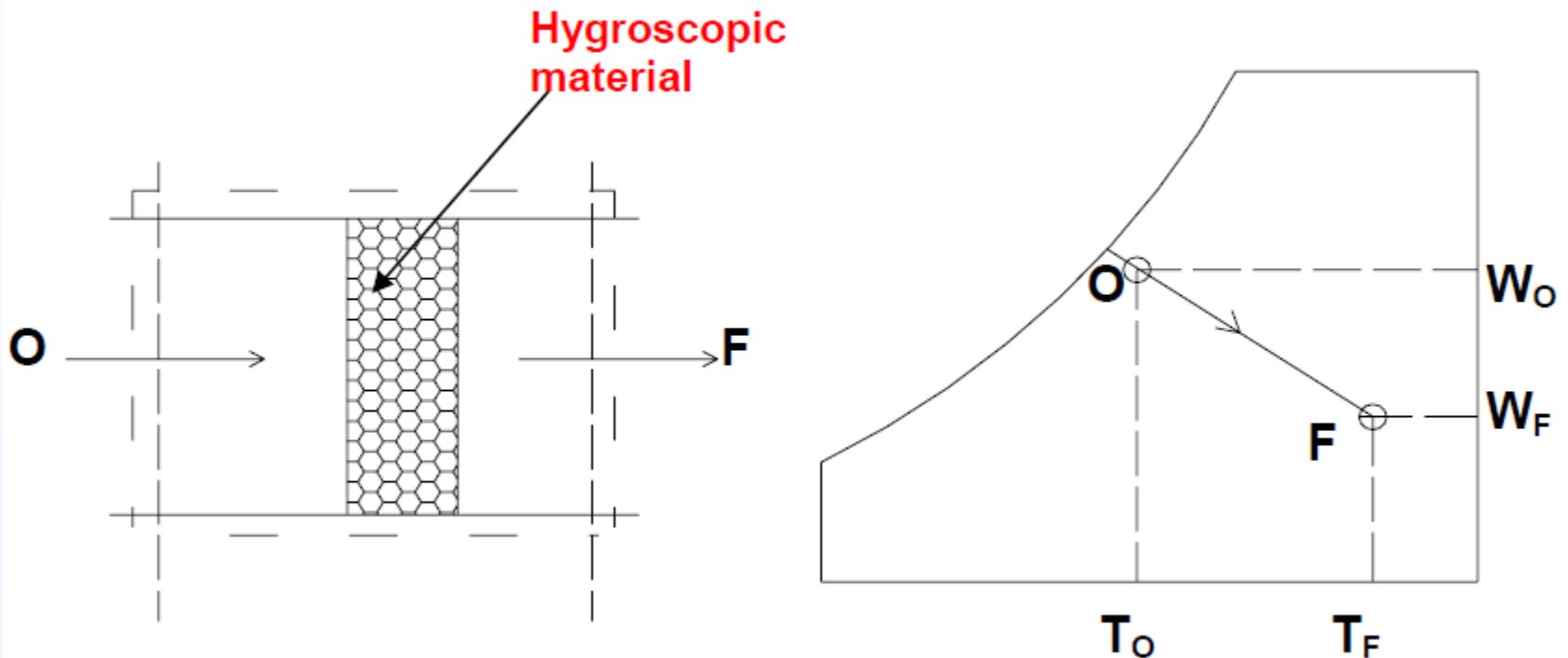
Cooling & humidification

- If the water temperature is greater than WBT, then there will be a net heat transfer from water to air. If the water temperature is less than WBT, then the net heat transfer will be from air to water.





Heating and de-humidification





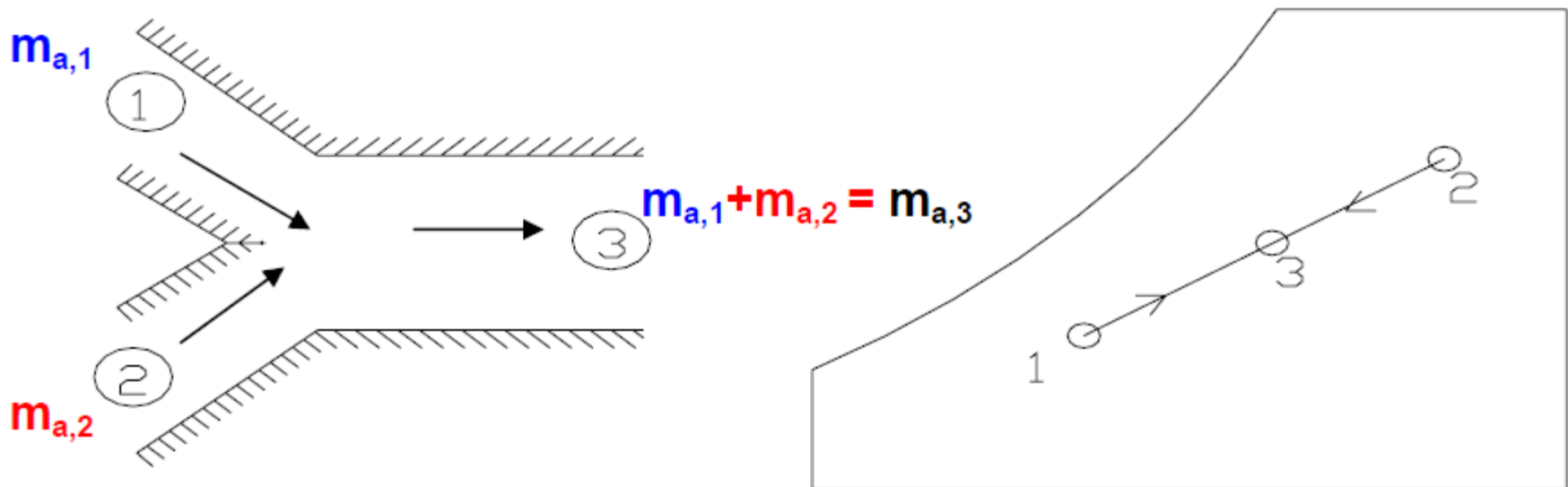
Mixing of air streams:

From the mass balance of dry air and water vapor:

$$m_{a,1}w_1 + m_{a,2}w_2 = m_{a,3}w_3 = (m_{a,1} + m_{a,2})w_3$$

From energy balance:

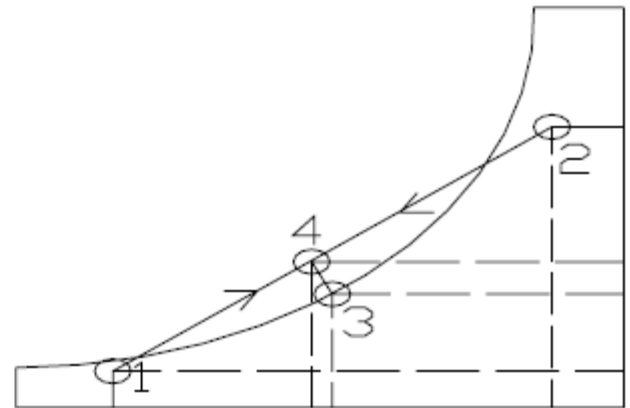
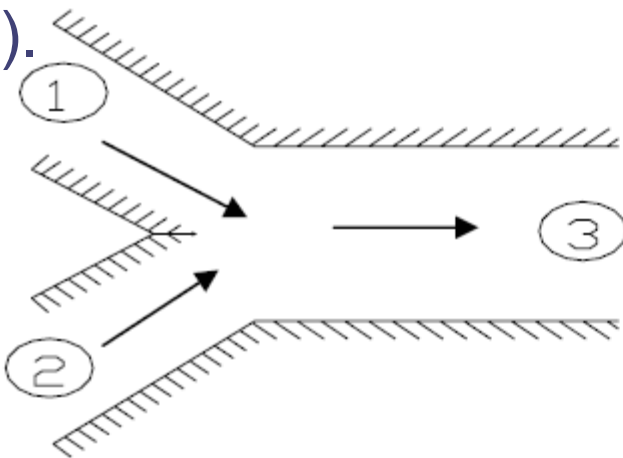
$$m_{a,1}h_1 + m_{a,2}h_2 = m_{a,3}h_3 = (m_{a,1} + m_{a,2})h_3$$





Mixing with condensation

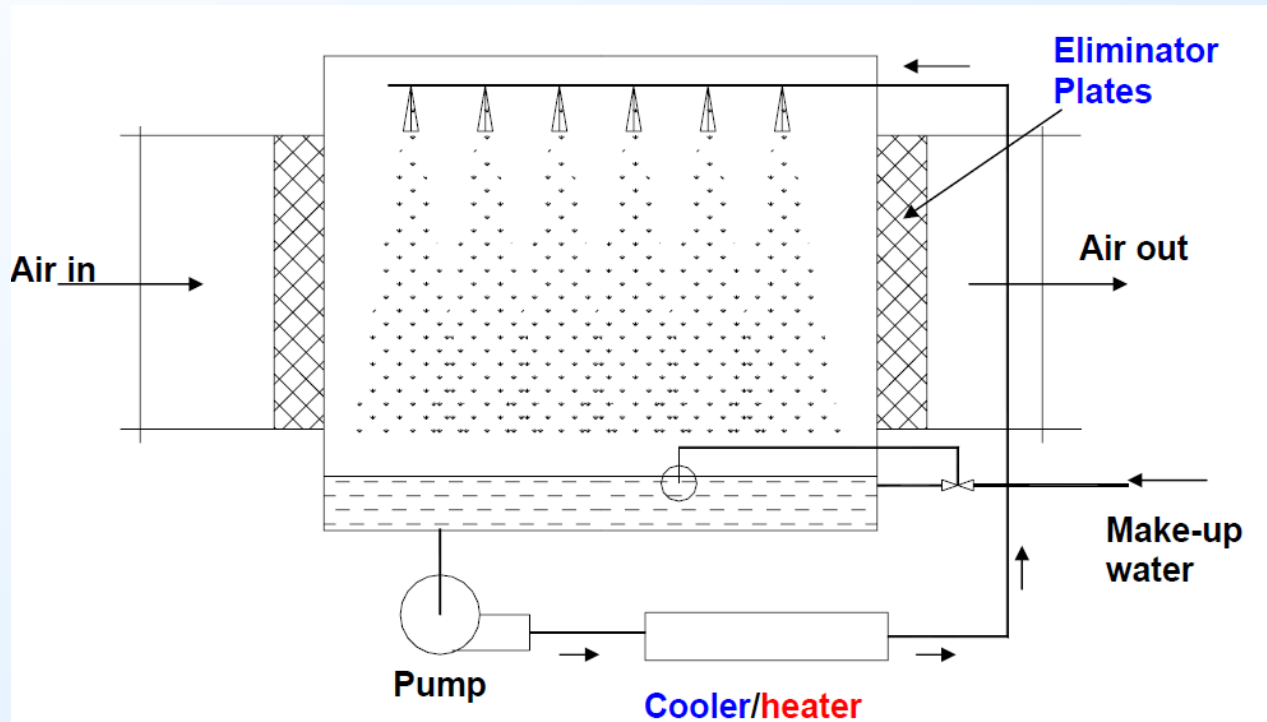
Dry air mixes with warm air at high relative humidity, the resulting mixture condition may lie in the two-phase region, as a result there will be condensation of water vapor and some amount of water will leave the system as liquid water. Due to this, the humidity ratio of the resulting mixture (point 3) will be less than that at point 4. Corresponding to this will be an increase in temperature of air due to the release of latent heat of condensation. This process rarely occurs in an air conditioning system, but this is the phenomenon which results in the formation of fog or frost (if the mixture temperature is below 0°C).





Air Washers

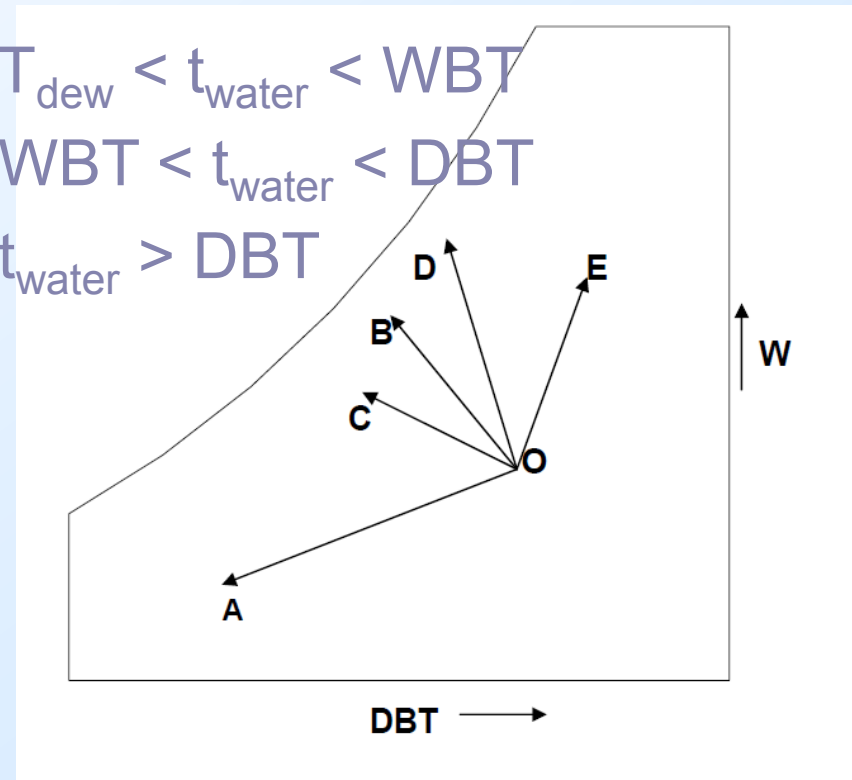
- An air washer is a device for conditioning air. As shown in the figure, in an air washer air comes in direct contact with a spray of water and there will be an exchange of heat and mass (water vapor) between air and water. The outlet condition of air depends upon the temperature of water sprayed in the air washer .





Various psychrometric processes that can take place in an air washer

- A) Cooling and dehumidification: $t_{\text{water}} < T_{\text{dew}}$
- B) Adiabatic saturation: $t_{\text{water}} = \text{WBT}$
- C) Cooling and humidification: $T_{\text{dew}} < t_{\text{water}} < \text{WBT}$
- D) Cooling and humidification: $\text{WBT} < t_{\text{water}} < \text{DBT}$
- E) Heating and humidification: $t_{\text{water}} > \text{DBT}$

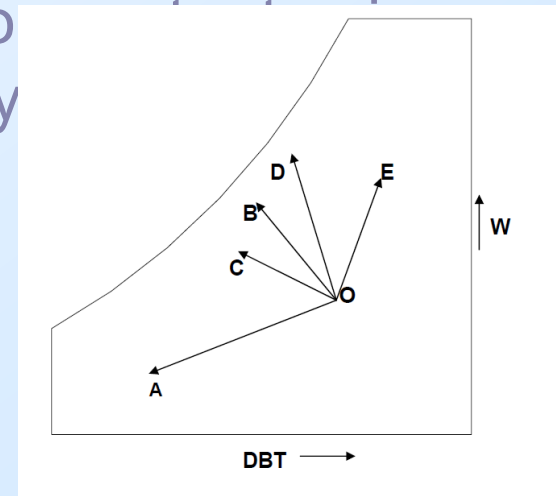




- **A) Cooling and dehumidification:** $t_{\text{water}} < T_{\text{dew}}$. Since the exit enthalpy of air is less than its inlet value, from energy balance it can be shown that there is a transfer of total energy from air to water. Hence to continue the process, water has to be externally cooled. Here both latent and sensible heat transfers are from air to water.
- **B) Adiabatic saturation:** $t_{\text{water}} = \text{WBT}$. Here the sensible heat transfer from air to water is exactly equal to latent heat transfer from water to air. Hence, no external cooling or heating of water is required. That is this is a case of pure water recirculation. This the process that takes place in a perfectly insulated evaporative cooler.
- **C) Cooling and humidification:** $T_{\text{dew}} < t_{\text{water}} < \text{WBT}$. Here the sensible heat transfer is from air to water and latent heat transfer is from water to air, but the total heat transfer is from air to water, hence, water has to be cooled externally.



- **D) Cooling and humidification: $WBT < t_{\text{water}} < DBT$.**
Here the sensible heat transfer is from air to water and latent heat transfer is from water to air, but the total heat transfer is from water to air, hence, water has to be heated externally. This is the process that takes place in a cooling tower. The air stream extracts heat from the hot water coming from the condenser, and the cooled water is sent back to the condenser.
- **E) Heating and humidification: $t_{\text{water}} > DBT$.** Here both sensible and latent heat transfers are from water to air, hence, water has to be heated externally



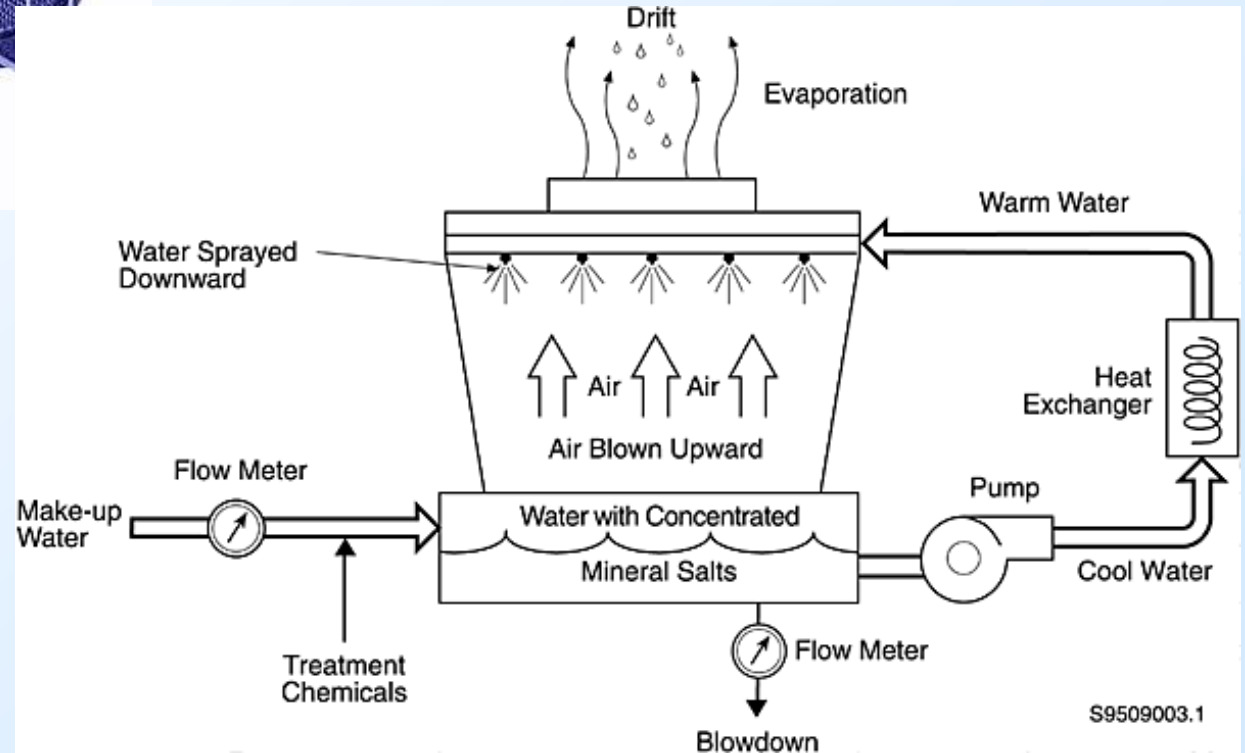
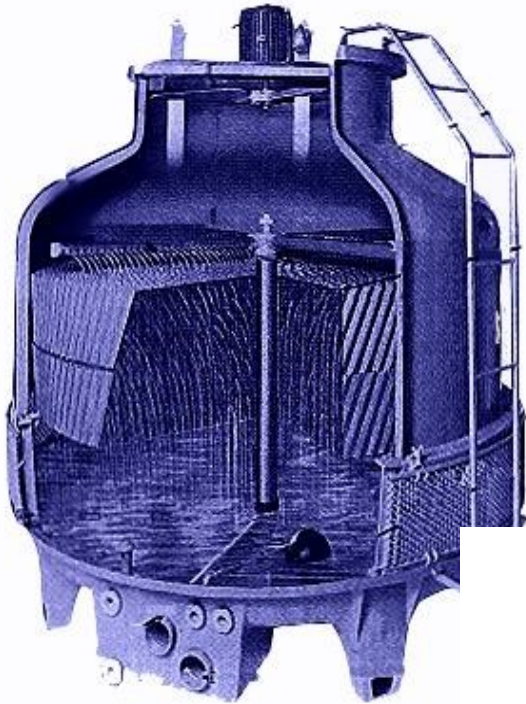


冷卻水塔 Cooling Tower

- 冷卻水塔為最典型的直接接觸式熱交換器，其散熱模式除了將較高溫的水已顯熱傳熱給空氣外，最主要是透過水與空氣的接觸，以氣化潛熱方式將冷卻水熱能轉移給空氣，達到降低水溫的目的。冷卻水塔型式種類相當多，工業上的應用也相當的普遍，例如工廠製程冷卻、大樓水冷式空調機、發電廠等等，因為水需要回收再利用，所以需用冷卻水塔將熱傳給空氣，簡而言之，冷卻水塔是用來節省水而達到散熱效果的最佳裝置。目的是要重複循環冷卻水，達到省水的功能。



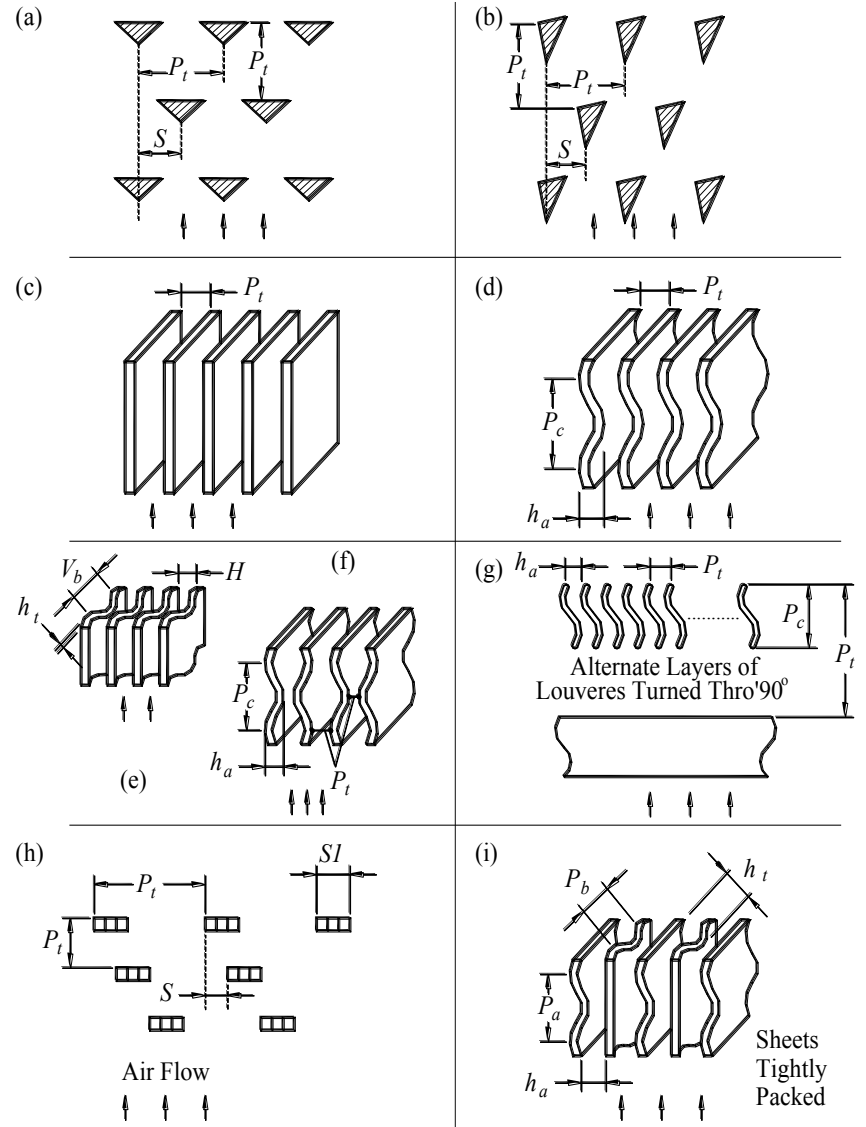
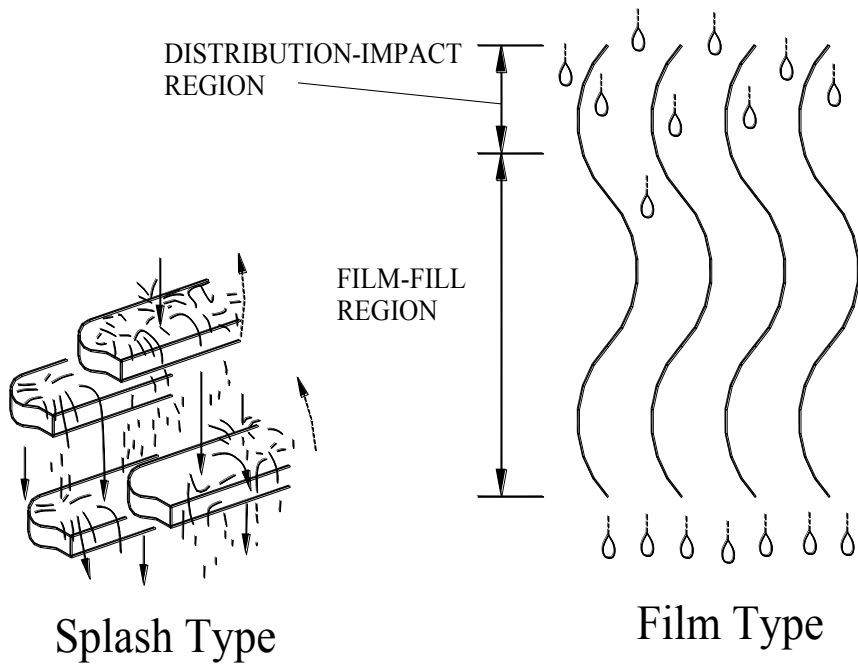
Cooling Tower





Typical configuration of Fillet

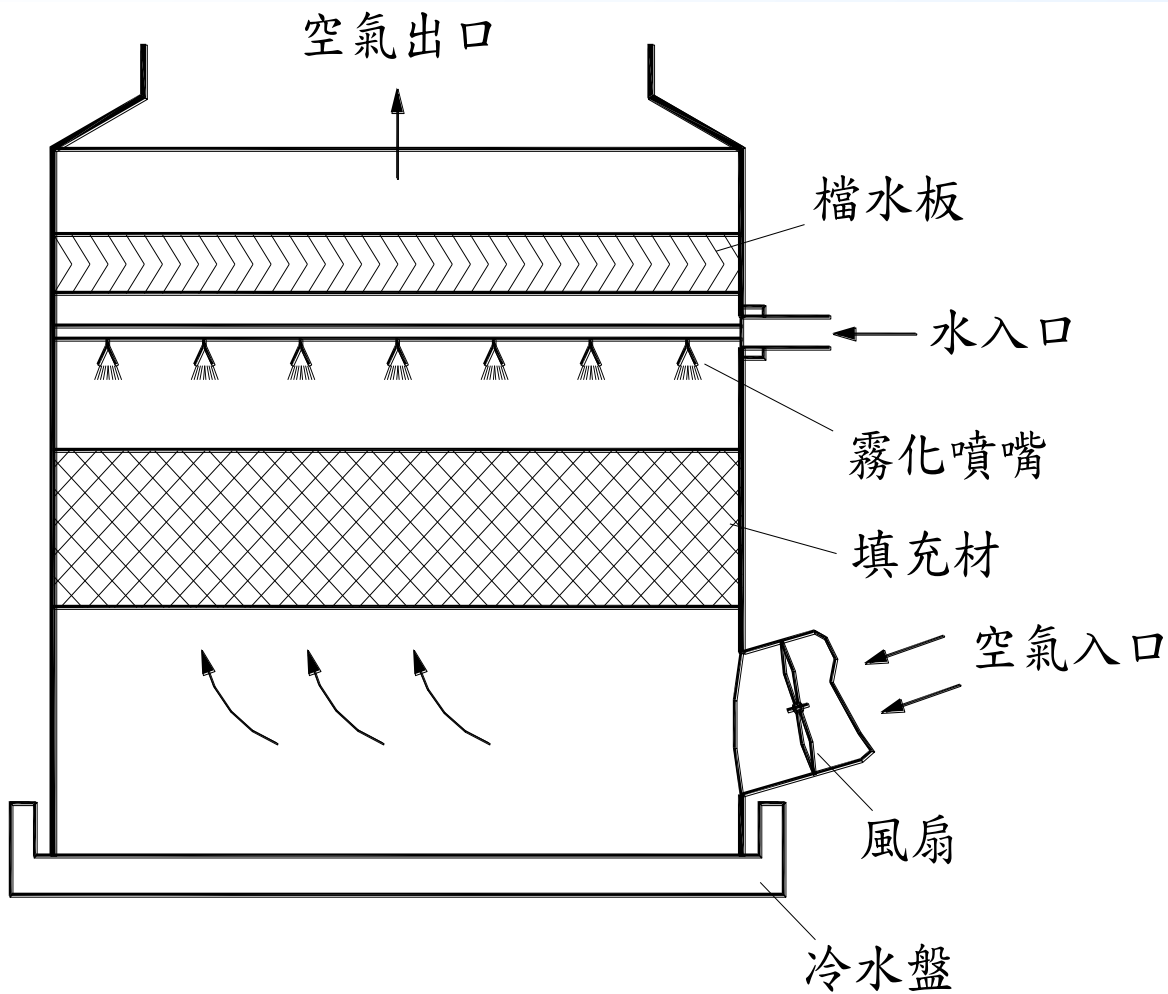
Type of Fillet

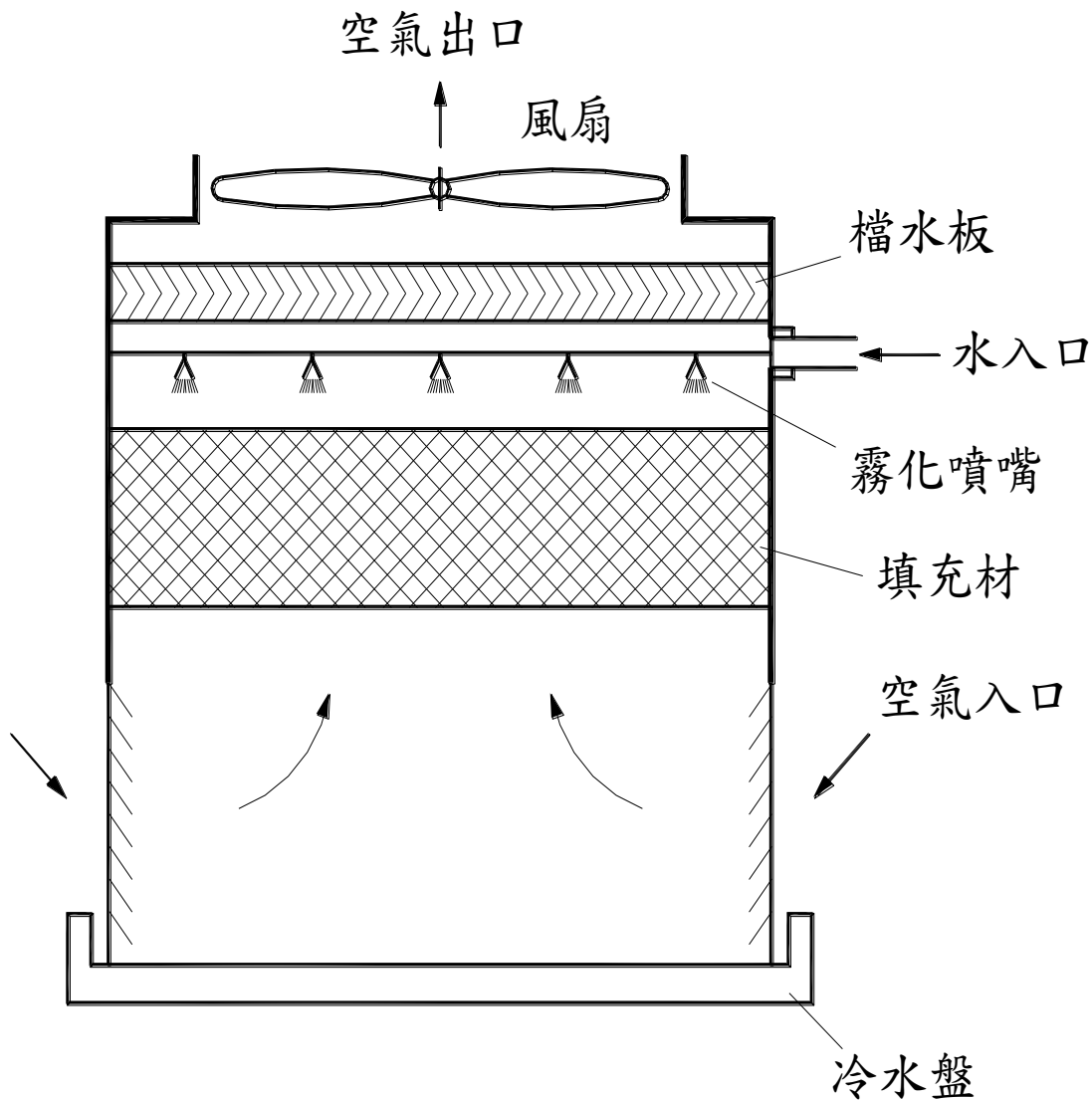


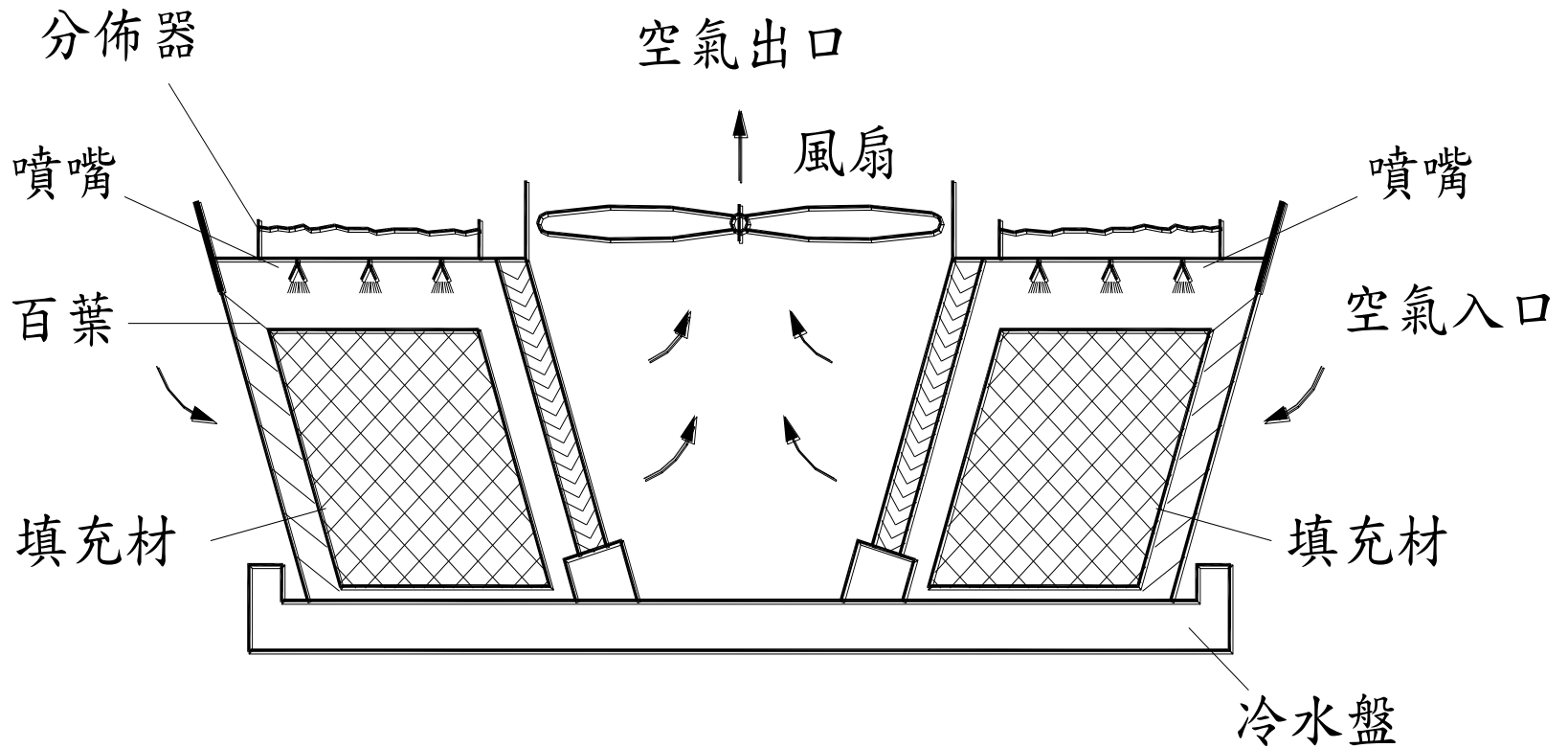


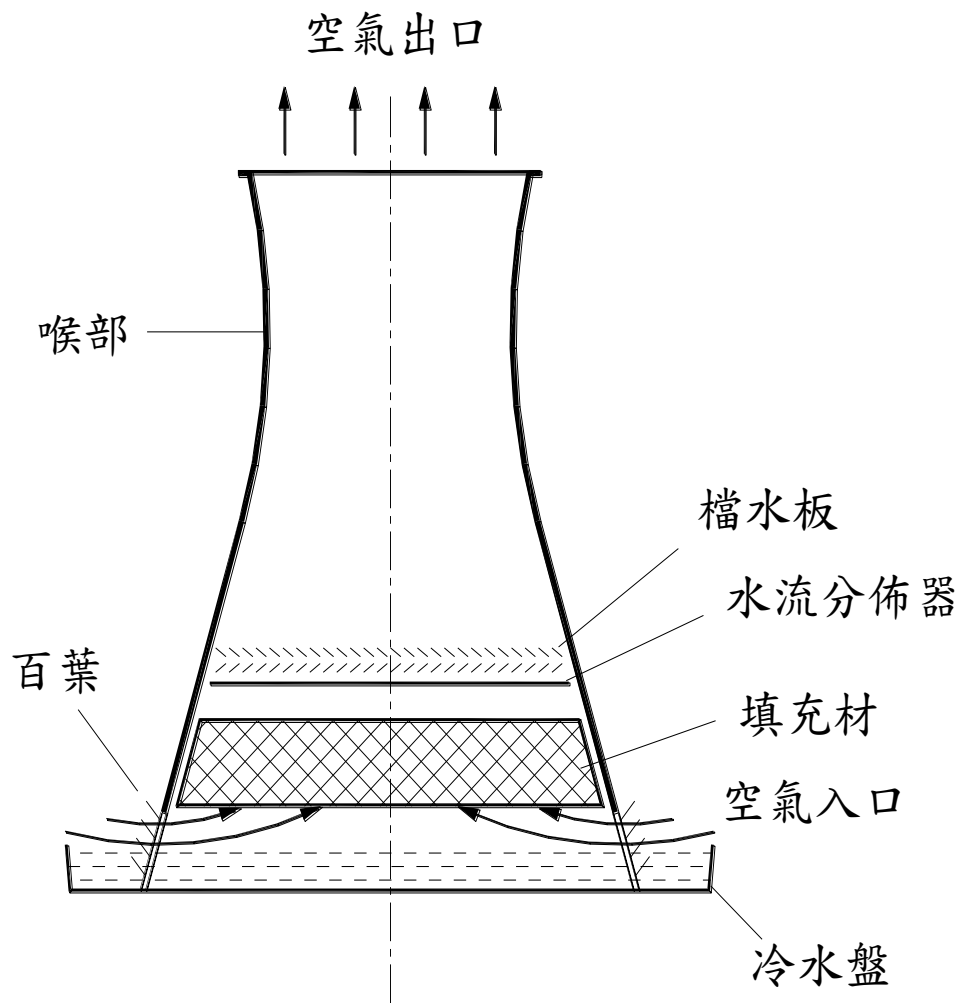
冷卻水塔的形式

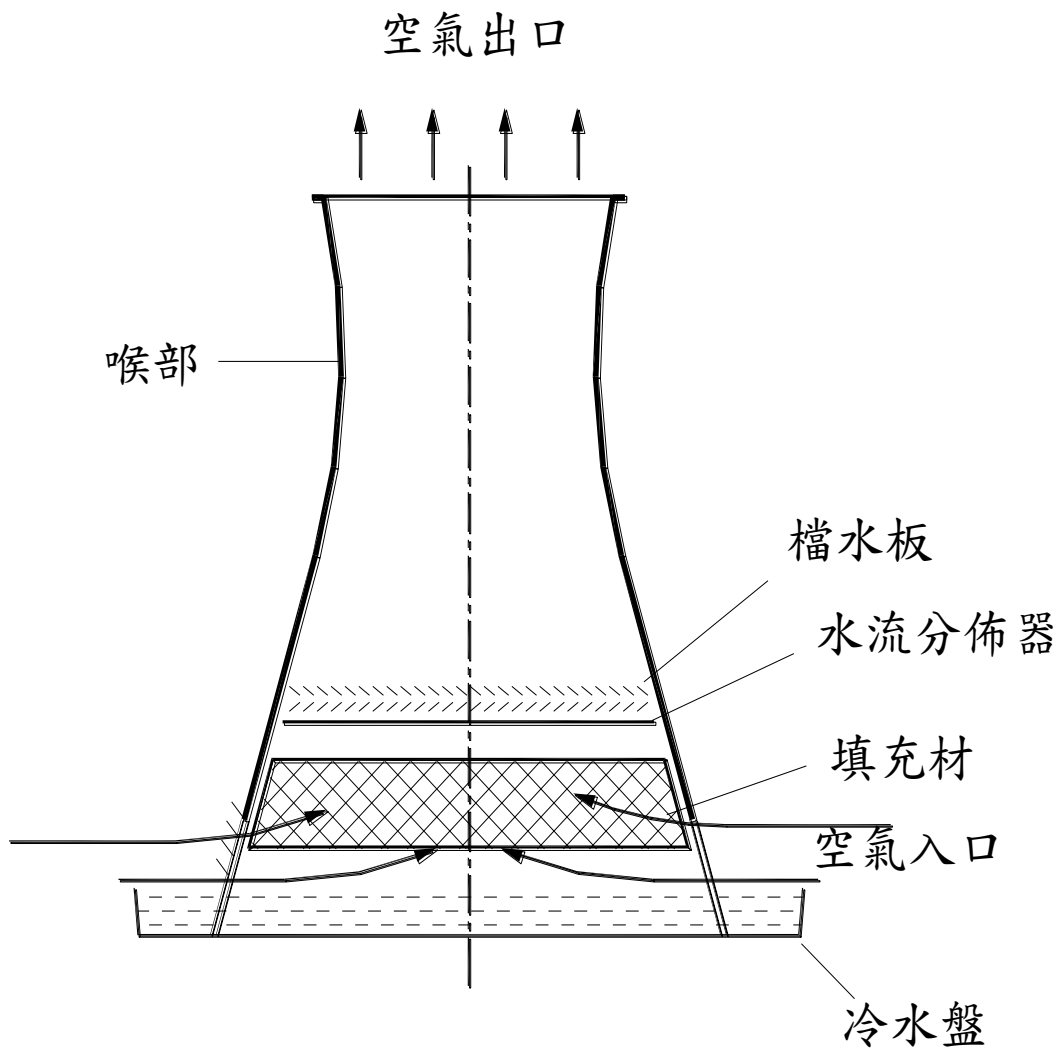
- ▶ Forced draft counter flow
- ▶ Induced draft counter flow
- ▶ Induced draft cross flow
- ▶ Natural draft counter flow
- ▶ Natural draft mixed flow
- ▶ Natural draft cross-flow
- ▶ Assisted draft counter flow
- ▶ Assisted draft cross-flow

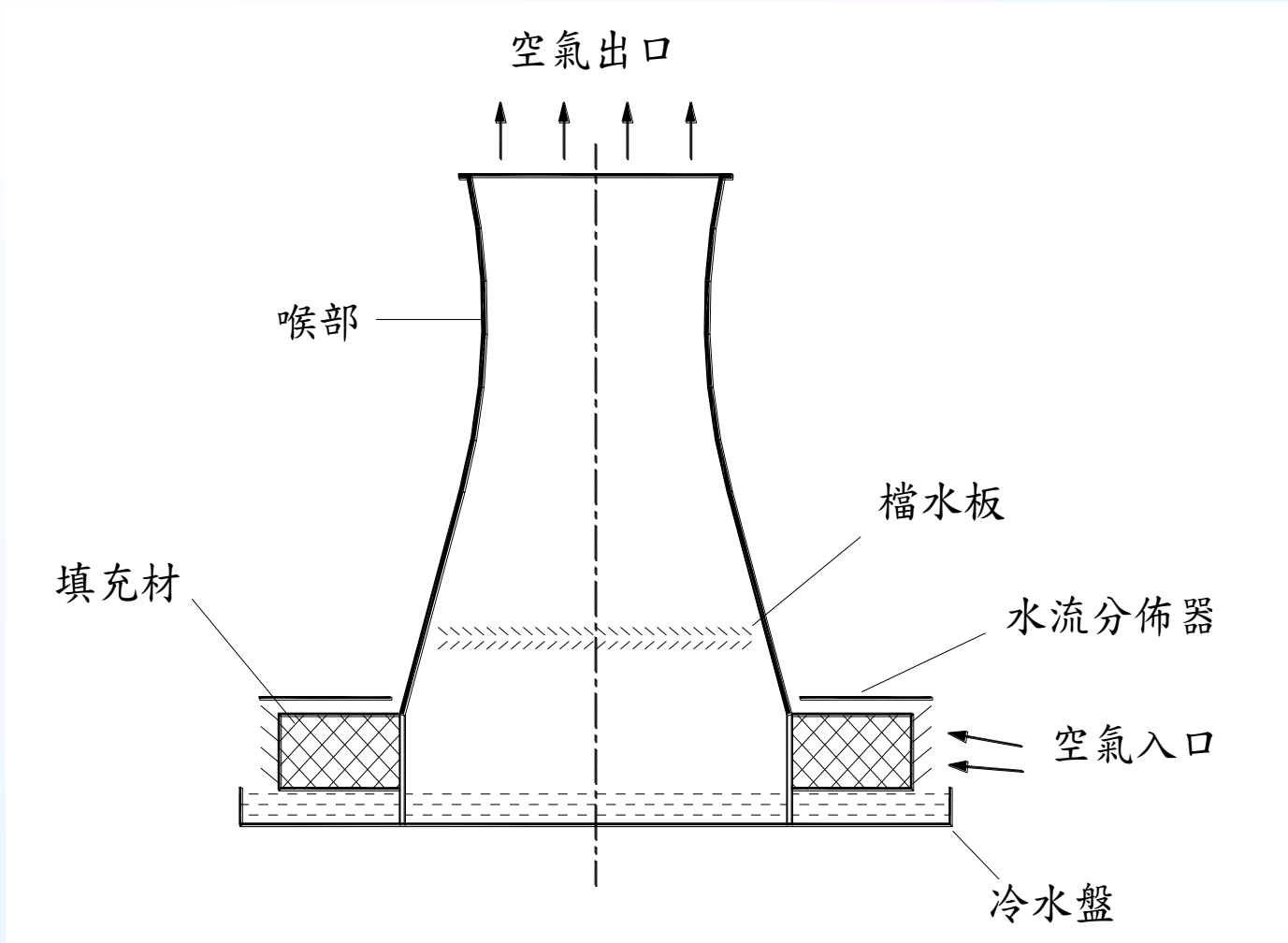


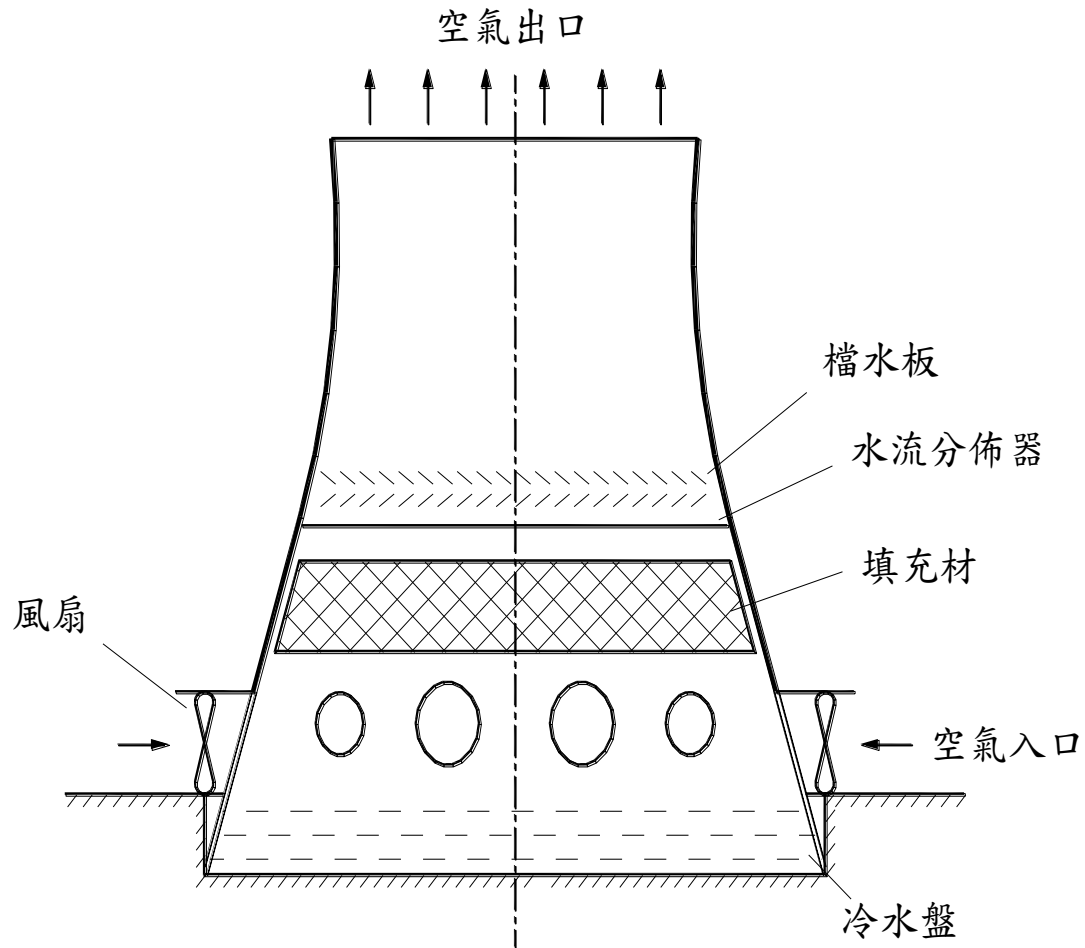


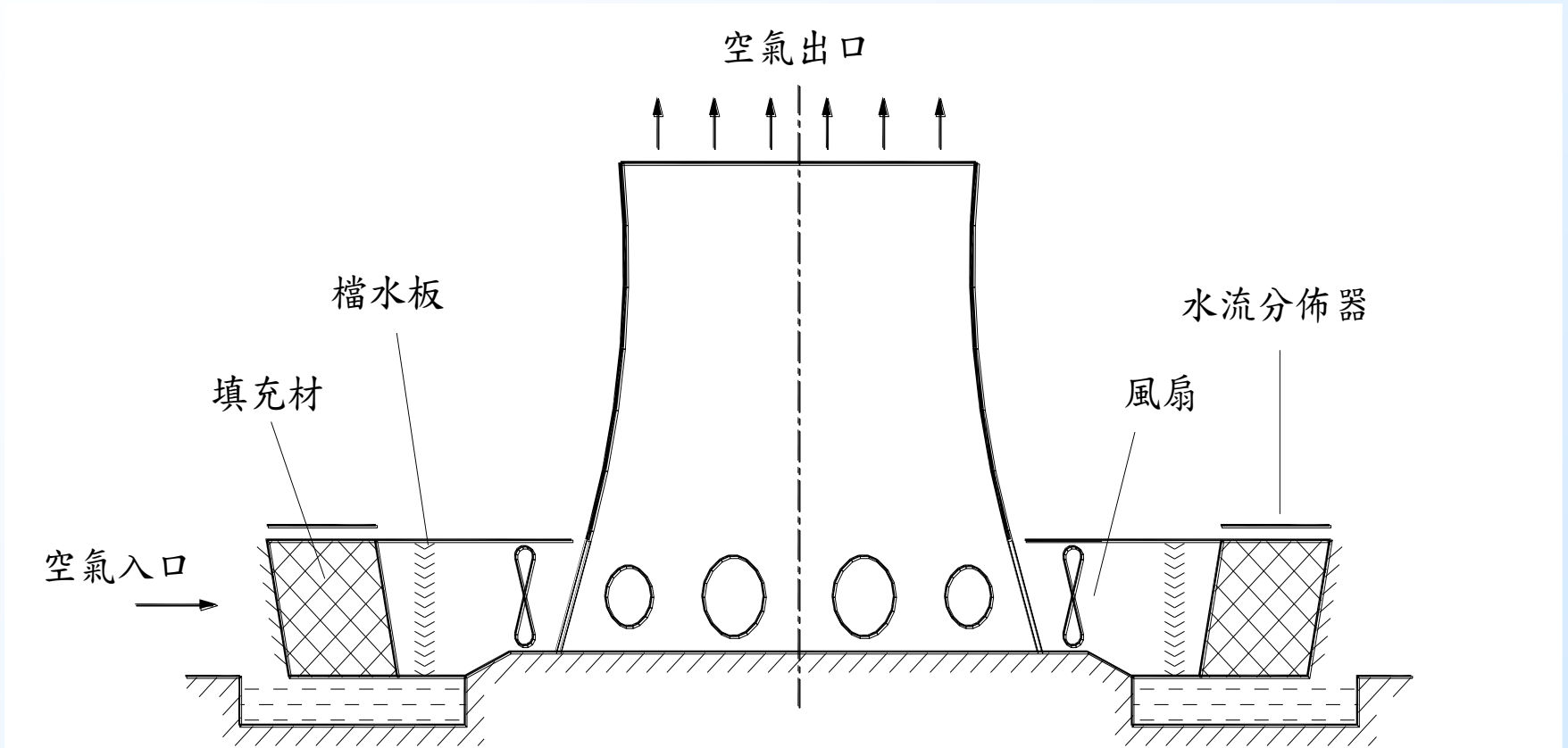






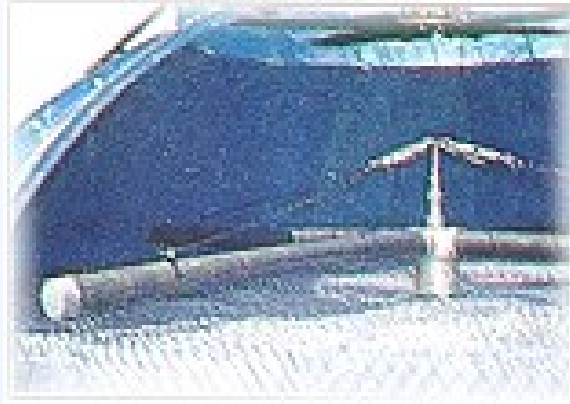




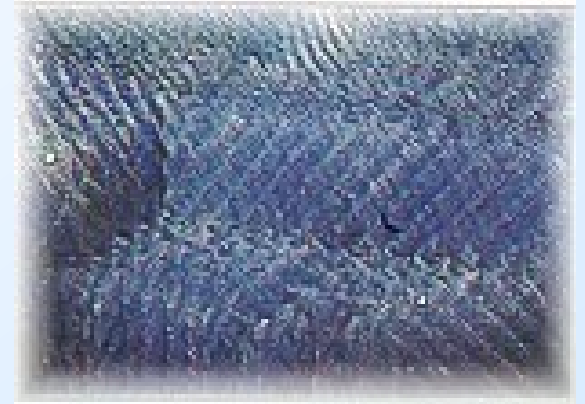




(a)



(b)



(c)

Cooling Tower (a) Fan; (b) distributor; (c) fillet



冷卻水塔的熱流分析

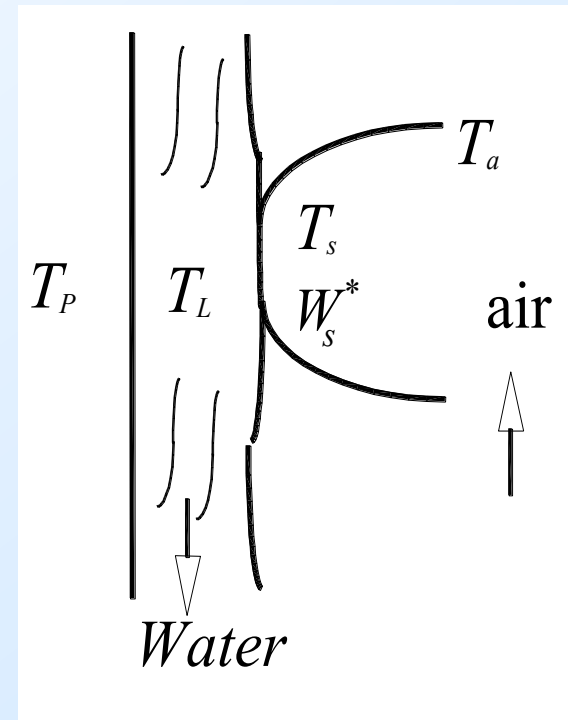
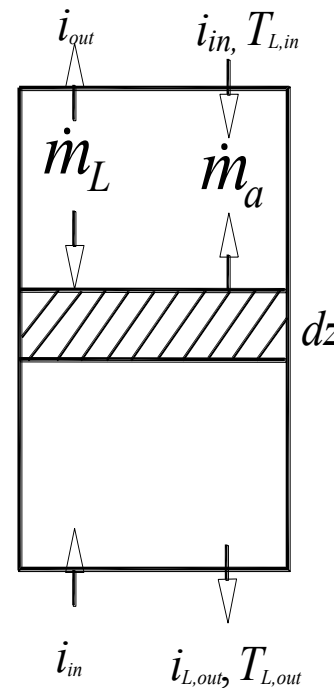
$$dQ = \frac{h_{c,o} dA_o}{c_{p,a}} (i - i_w)$$

再根據路易關係式(Lewis relation)

$$Le = \frac{h_{c,o}}{h_{D,o} c_{p,a}} \approx 1$$

所以式6-36可改寫成

$$dQ = h_{D,o} (i - i_a) dA_o$$





冷卻水塔的熱流分析 (Conti..)

$$dQ = h_{D,o} a (i - i_a) dV \quad (17-2)$$

其中 a 為的單位體積下的有效界面面積 (m^2/m^3)，考慮如圖 17-12 所示，水在界面上的蒸發現象，由於水量蒸發的量與水的流量相比甚小，因此能量的平衡如下：

$$\therefore dQ = i_w \rho_w \dot{V}_w dV \quad (17-3)$$

因此，式 17-2 可以改寫如下 (請注意積分的端點值，由於逆向流的安排，所以空氣入口側為冷卻水的出口側)：

$$\frac{h_{D,o} a V}{i_w \rho_w \dot{V}_w} = \int_{T_{L,out}}^{T_{L,in}} \frac{c_p}{i - i_a} dT \quad (17-4)$$

或

$$\frac{h_{D,o} a V}{i_w \rho_w \dot{V}_w} = \int_{i_{a,in}}^{i_{a,out}} \frac{di}{i - i_a} \quad (17-5)$$



冷卻水塔的熱流分析 (Conti..)

在冷卻水塔的設計習慣上，式17-4的積分結果稱之為 NTU (number of transfer unit)，不過這個 NTU 與先前介紹的熱交換器設計的 NTU 只是形式上

類似， $\frac{h_{D,o}aV}{\dot{m}_L}$ 或 $\frac{h_{D,o}aV}{\dot{m}_a}$ 代表冷卻水塔的「能力尺寸」，一旦冷卻水塔

的質傳特性 $h_{D,o}a$ 為已知，則冷卻水塔的尺寸大小便可算出，此一質傳特性與填充材形狀、安排與流動特性有關，這個積分值代表平均的焓差變化到單位水側溫度的變化；此一積分值或稱之為墨客積分(Merkel Integral, I_M)，亦稱之為填充函數(packaging function)，讀者要特別注意，墨客積分內的 i 為水與空氣界面溫度下對應的空氣飽和焓值(不是水的焓值！)

式17-4或17-5的積分計算，可以使用任何合適的積分法，美國冷卻水塔協會(CTI, 1967)則建議使用Tchebycheff 4點快速積分法來計算，計算過程說明如下：

$$\int_{x_i}^{x_e} y dx = \frac{(x_e - x_i)}{4} (y_1 + y_2 + y_3 + y_4) \quad (17-7)$$



式17-4或17-5的積分計算，可以使用任何合適的積分法，美國冷卻水塔協會(CTI, 1967)則建議使用Tchebycheff 4點快速積分法來計算，計算過程說明如下：

$$\int_{x_i}^{x_e} y dx = \frac{(x_e - x_i)}{4} (y_1 + y_2 + y_3 + y_4) \quad (17-7)$$

其中

y_1 為 $x = x_i + 0.1(x_e - x_i)$ 時的 y 值

y_2 為 $x = x_i + 0.4(x_e - x_i)$ 時的 y 值

y_3 為 $x = x_i + 0.6(x_e - x_i)$ 時的 y 值

y_4 為 $x = x_i + 0.9(x_e - x_i)$ 時的 y 值



因此，式17-4的積分結果可表示如下(積分由空氣入口處到處口處，但由於空氣與水為逆向流動安排設計，因此積分的水溫度是由水出口到水入口)：

$$\int_{T_{L,out}}^{T_{L,in}} \frac{c_p}{i-i_a} dT = \frac{T_{L,in} - T_{L,out}}{4} \left(\frac{c_p}{i-i_a} \Big|_{T=T_1} + \frac{c_p}{i-i_a} \Big|_{T=T_2} + \frac{c_p}{i-i_a} \Big|_{T=T_3} + \frac{c_p}{i-i_a} \Big|_{T=T_4} \right) \quad (17-8)$$

其中

$$T_1 = T_{L,in} + 0.1(T_{L,out} - T_{L,in}) \quad (17-9)$$

$$T_2 = T_{L,in} + 0.4(T_{L,out} - T_{L,in}) \quad (17-10)$$

$$T_3 = T_{L,in} + 0.6(T_{L,out} - T_{L,in}) \quad (17-11)$$

$$T_4 = T_{L,in} + 0.9(T_{L,out} - T_{L,in}) \quad (17-12)$$

由於冷卻水塔的操作過程中， \dot{m}_L 與 \dot{m}_a 為固定值，故其比值 ϕ 亦為定值，因此：

$$di = \phi \cdot di_L = \phi \cdot c_{p,L} \cdot dT_L \quad (17-13)$$



冷卻水塔設計最重要的兩個參數是 $\phi = \frac{\dot{m}_L}{\dot{m}_a}$ 與冷卻水塔的體積大小(或是水

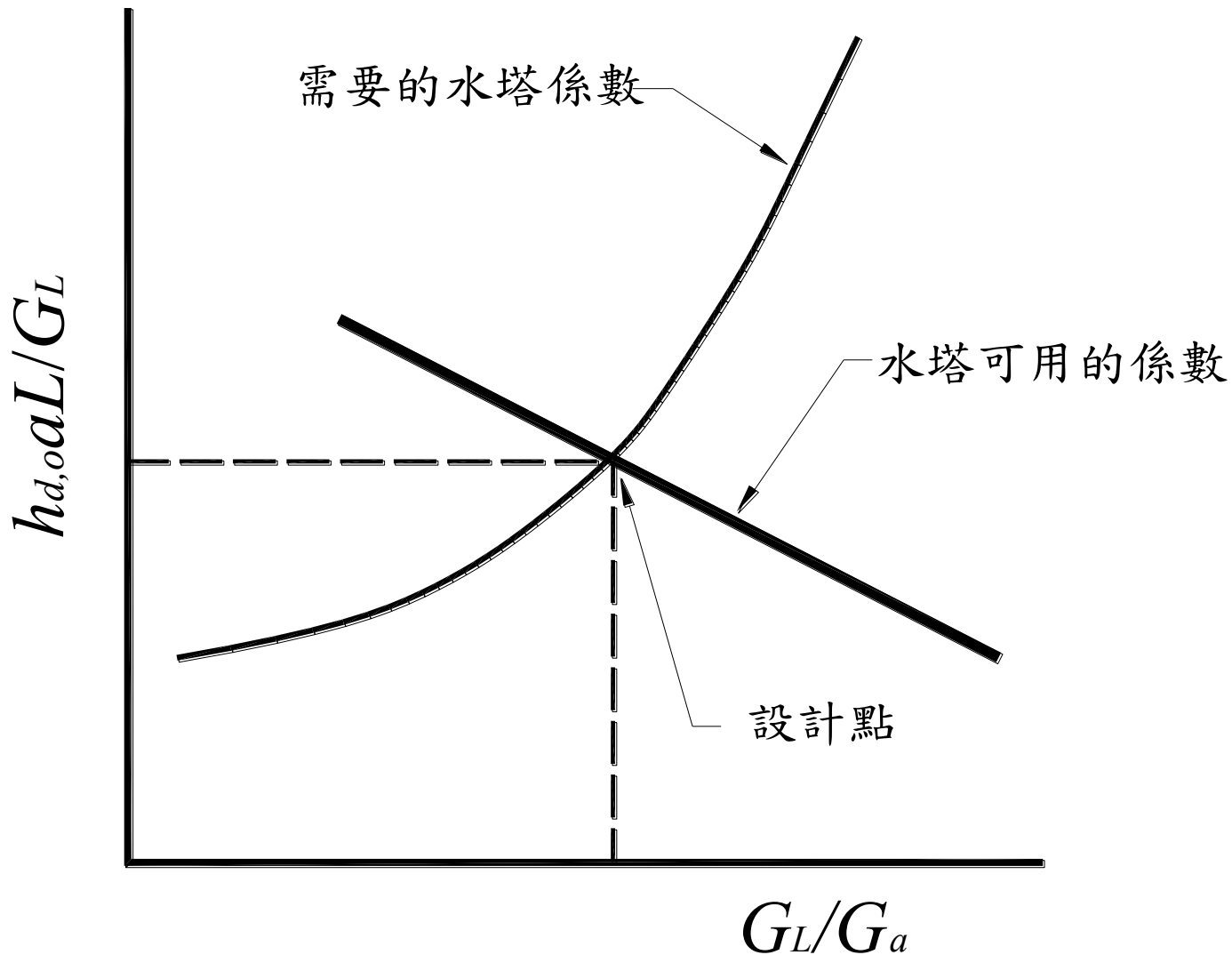
塔的高度 H ，如果水塔的截面積為已知)，然而更實際的來講，一般冷卻水流量可能為已知(例如空調主機系統的設計已確定，冷卻水塔的目的僅在於完成冷凝器水側的散熱量)，在上一個章節中，我們介紹當空氣的進口濕球

條件與 $\phi = \frac{\dot{m}_L}{\dot{m}_a}$ (或是 G_L/G_a) 固定時，可以算出這個條件下的所需要的

NTU ，由於一般設計上的濕球溫度均為給定，因此真正的彈性在於變動

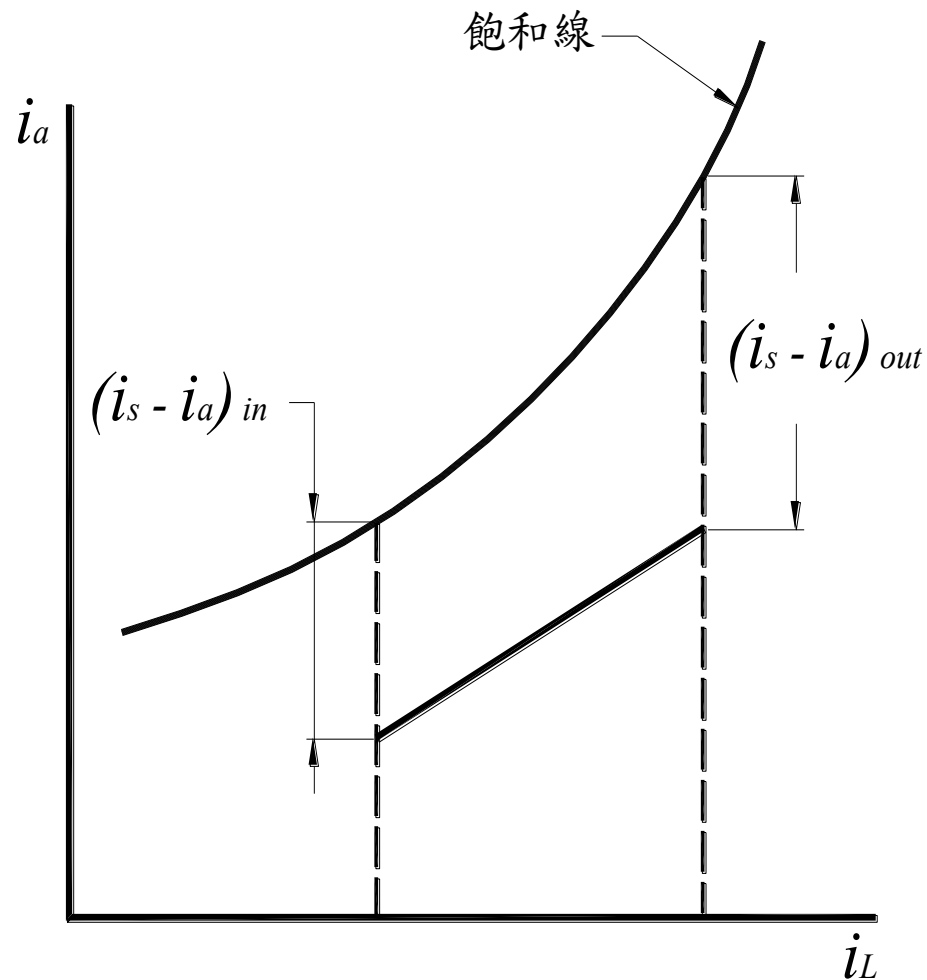
$\phi = \frac{\dot{m}_L}{\dot{m}_a}$ 以滿足設計上的需求，因此我們可變動 $\phi = \frac{\dot{m}_L}{\dot{m}_a}$ 算出冷卻水塔在濕球

溫度固定時所需要的 NTU 的曲線



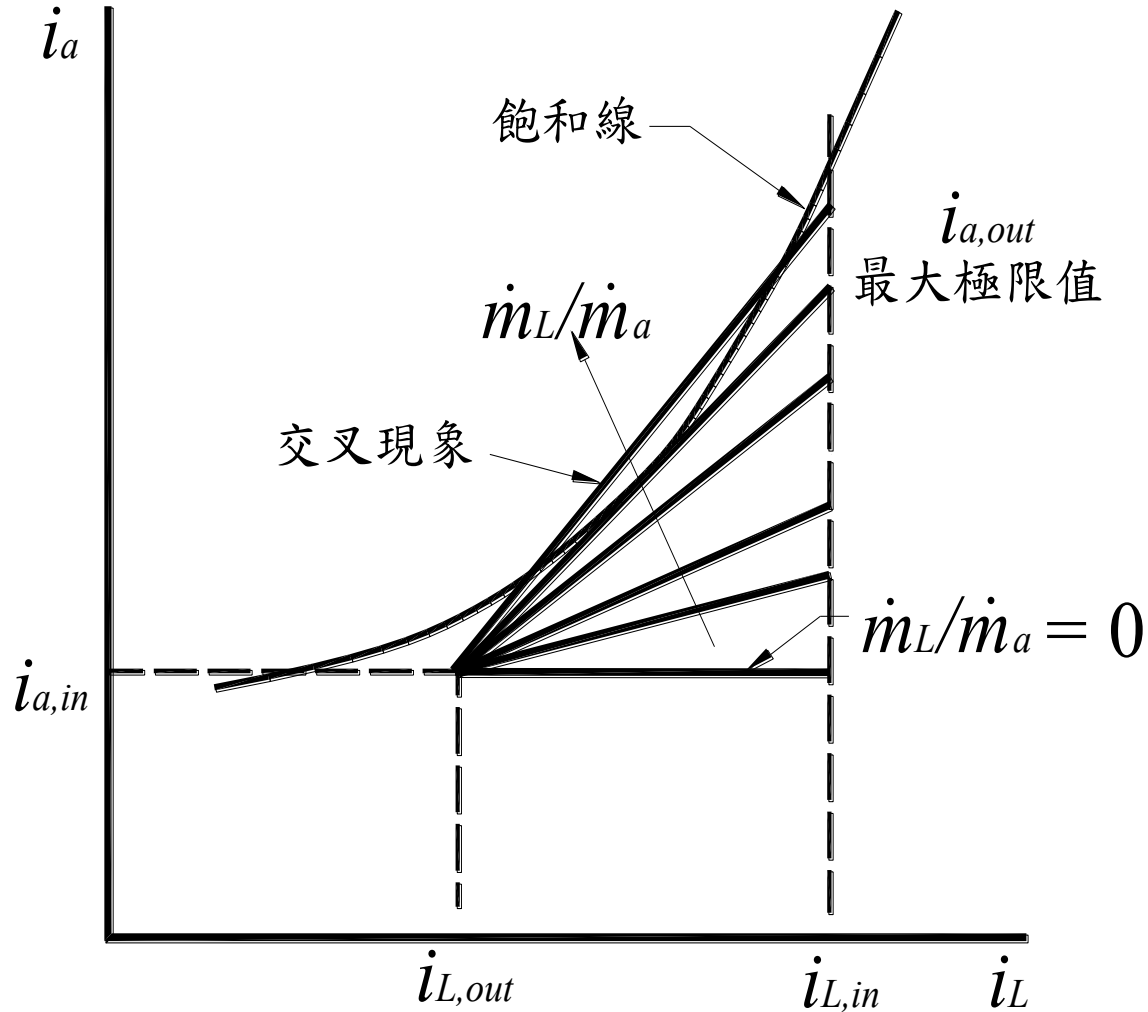


空氣與界面上的驅動勢(driving potential)與冷卻水焓的關係





Limits of operation line





冷卻水塔的特性方程式

$$\frac{h_{D,o} a V}{i_{a,m} A_{fr}} = \frac{h_{D,o} a A_{fr} H}{i_{a,m} A_{fr}} = \frac{h_{D,o} a H}{i_{a,m}} = \frac{h_{D,o} a H}{G_L} \quad (17-14)$$

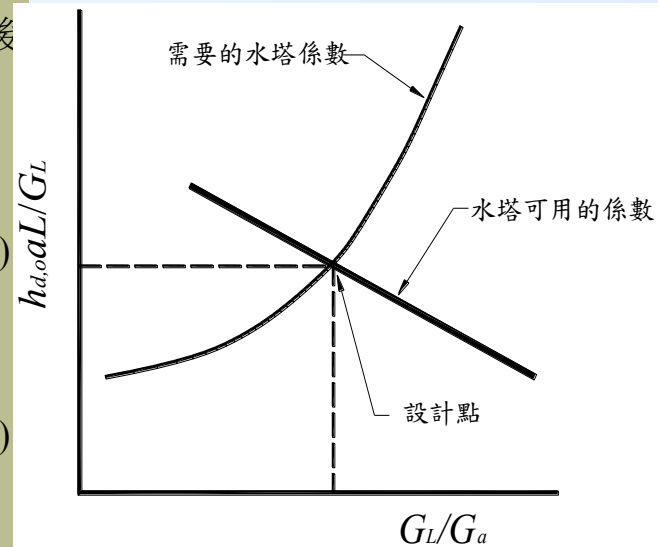
其中 A_{fr} 為填充材的進口截面積，而水塔特性通常可由實驗測試整理後歸納出如下的關係式：

$$\frac{h_{D,o} a}{G_L} = a_d \left(\frac{G_L}{G_a} \right)^{-b_d} \quad (17-15)$$

$$K = a_p \left(\frac{G_L}{G_a} \right) + b_p \quad (17-16)$$

其中 K 為單位長度的壓損係數，空氣通過冷卻水塔的總壓降可計算如下：

$$\Delta P = KH \frac{i_{a,m}}{2\rho_{a,m}} \quad (17-17)$$





評估冷卻水塔

Measured Parameters

- Wet bulb temperature of air
- Dry bulb temperature of air
- Cooling tower inlet water temperature
- Cooling tower outlet water temperature
- Exhaust air temperature
- Electrical readings of pump and fan motors
- Water flow rate
- Air flow rate



評估冷卻水塔

Performance Parameters

1. Range
2. Approach
3. Effectiveness
4. Cooling capacity
5. Evaporation loss
6. Cycles of concentration
7. Blow down losses
8. Liquid / Gas ratio



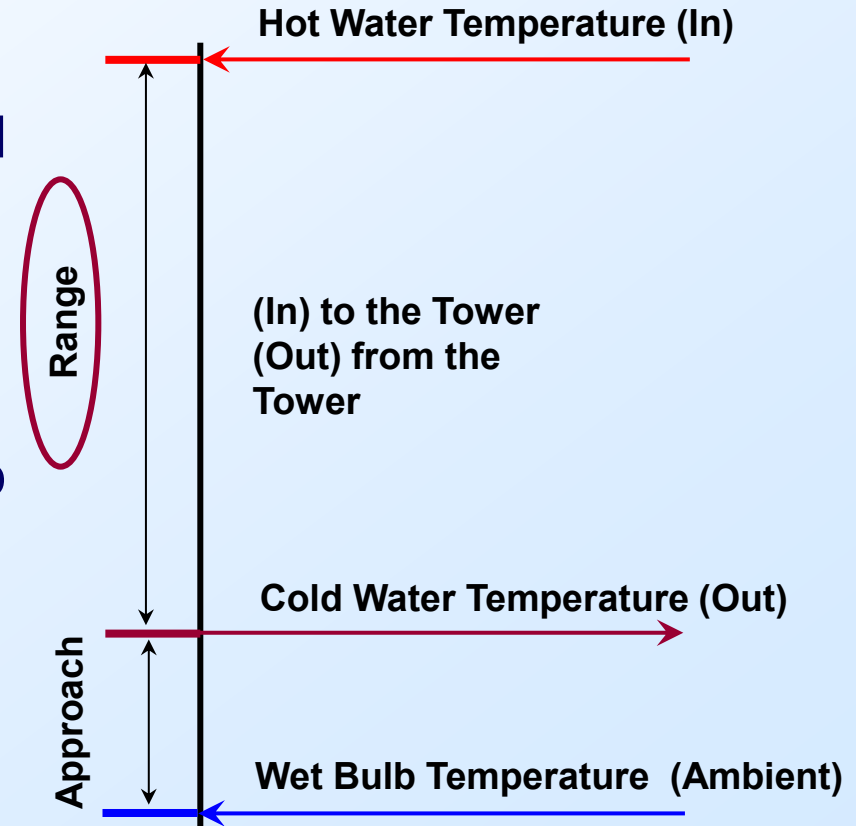
評估冷卻水塔

1. Range

Difference between cooling water inlet and outlet temperature:

Range ($^{\circ}\text{C}$) = CW inlet temp – CW outlet temp

High range = good performance





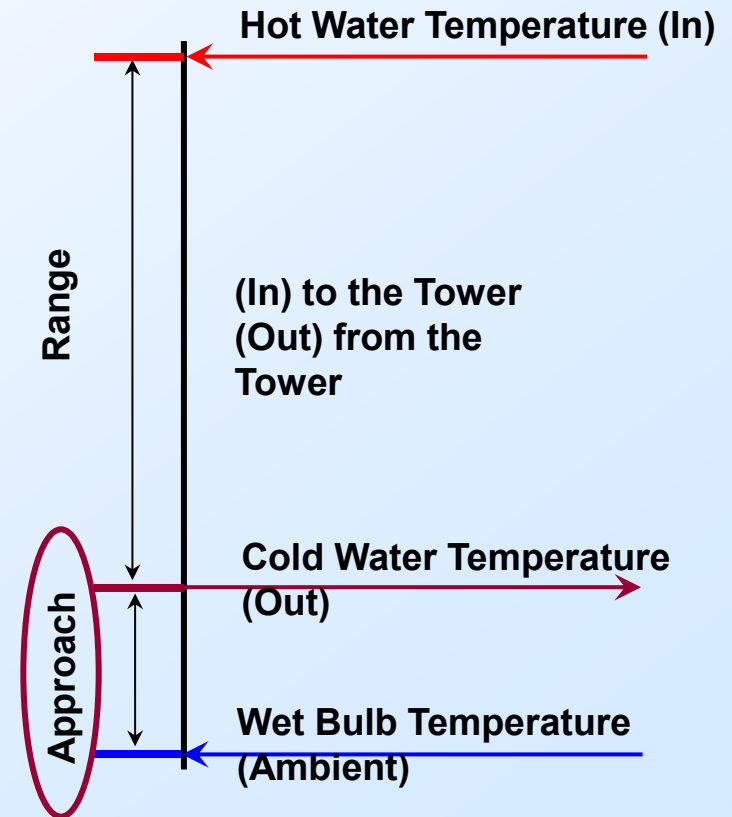
評估冷卻水塔

2. Approach

Difference between cooling tower outlet cold water temperature and ambient wet bulb temperature:

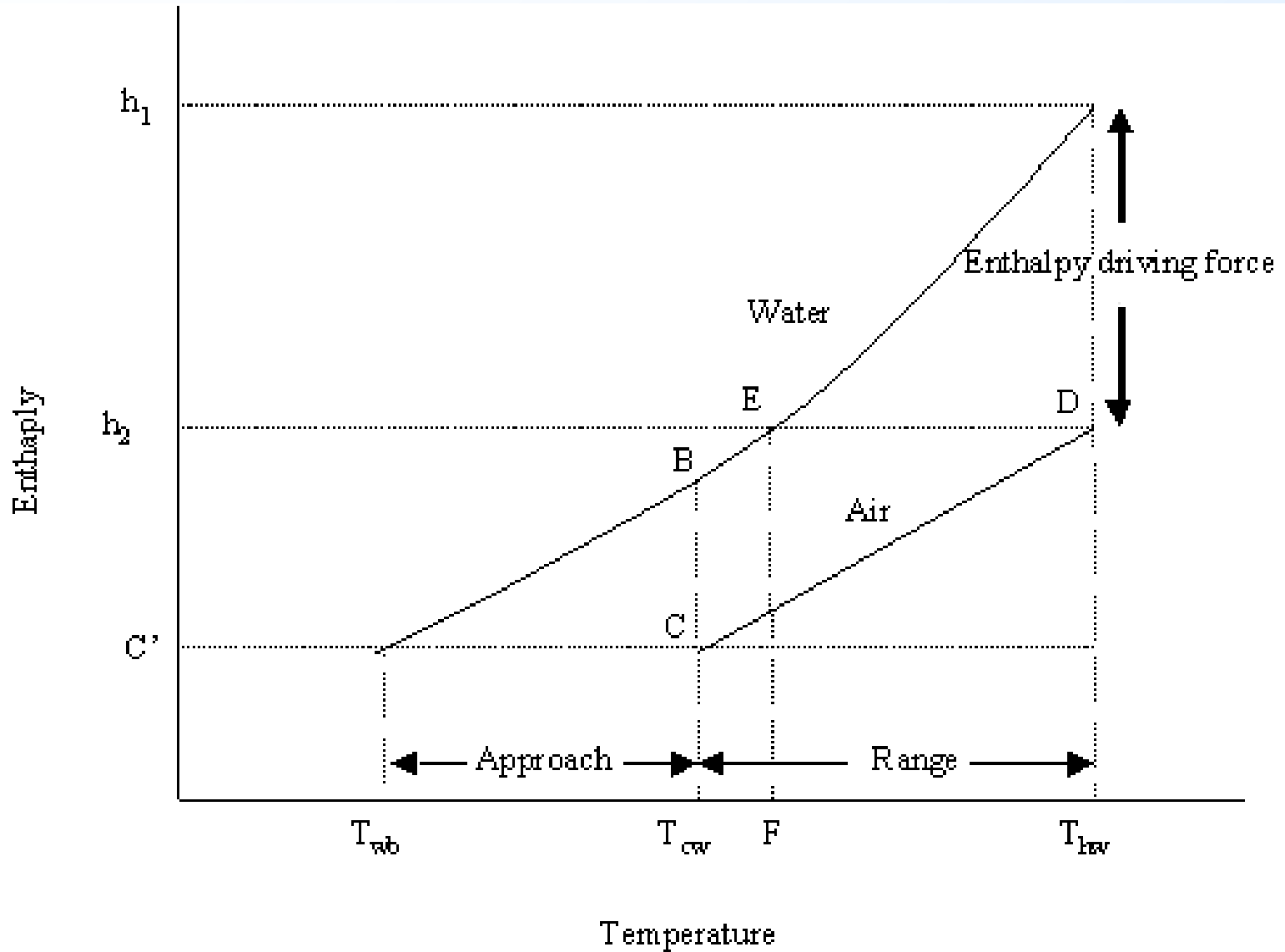
Approach ($^{\circ}\text{C}$) =
CW outlet temp – Wet bulb temp

Low approach = good performance





Temperature Enthalpy Diagram of Air Water System





評估冷卻水塔

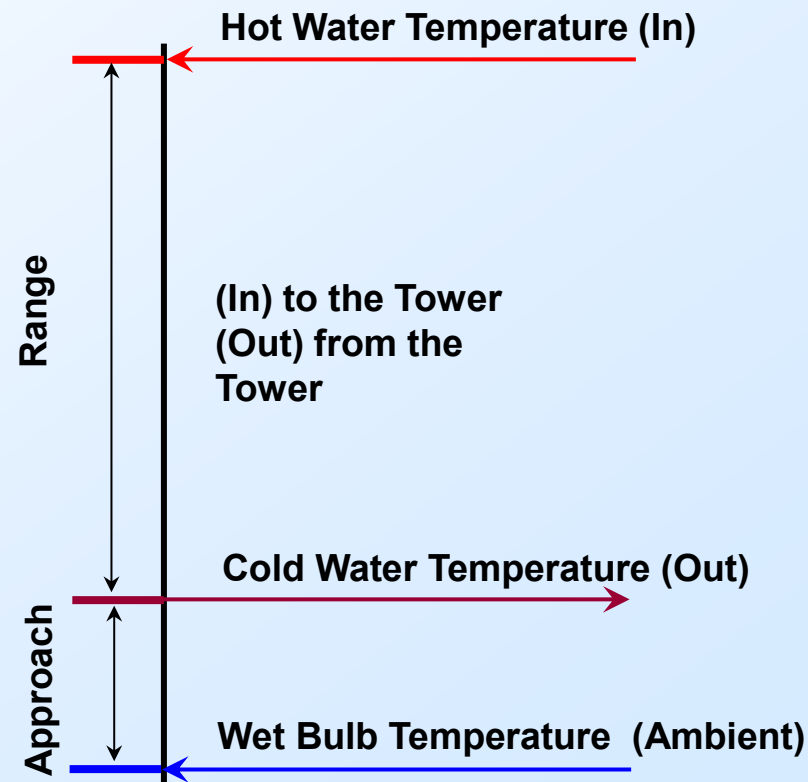
3. Effectiveness

Effectiveness in %

= Range / (Range + Approach)

= 100 x (CW temp – CW out temp) / (CW in temp – Wet bulb temp)

High effectiveness = good performance





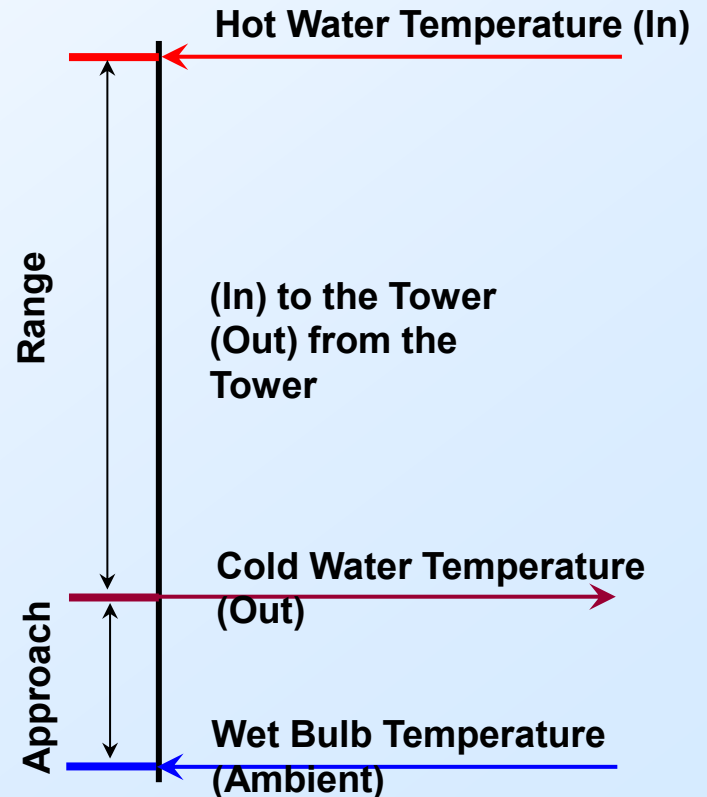
評估冷卻水塔

4. Cooling Capacity

Heat rejected in kW or
tons of refrigeration (TR)

= mass flow rate of water
X specific heat X
temperature difference

High cooling capacity =
good performance





評估冷卻水塔

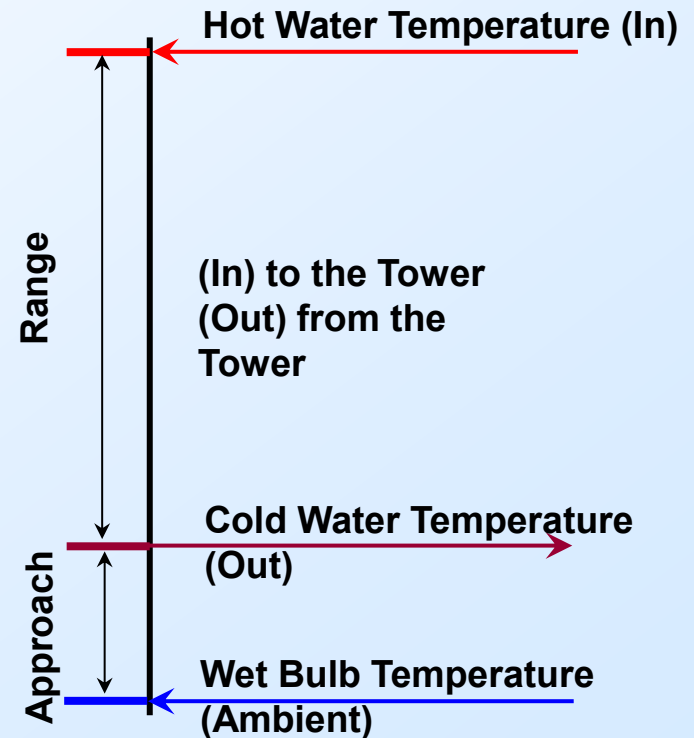
5. Evaporation Loss

Water quantity (m^3/hr)
evaporated for cooling duty

= theoretically, 1.8 m^3 for every
 $10,000,000 \text{ kCal}$ heat rejected

= $0.00085 \times 1.8 \times$ circulation
rate (m^3/hr) $\times (T1-T2)$

$T1-T2 =$ Temp. difference
between inlet and outlet water





6. Cycles of concentration (C.O.C.)

Ratio of dissolved solids in circulating water to the dissolved solids in make up water.

Cycles of concentration represents the accumulation of dissolved minerals in the recirculating cooling water. Discharge of draw-off (or blowdown) is used principally to control the buildup of these minerals.

The chemistry of the make-up water, including the amount of dissolved minerals, can vary widely. Make-up waters low in dissolved minerals such as those from surface water supplies (lakes, rivers etc.) tend to be aggressive to metals (corrosive). Make-up waters from ground water supplies (such as wells) are usually higher in minerals, and tend to be scaling (deposit minerals). Increasing the amount of minerals present in the water by cycling can make water less aggressive to piping; however, excessive levels of minerals can cause scaling problems.

7. Cycles of concentration (C.O.C.)

Depend on cycles of concentration and the evaporation losses

Blow Down = Evaporation Loss / (C.O.C. – 1). The portion of the circulating water flow that is removed (usually discharged to a drain) in order to maintain the amount of Total Dissolved Solids (TDS) and other impurities at an acceptably low level.



評估冷卻水塔

8. Liquid Gas (L/G) Ratio

Ratio between water and air mass flow rates

Heat removed from the water must be equal to the heat absorbed by the surrounding air

$$L(T1 - T2) = G(h2 - h1)$$

$$L/G = (h2 - h1) / (T1 - T2)$$

T1 = hot water temp (°C)

T2 = cold water temp (°C)

Enthalpy of air water vapor mixture at inlet wet bulb temp (h1)
and outlet wet bulb temp (h2)



Energy Efficiency Opportunities

1. Selecting a cooling tower

Capacity

- Heat dissipation (kW)
- Circulated flow rate (m^3/hr)
- Other factors



Energy Efficiency Opportunities

1. Selecting a cooling tower

Range

- Range determined by process, not by system

Approach

- Closer to the wet bulb temperature
 - → Bigger size cooling tower
 - → More expensive



Energy Efficiency Opportunities

1. Selecting a cooling tower

Heat Load

- Determined by process
- Required cooling is controlled by the desired operating temperature
- High heat load = large size and cost of cooling tower



Energy Efficiency Opportunities

1. Selecting a cooling tower

Wet bulb temperature – considerations:

- Water is cooled to temp higher than wet bulb temp
- Conditions at tower site
- Not to exceed 5% of design wet bulb temp
- Is wet bulb temp specified as ambient (preferred) or inlet
- Can tower deal with increased wet bulb temp
- Cold water to exchange heat



Energy Efficiency Opportunities

1. Selecting a cooling tower

Relationship range, flow and heat load

- Range increases with increased
 - Amount circulated water (flow)
 - Heat load
- Causes of range increase
 - Inlet water temperature increases
 - Exit water temperature decreases
- Consequence = larger tower



Energy Efficiency Opportunities

1. Selecting a cooling tower

Relationship Approach and Wet bulb temperature

- If approach stays the same (e.g. 4.45°C)
- Higher wet bulb temperature (26.67°C)
 - = more heat picked up (15.5 kCal/kg air)
 - = smaller tower needed
- Lower wet bulb temperature (21.11°C)
 - = less heat picked up (12.1 kCal/kg air)
 - = larger tower needed



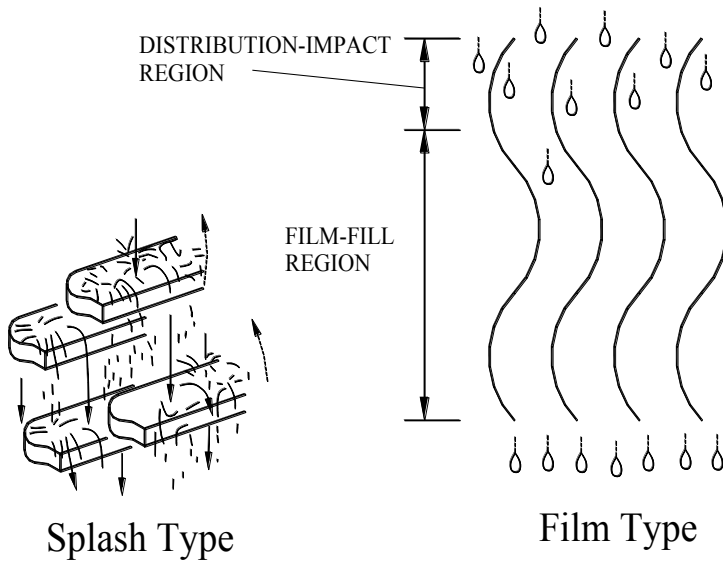
Energy Efficiency Opportunities

2. Fill media

- **Hot water distributed over fill media and cools down through evaporation**
- **Fill media impacts electricity use**
 - **Efficiently designed fill media reduces pumping costs**
 - **Fill media influences heat exchange: surface area, duration of contact, turbulence**



Energy Efficiency Opportunities



2. Fill media

Comparing 3 fill media: film fill more efficient

	<i>Splash Fill</i>	<i>Film Fill</i>	<i>Low Clog Film Fill</i>
Possible L/G Ratio	1.1 – 1.5	1.5 – 2.0	1.4 – 1.8
Effective Heat Exchange Area	30 – 45 m ² /m ³	150 m ² /m ³	85 - 100 m ² /m ³
Fill Height Required	5 – 10 m	1.2 – 1.5 m	1.5 – 1.8 m
Pumping Head Requirement	9 – 12 m	5 – 8 m	6 – 9 m
Quantity of Air Required	High	Much Low	Low

(BEE India, 2004; Ramarao; and Shivaraman)



Energy Efficiency Opportunities

3. Pumps and water distribution

- **Pumps: see pumps session**
- **Optimize cooling water treatment**
 - Increase cycles of concentration (COC) by cooling water treatment helps reduce make up water
 - Indirect electricity savings
- **Install drift eliminators**
 - Reduce drift loss from 0.02% to only 0.003 – 0.001%



Energy Efficiency Opportunities

4. Cooling Tower Fans

- Fans must overcome system resistance, pressure loss: impacts electricity use
- Fan efficiency depends on blade profile
 - Replace metallic fans with FBR blades (20-30% savings)
 - Use blades with aerodynamic profile (85-92% fan efficiency)



Summary – 直接接觸熱傳

- 直接接觸為非常有效的熱交換技術。
- 通常除了顯熱的熱傳外，常伴隨潛熱(質量傳遞)熱傳。
- 直接接出熱交換器通常必須使用填料來增加流體的接觸面積與時間。填料通常不參與熱交換。
- 污染為直接接觸式熱交換器的最大問題。

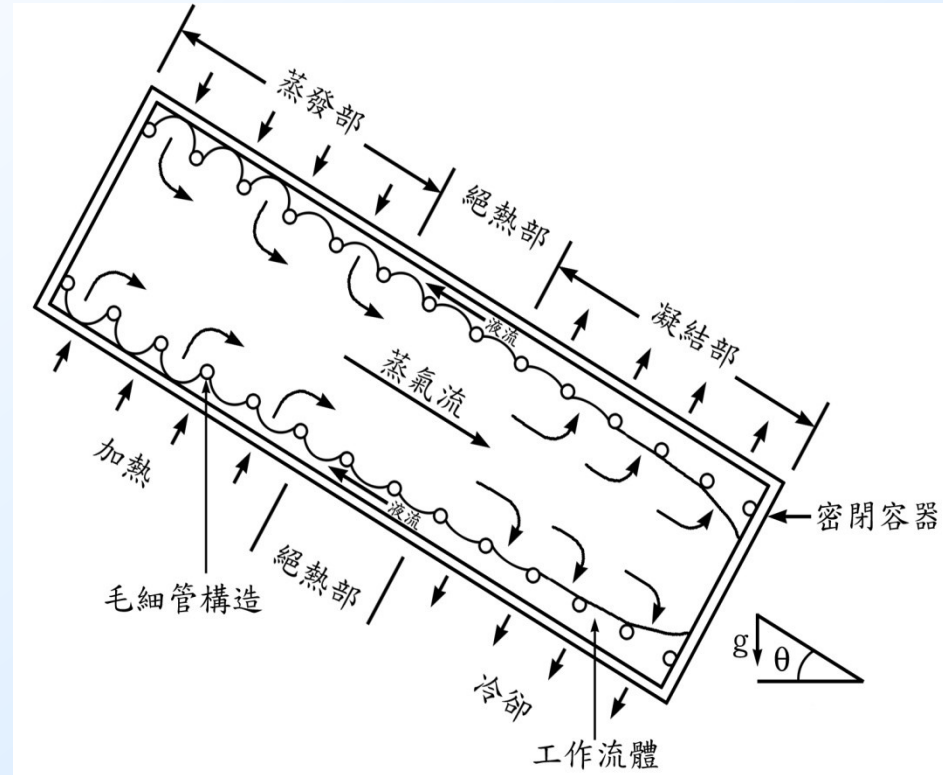


熱管之原理、製作與應用



熱管的組成主要有三項基本要素

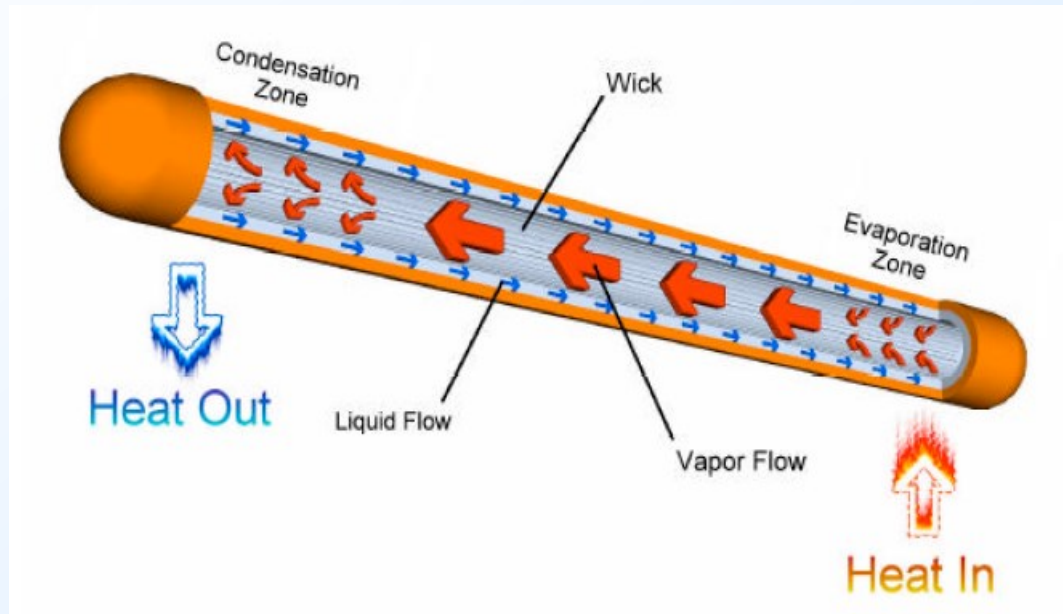
- 密閉容器(container)
- 毛細管構造(wick)
- 工作流體(working fluid)





熱管特性

- 較大的傳熱能力
- 較高的等溫性
- 熱響應迅速
- 具有熱流密度變換功能
- 重量輕
- 無可動件





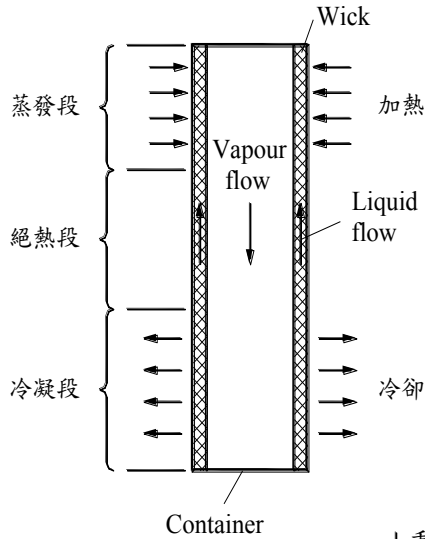
熱傳導係數

材 質	熱 傳 導 係 數
熱 管	50,000 – 200,000 W/m °C
鋁	180 W/m °C
銅	380 W/m °C
鑽石	2,000 W/m °C

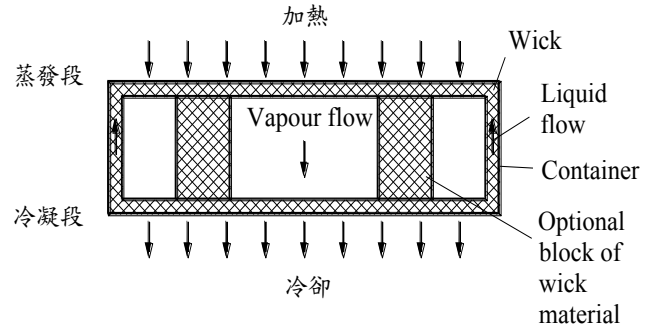


熱管種類

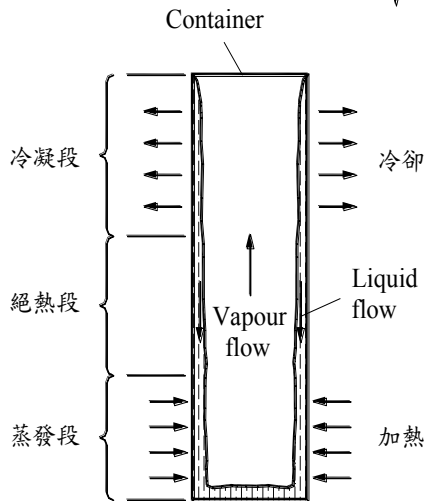
(I) 毛細現象驅動型熱管



(II) Vapor chamber

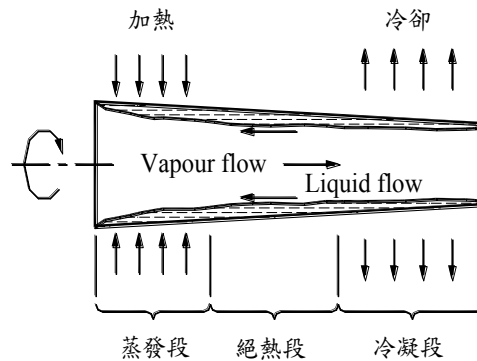


(III) 熱虹吸重力型熱管



重力方向

(IV) 旋轉式離心力驅動型熱管





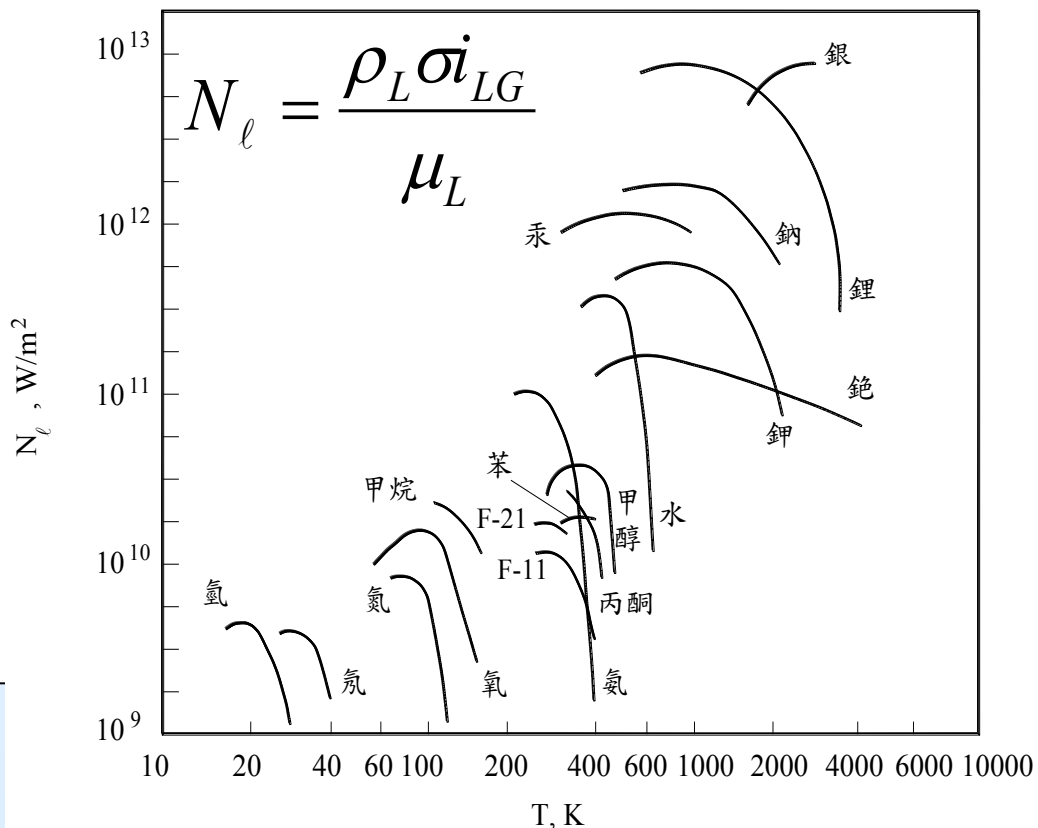
熱管優點

- 沒有運動部件，每根熱管都是永久性密封的，沒有額外的能量消耗，可大大提升操作的可靠性。
- 熱管熱交換器的結構可使用逆流設計，而且熱管本身的溫降很小，近乎等溫運作，故熱交換效率高。
- 由於冷、熱流體都在熱管外表面流過，所以容易用增加鰭片的方法來提高冷、熱流體與熱管表面的對流熱傳性能。
- 每根熱管完全獨立，容易更換，管排寬度及熱管外表面鰭片高度和間距可以根據性能要求及維修、清洗的要求，進行適當的選擇。
- 設備的傳熱性能為可逆安排設計，即冷、熱流體可以變換，這對空調系統的節能十分有利。
- 結構比較緊湊密集，單位體積的傳熱面積大，通常在流動方向上熱交換器的尺寸不大於**500 mm**。
- 即使於冷、熱氣流間溫差很小（如僅十幾度）的情況，也能得到一定的熱回收效率。
- 溫度低於露點，熱管熱交換器仍可以適用，例如用於溶劑的回收。
- 壽命長。



熱管工作流體的特性

- 熱傳性能好
- 與管蕊、管殼材料能長期相容
- 化學組成穩定
- 適中的飽和蒸氣壓
- 熱傳導係數高
- 潤濕性能好





工作流體與適用溫度範圍

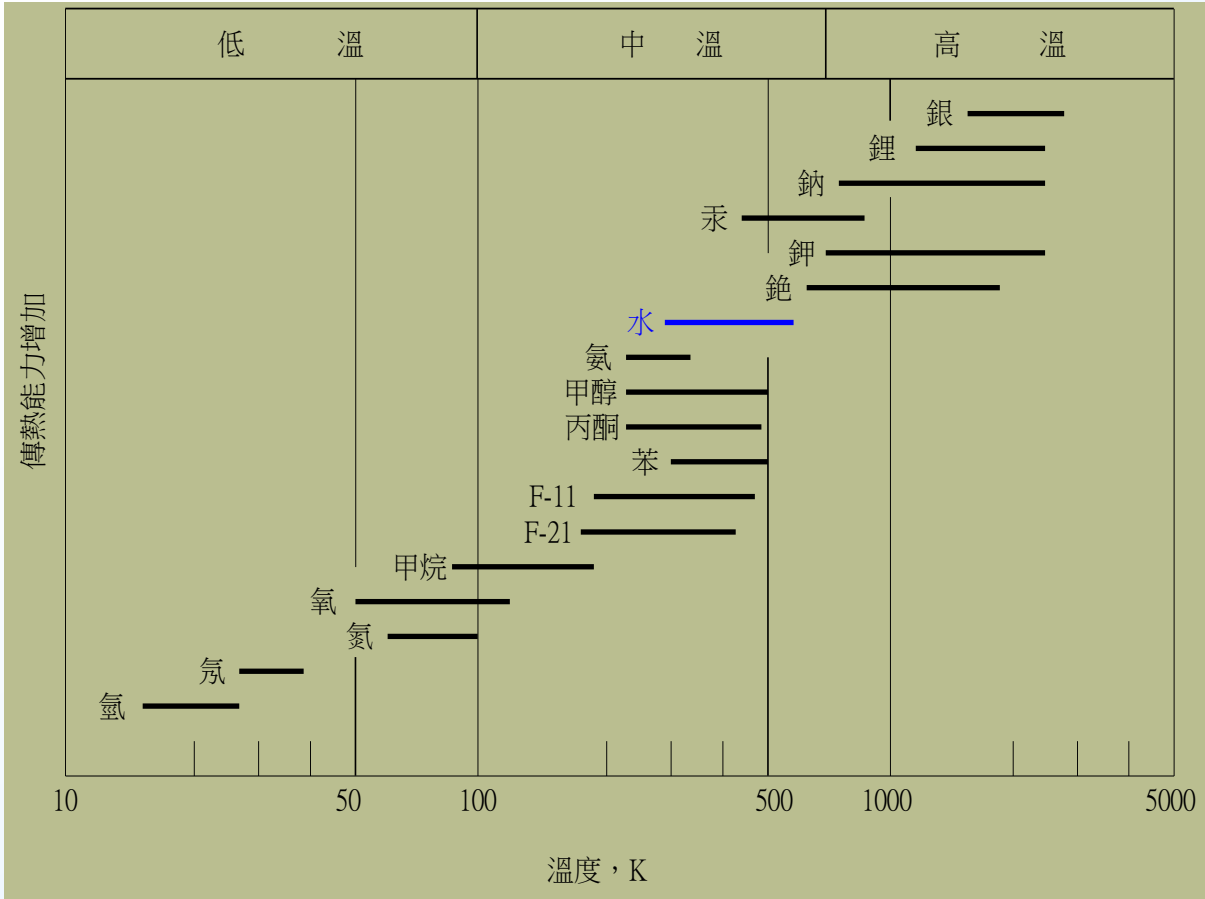


Figure of Merit

$$M = \frac{\rho_l \sigma_l h_{fg}}{\mu_l}$$

ρ : 密度

σ : 表面張力

H_{fg} : 液汽相變潛熱

μ : 黏性係數

M值越大越佳

M值是溫度的函數



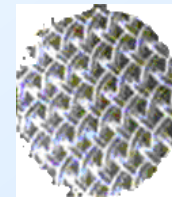
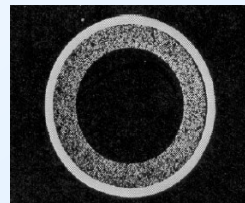
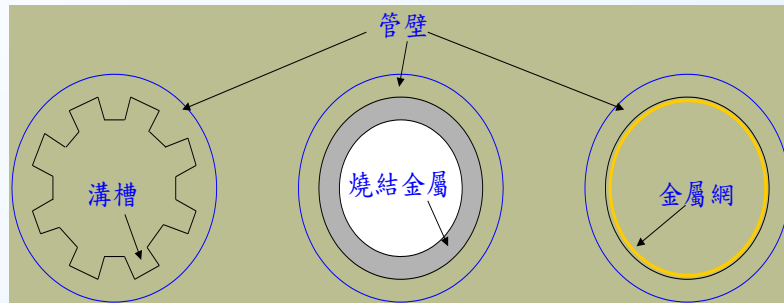
熱管工作流體與材料不相容的原因與影響

影 響	原 因
熱管熱阻增大	<ol style="list-style-type: none">1. 反應產物沉積，使熱傳係數減小2. 由於化學反應產生不凝結氣體3. 管殼、管蕊、工作流體的出氣4. 工作流體分解產生不凝結氣體
熱管傳熱能力下降	<ol style="list-style-type: none">1. 管蕊被固體顆粒堵住，流動阻力增大2. 管蕊內有氣泡存在，使流動阻力增加或使液流中斷3. 由於化學反應使管蕊潤濕能力下降4. 由於工作流體中溶解了反應產物或金屬離子，使表面張力變小5. 工作流體中溶解了反應物使粘度增加6. 管蕊被腐蝕，無法輸送液體
管殼損壞	<ol style="list-style-type: none">1. 管殼的電化學腐蝕2. 管殼材料溶解在工作流體中

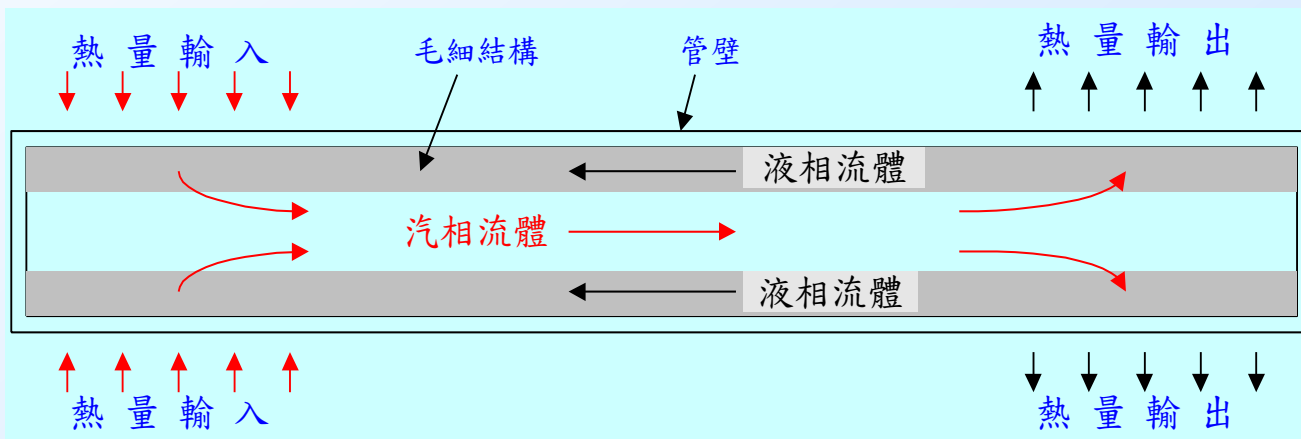


熱管毛細結構

- ▶ 管殼
- ▶ 工作流體
- ▶ 毛細結構

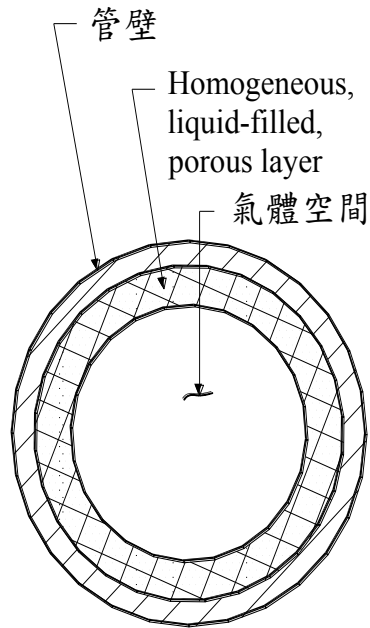


流體相變化傳熱

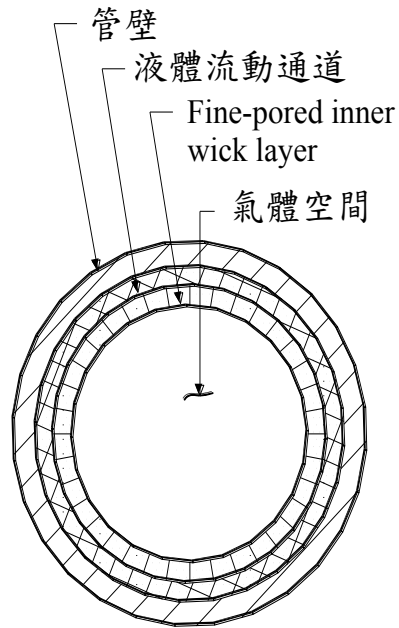




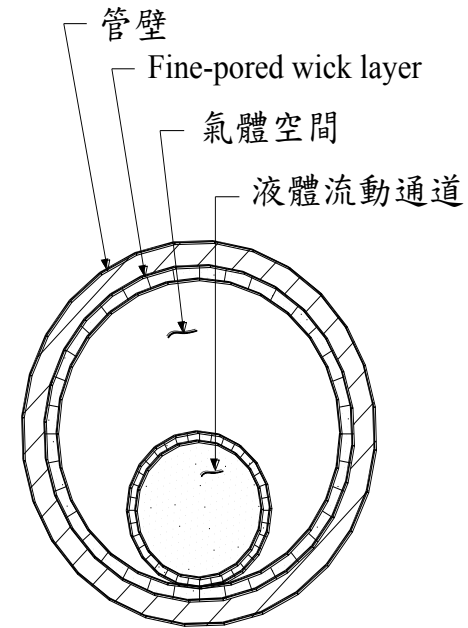
管蕊形式



單層管蕊



雙層管蕊

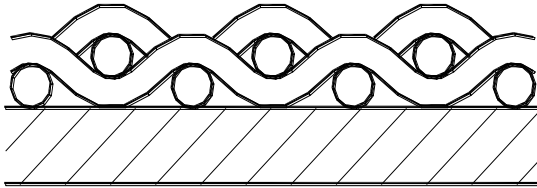


幹道式管蕊

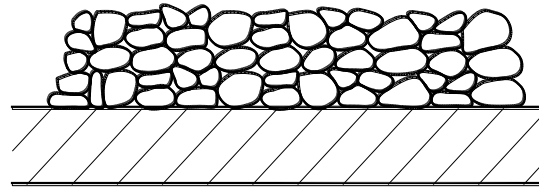
普通熱管中管蕊的樣式可分為二類，即均勻管蕊和組合管蕊。常用的均勻管蕊有編織層網、軸向槽道、燒結蕊等；這類管蕊大多具有結構簡單的特點，但性能稍差。組合管蕊的結構較複雜，但具有較好的傳熱性能與較小的熱阻



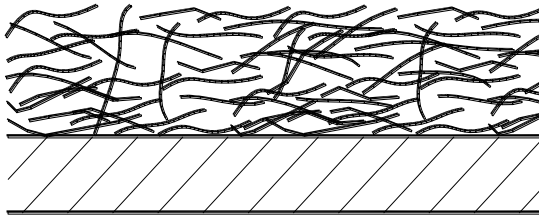
常用的均勻管蕊



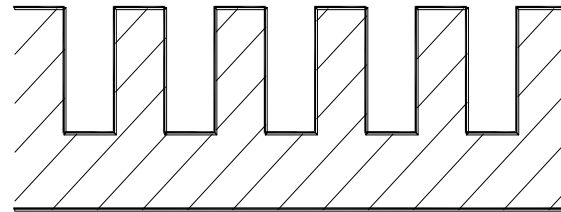
WOVEN MESH SCREEN



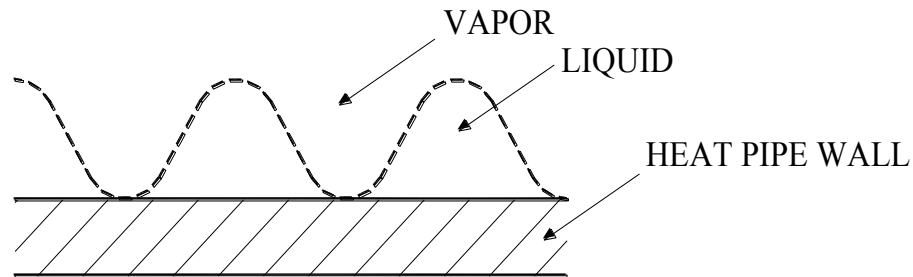
SINTERED METAL POWDER



SINTERED METAL FIBERS



GROOVES IN HEAT PIPE WALL

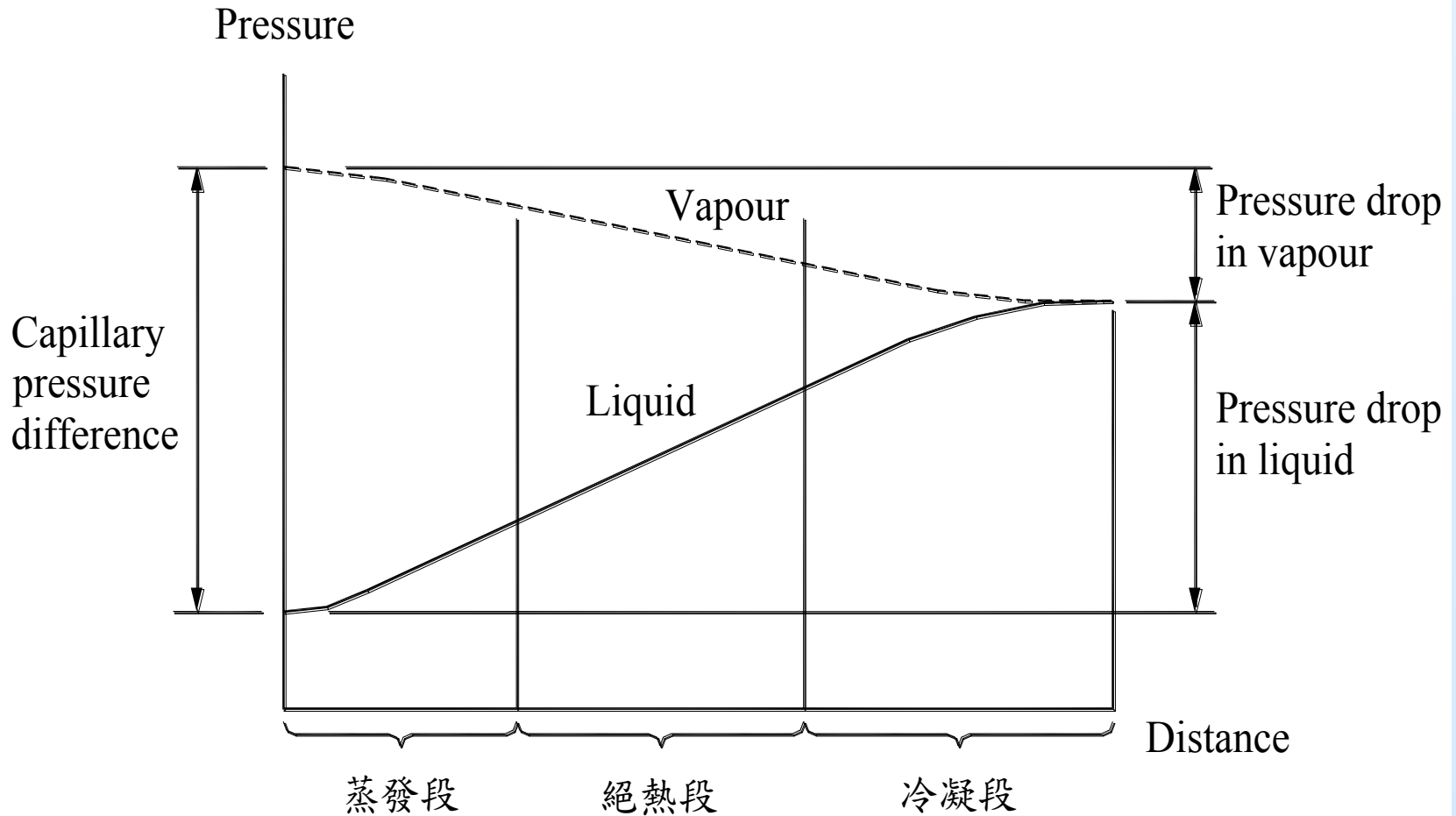


SLOTTED METAL SHEET



熱管內壓力變化

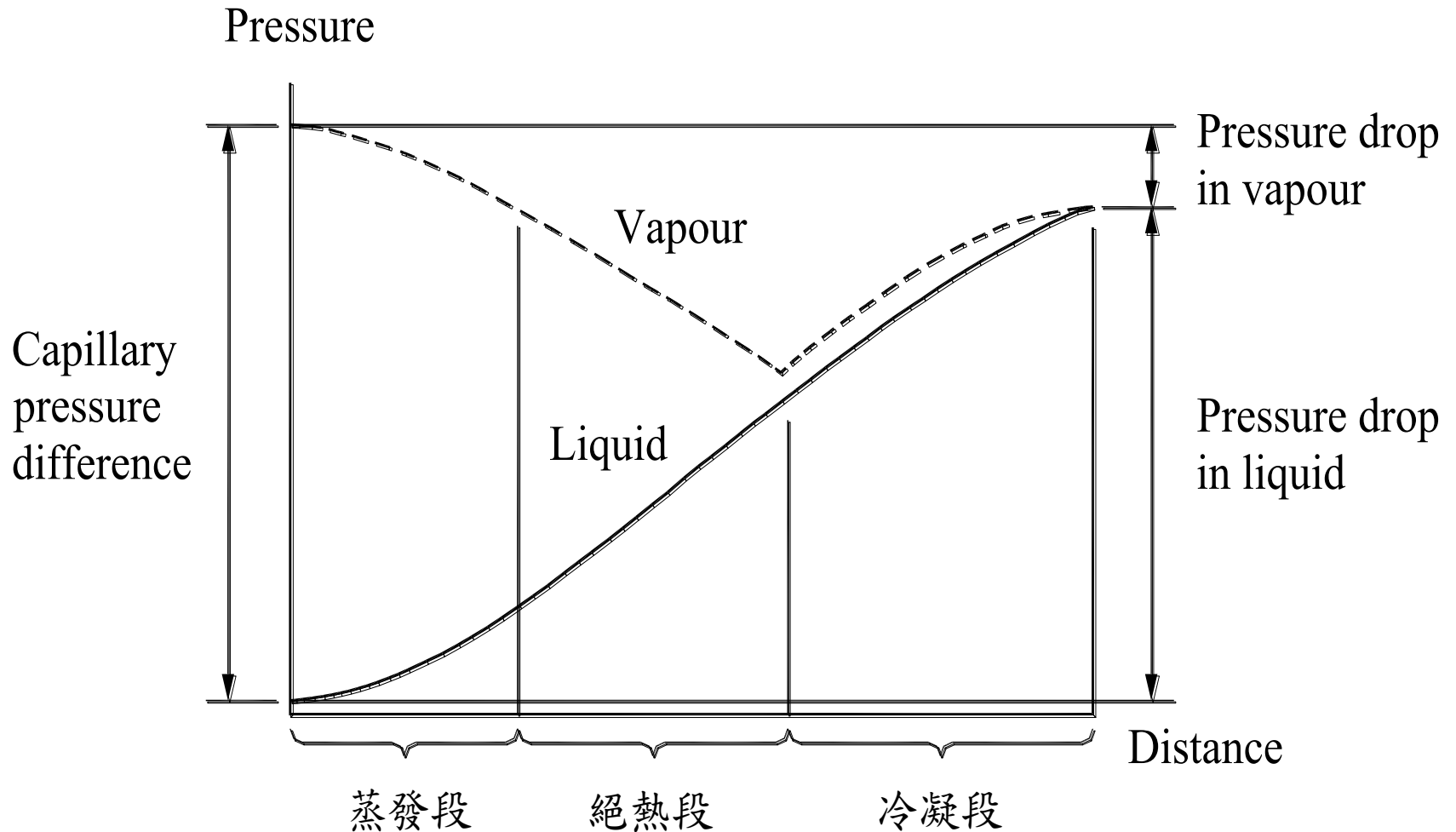
較低氣體速度下，熱管內的氣液的壓力分布





熱管內壓力變化

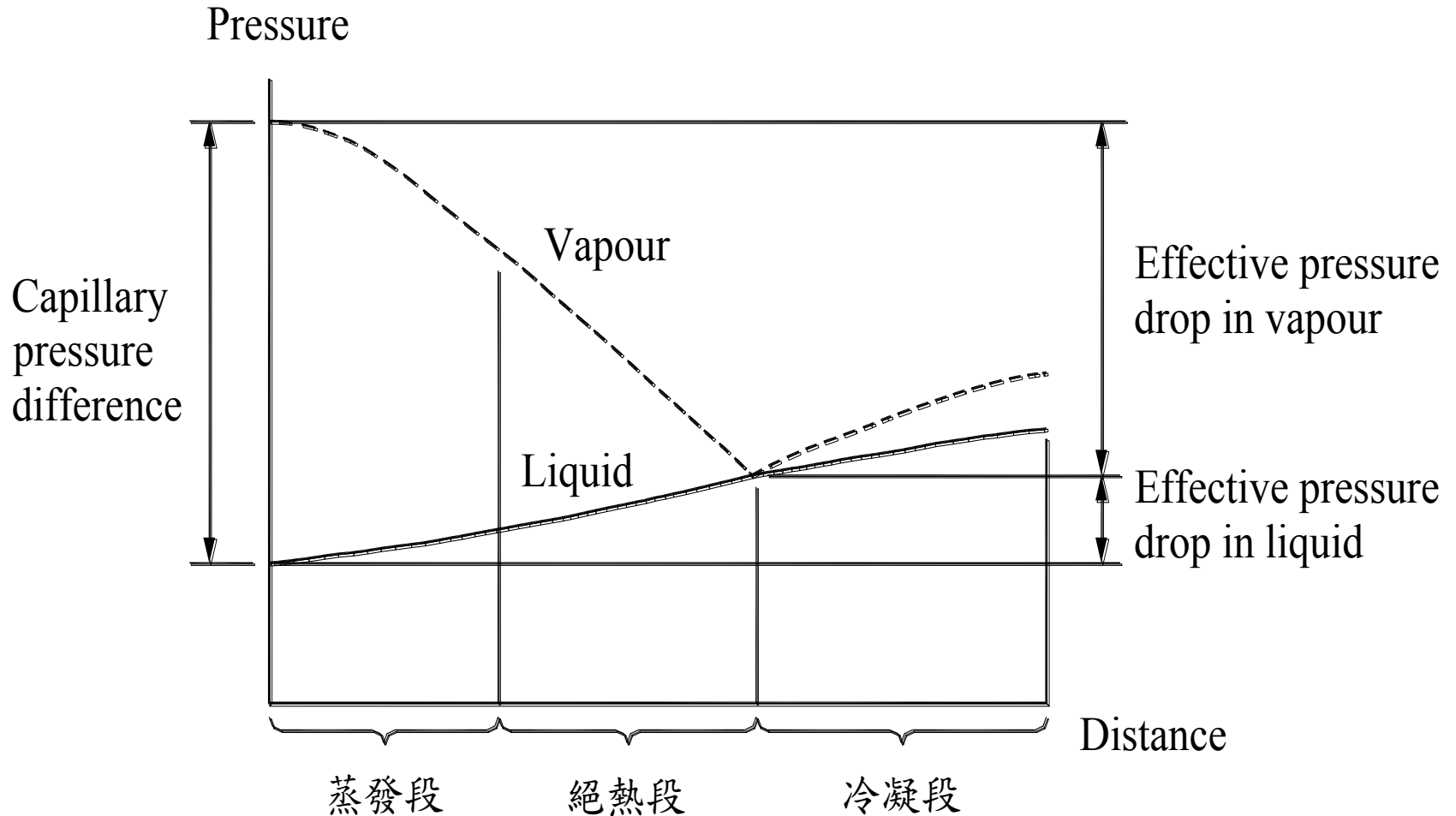
適中氣體速度下，熱管內的氣液的壓力分布





熱管內壓力變化

較高氣體速度下，熱管內的氣液的壓力分布





相容性實驗測試結果

Source: A. Faghri

工作流體	相容材料	不相容材料
水	不鏽鋼、銅、鎳、鈦	鋁
阿摩尼亞	鋁、不銹鋼、鐵、鎳	銅
丙酮	鋁、不銹鋼、銅	
甲醇	不銹鋼、鐵、銅、鎳	鋁
冷媒	鋁	



壓力損失與平衡

$$\Delta P_{cap,max} \geq \Delta P_l + \Delta P_v \pm \Delta P_g$$

ΔP_l : 液相壓損

ΔP_v : 汽相壓損

ΔP_g : 重力壓損

$$\Rightarrow \Delta P_{cap,max} = \frac{2\sigma \cos \theta}{r_p}$$

σ : 表面張力

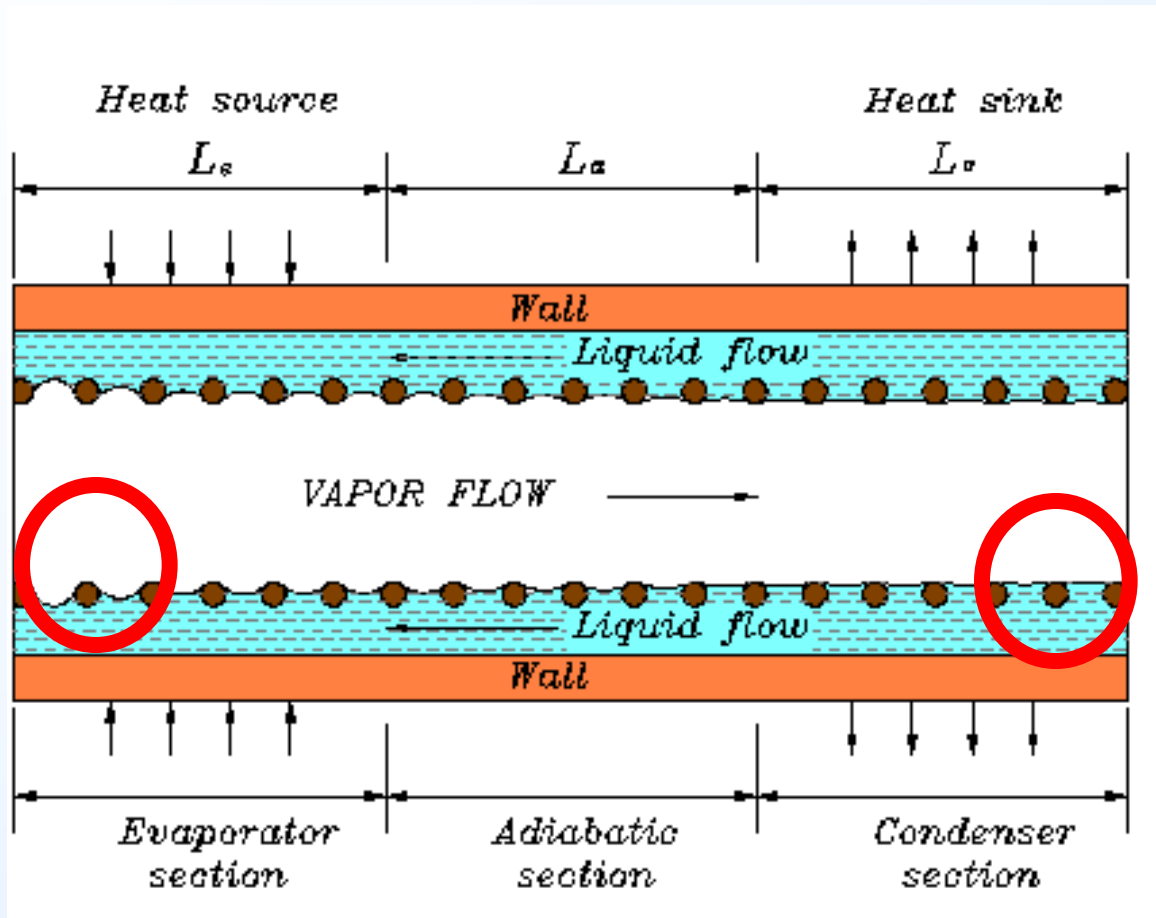
θ : 接觸角

r_p : 毛細孔半
徑





熱管的液汽介面

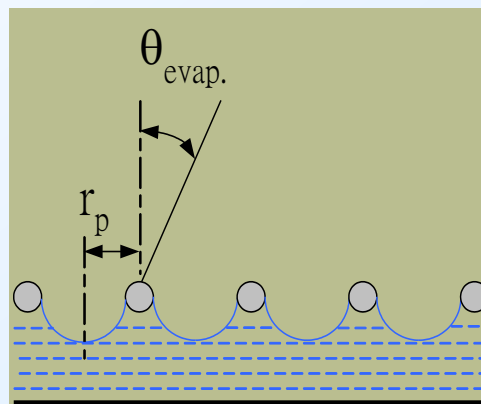




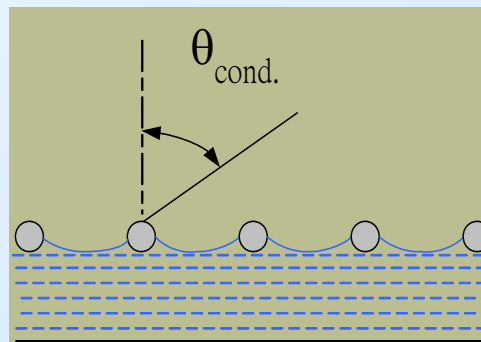
毛細壓力

液汽介面

$$\Delta P_{cap} = 2\sigma \left(\frac{\cos \theta_{evap}}{r_p} - \frac{\cos \theta_{cond}}{r_p} \right) \approx 1 - \approx 0$$
$$= P_G - P_L$$



蒸發區



冷凝區



$$\Delta P_{cap,max} = \frac{2\sigma}{r_p}$$



熱管壓力損失

$$\Delta P_{cap,max} \geq \Delta P_l + \Delta P_v + \Delta P_g$$

Darcy's
Law

$$\Delta P_l = \frac{\mu_l \dot{m}_l L_{eff}}{\rho_l K A_w}$$

$$Q = \dot{m}_l \times h_{fg}$$

L_{eff} ：熱管有效長度= $[L_a + (L_e + L_c)/2]$

重要參數

K：多孔性物質的滲透係數

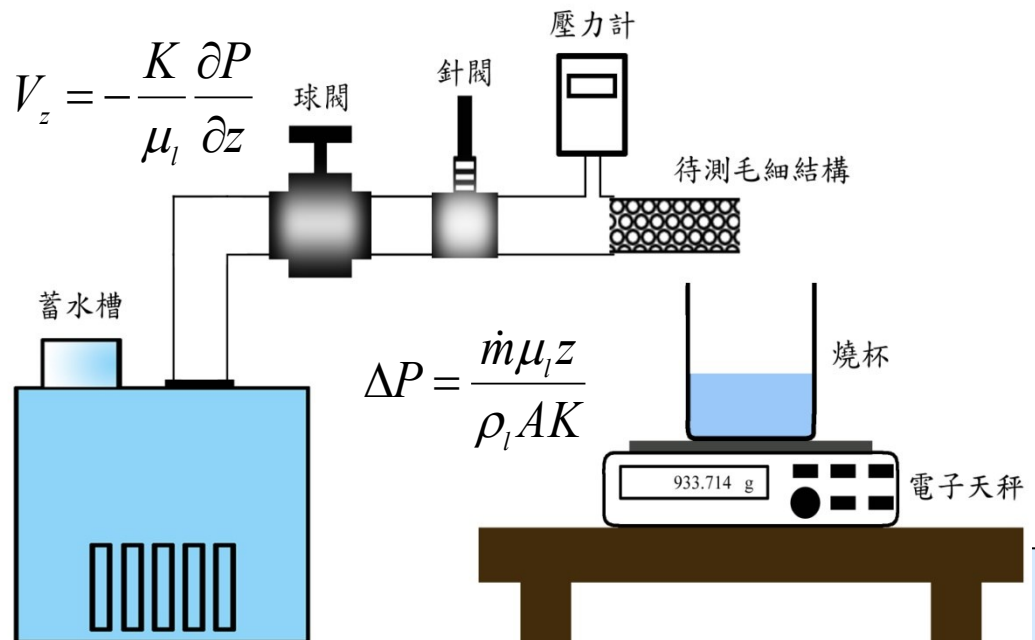
(和孔隙度、孔隙分佈與孔隙幾何尺寸有關)

A_w ：液體流通截面積



滲透度

- 對於液體流動，滲透度是一種毛細阻力的基準，此值越大代表流動阻力愈小，流體在毛細結構裡的流動性愈好，反之，則流動阻力上升。





熱管設計的限制

● 蒸氣壓限制：在熱管內的蒸氣壓通常不高，而沿著冷凝段方向由於摩差的損耗，蒸氣壓會逐步下降，然而氣體與先前介紹的液體不同，液體藉由表面張力的影響可以忍受負壓(即拉力)，但氣體的壓力不可能有負壓的情況，因此即使在冷凝段的盡頭，蒸氣的壓力必定是大於零。

● 音速限制：在低溫與低壓的應用場所中，氣體的相對密度較低而流速也比較高，如果氣體的速度高到等於該位置的音速時，我們稱之此時達到聲速限制，此時並稱之為 **choked**，也就是說無法再增加流速或流量，同樣的音速限制與蒸發段進口的溫度有關。

● 攜帶限制：熱管內氣體與液體的相對流速通常甚快，此一氣體流動的慣性力可能將管蕊上的液體剝離，因此這些脫離管蕊的液體便會造成氣體流向冷凝段的障礙，如果此一濺灑的情持續惡化，將會促使蒸發段出現乾涸的現象(**dry out**)，此時稱之為攜帶限制。



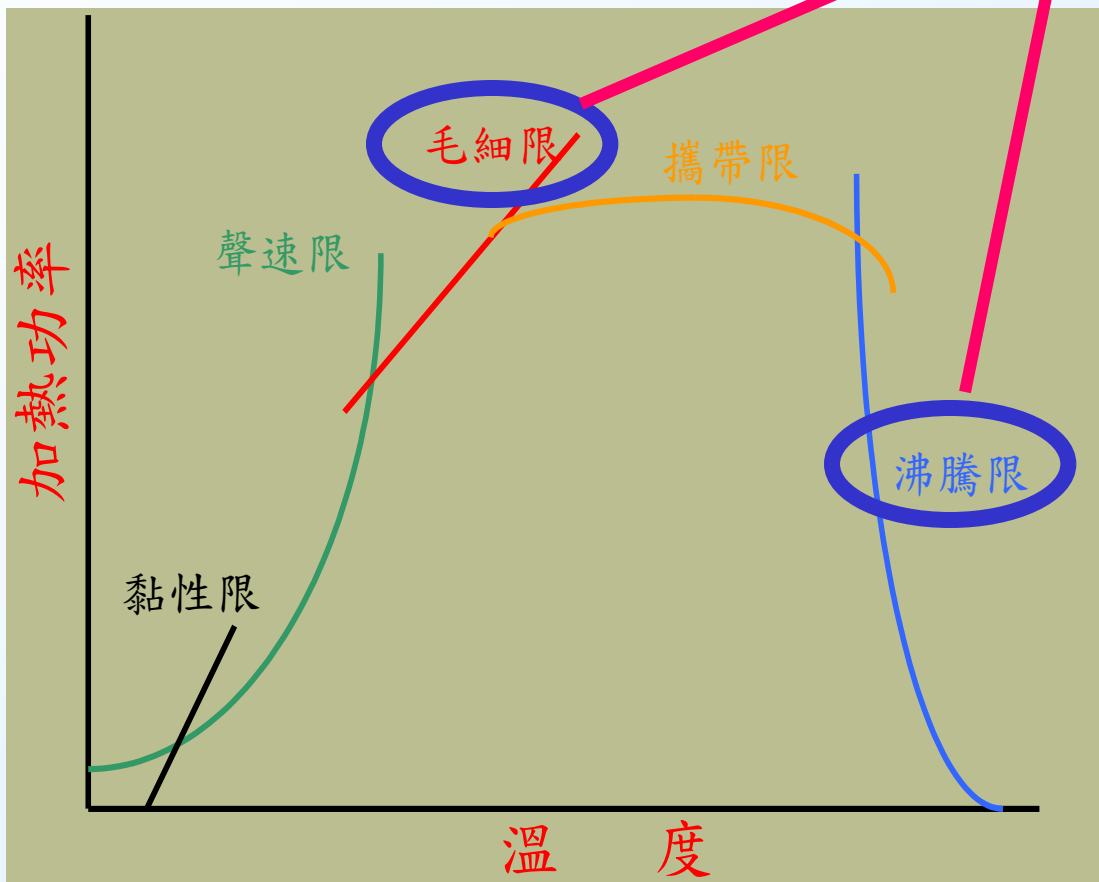
熱管設計的限制 (Conti..)

- 循環限制：熱管中的驅動壓力差與管蕊及工作流體有很大的關聯，一旦建立最大毛細壓力，則熱管內的流量將會維持在一個固定值以下，若持續的增加熱傳量將可能使蒸發段出現乾涸的現象，此時的熱通量稱之為循環限制。
- 沸騰限制：通常在蒸發段內，液體是以蒸發的型態變化成氣體，但是如果熱通量過大，則液體有可能以沸騰的形式轉變成氣體，如果沸騰相當的激烈，則可能造成蒸發段表面出現乾涸的現象，此時稱之為沸騰限制，不過因為沸騰促成的乾涸的現象很難預測，因此習慣上多以蒸發段出現沸騰現象時稱之，不過這裡仍要特別強調，沸騰限制對毛細驅動的影嚮較大，以重力驅動的熱虹吸型熱管，沸騰並不會立即帶來乾涸的嚴重後果(這是因為熱虹吸型熱管所填充的工作流體通常比較多)。



熱管的限制

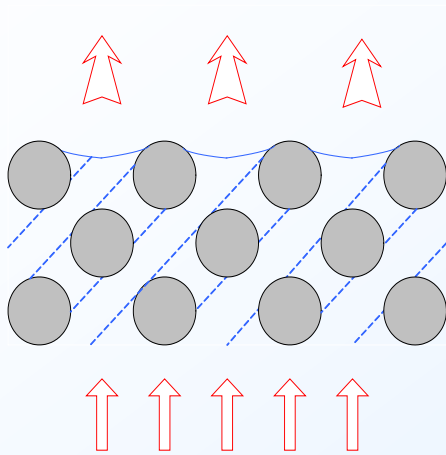
銅水熱管所遇之極限



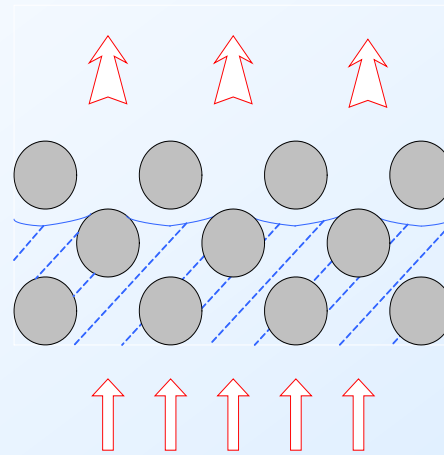


熱管的限制

□ 熱傳導-熱對流

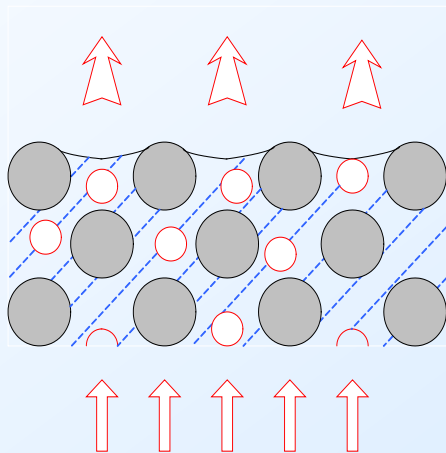


□ 液體退後

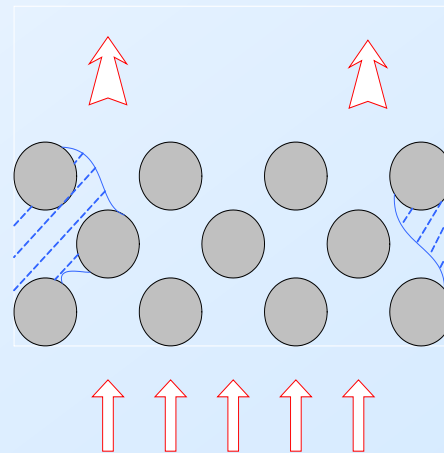


→ Capillary Limit

□ 核沸騰



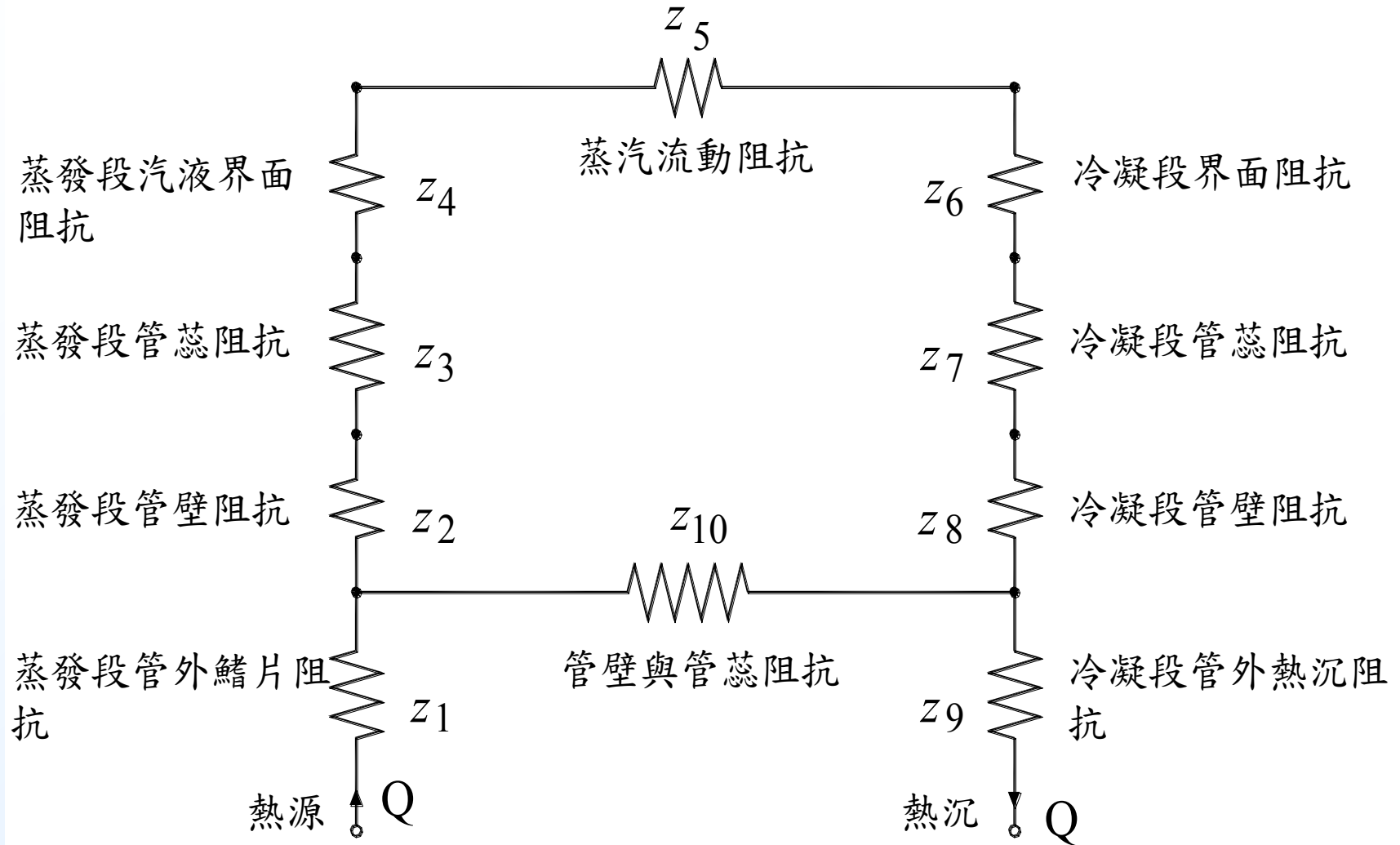
□ 薄膜沸騰



→ Boiling Limit

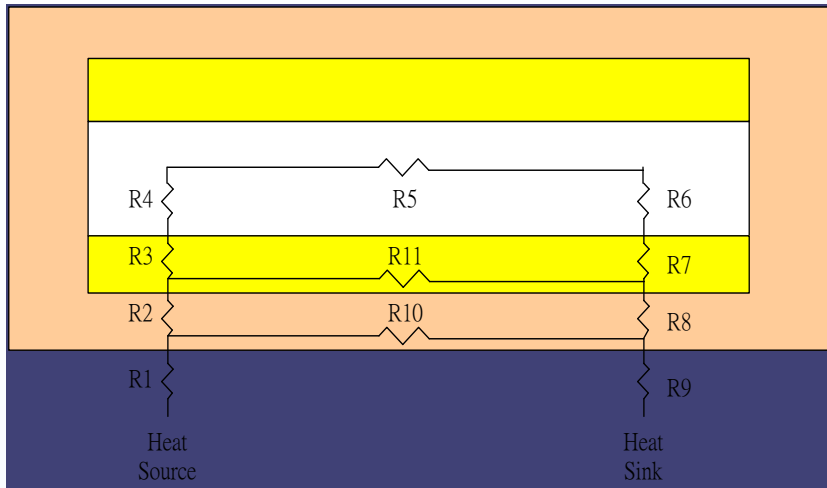


熱管阻抗





熱管的熱阻



$$R_{hp} = R_2 + R_3 + R_4 + R_5 + R_6 + R_7 + R_8$$

	熱阻	大小(°C/W)
R_2	$\ln(D_o/D_i)/2\pi L_e K_{wall}$	10^{-3}
R_3	$\ln(D_i/D_v)/2\pi L_e K_{wick,eff}$	$10^{-2} \sim 10^{-1}$
R_4	$RT_v^2(2\pi RT_v)^{1/2}/h_{fg}^2 P_v A_e$	10^{-7}
R_5	$128\mu_v L_{eff} T_v P_v / \pi \rho_v^2 D_v^4 h_{fg}^2$	10^{-10}
R_6	$RT_v^2(2\pi RT_v)^{1/2}/h_{fg}^2 P_v A_c$	10^{-7}
R_7	$\ln(D_o/D_i)/2\pi L_c K_{wick,eff}$	$10^{-2} \sim 10^{-1}$
R_8	$\ln(D_o/D_i)/2\pi L_c K_{wall}$	10^{-3}

← 數值大小依實際而定

D_o : 外徑

D_i : 內徑

D_v : 蒸汽通道

L_e : 蒸發段長度

L_c : 冷凝段長度

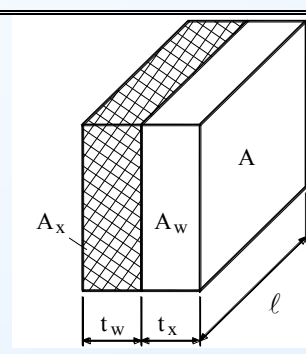
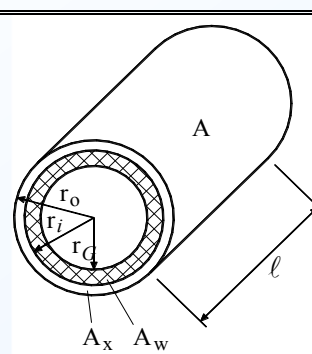
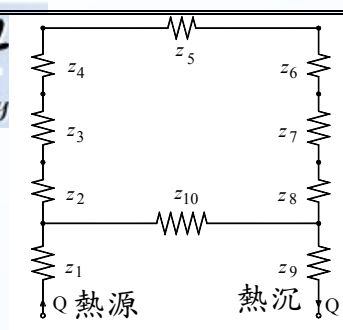


- 在實際應用上，熱管的熱源會置於蒸發段而熱沉會置於冷凝段，因此熱傳的總阻抗為溫度差及熱傳量的比值(如同電阻為電壓與電流的比值)，即：

$$z = \frac{\Delta T}{Q}$$

- 設計經驗上顯示合理的設計應滿足下列的條件(ESDU, 1979)：

$$\frac{z_{10}}{z_2 + z_3 + z_7 + z_8} > 20 \quad (20-6)$$



z_1 , 蒸發段管外鰭片阻抗

$$1/h_e A_e \text{ (如果有鰭片時為 } 1/\eta_o c h_e A_e \text{)}$$

z_2 , 蒸發段管壁阻抗

$$\frac{\log_e (r_o/r_i)}{2\pi \ell} \quad t_x/k_x A_e$$

z_3 , 蒸發段管蕊阻抗

$$\frac{\log_e (r_i/r_G)}{2\pi \ell} \quad t_w/k_w A_e$$

z_4 , 蒸發段汽液界面阻抗

$$\frac{\sqrt{RT_{eff}^3/2\pi}}{i_{LG}^2 \ell} \quad \frac{\sqrt{2\pi RT_{eff}^3}}{i_{LG}^2 A_e \rho_G}$$

z_5 , 蒸氣流動阻抗

$$\frac{\log_e (r_i/r_G)}{i_{LG} Q \rho_G}$$

z_6 , 冷凝段界面阻抗

$$\frac{\sqrt{RT_{eff}^3/2\pi}}{i_{LG}^2 \ell} \quad \frac{\sqrt{2\pi RT_{eff}^3}}{i_{LG}^2 A_G \rho_G}$$

z_7 , 冷凝段管蕊阻抗

$$\frac{\log_e (r_i/r_G)}{2\pi \ell} \quad t_w/k_w A_c$$

z_8 , 冷凝段管壁阻抗

$$\frac{\log_e (r_o/r_i)}{2\pi \ell} \quad t_x/k_x A_c$$

z_9 , 冷凝段管外熱沉阻抗

$$1/h_c A_c \text{ (如果有鰭片時為 } 1/\eta_o c h_c A_c \text{)}$$

z_{10} , 管壁與管蕊阻抗

$$\frac{\ell \quad \ell \quad \ell}{A_x k_x + A_w k_E}$$



毛細結構的參數

毛細結構重要參數

- 有效毛細半徑
- 孔隙度
- 等效熱傳導係數
- 滲透性



毛細結構特性比較

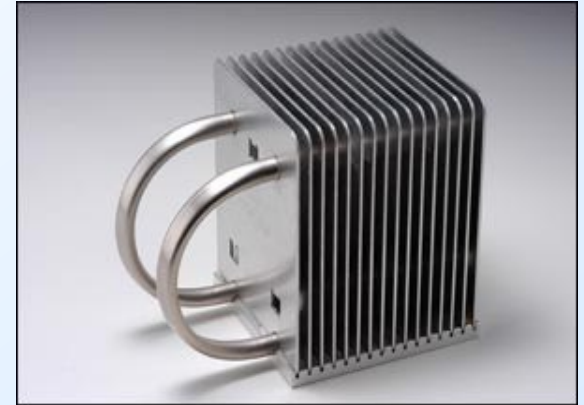
熱管特性	金屬網	粉末燒結	溝槽
毛細力	高	高	低
滲透性	低-中	低-中	中-高
熱阻	中-高	中	低
毛細限(水平時)	中	中	高
重力對毛細限之影響	中-高	低	高
沸騰限	中	高	中
可靠度	低-中	高	高

❖此表僅是表示一般之比較，而非絕對之結果。

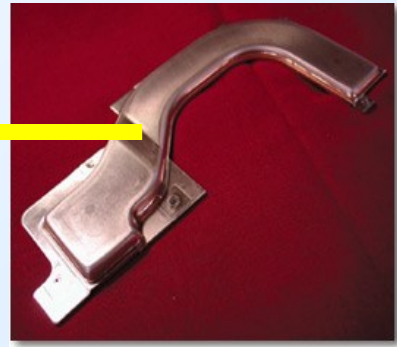
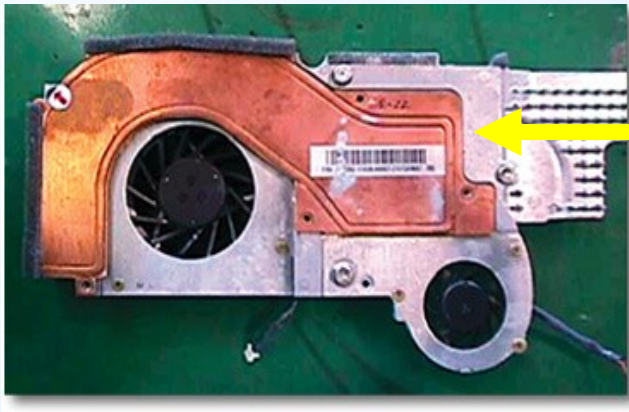


散熱器

Heat Pipe



Vapor Chamber



Thermal Column



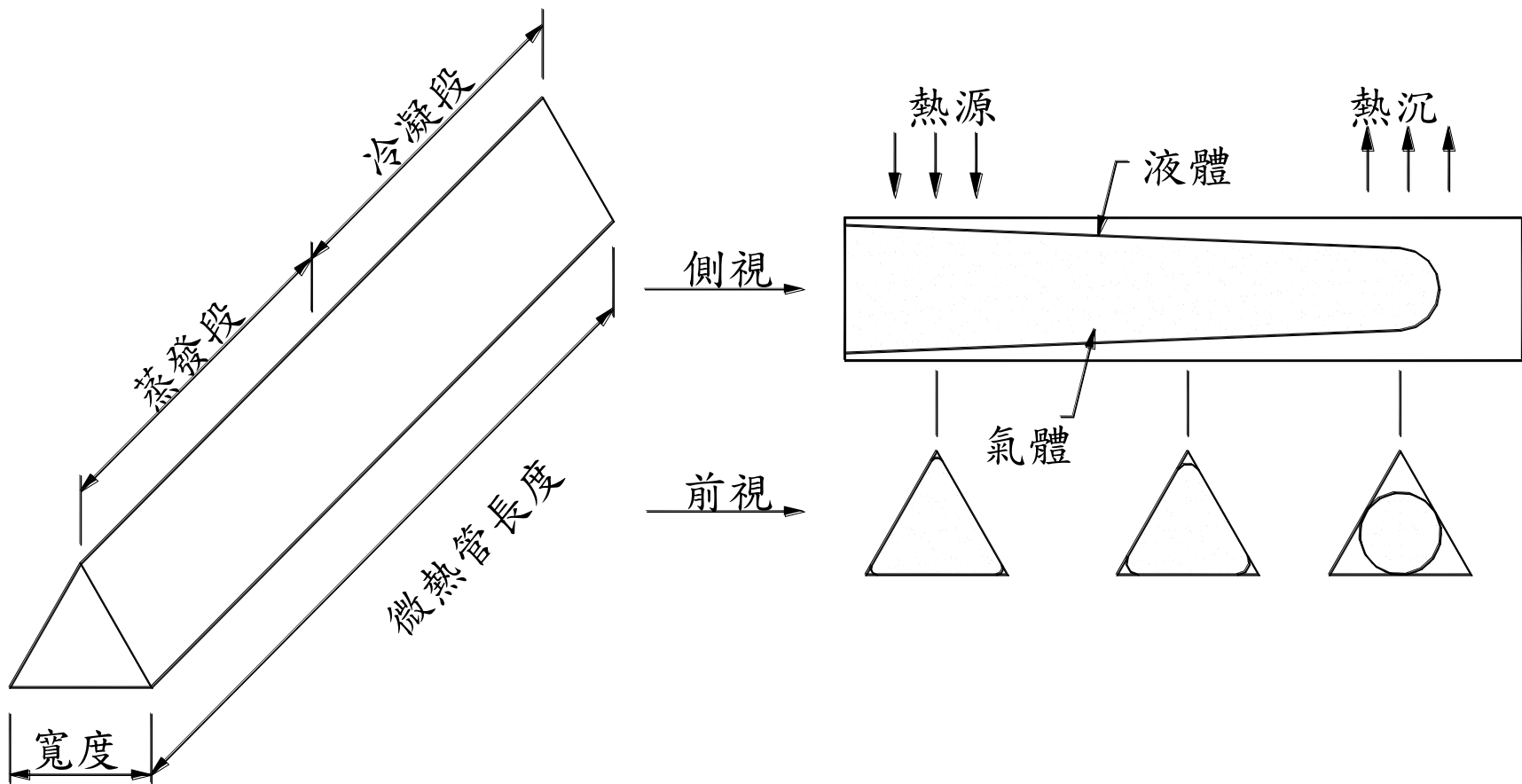


常見之熱回收用氣
對氣熱管熱交換器





特殊熱管 — 微熱管





特殊熱管 — 迴路熱管

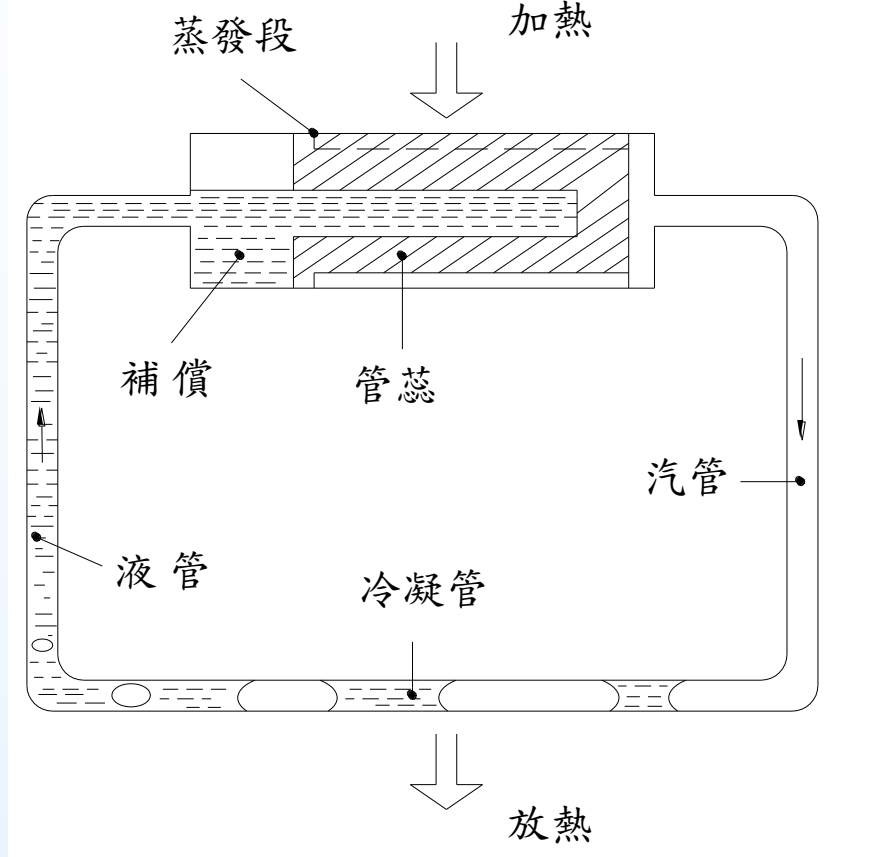


圖20-37 迴路式熱管

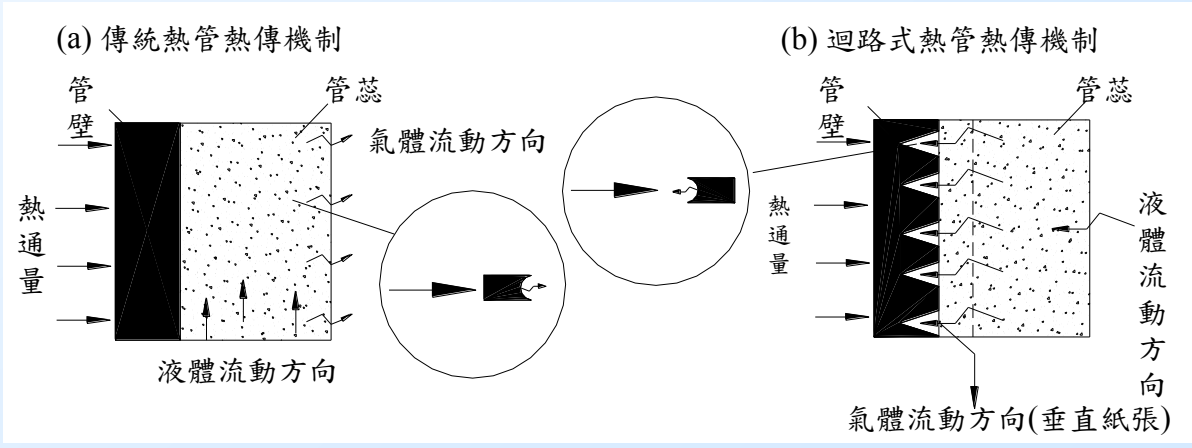


圖20-38 迴路式熱管與傳統熱管熱傳機制之差異



特殊熱管 — 震盪熱管

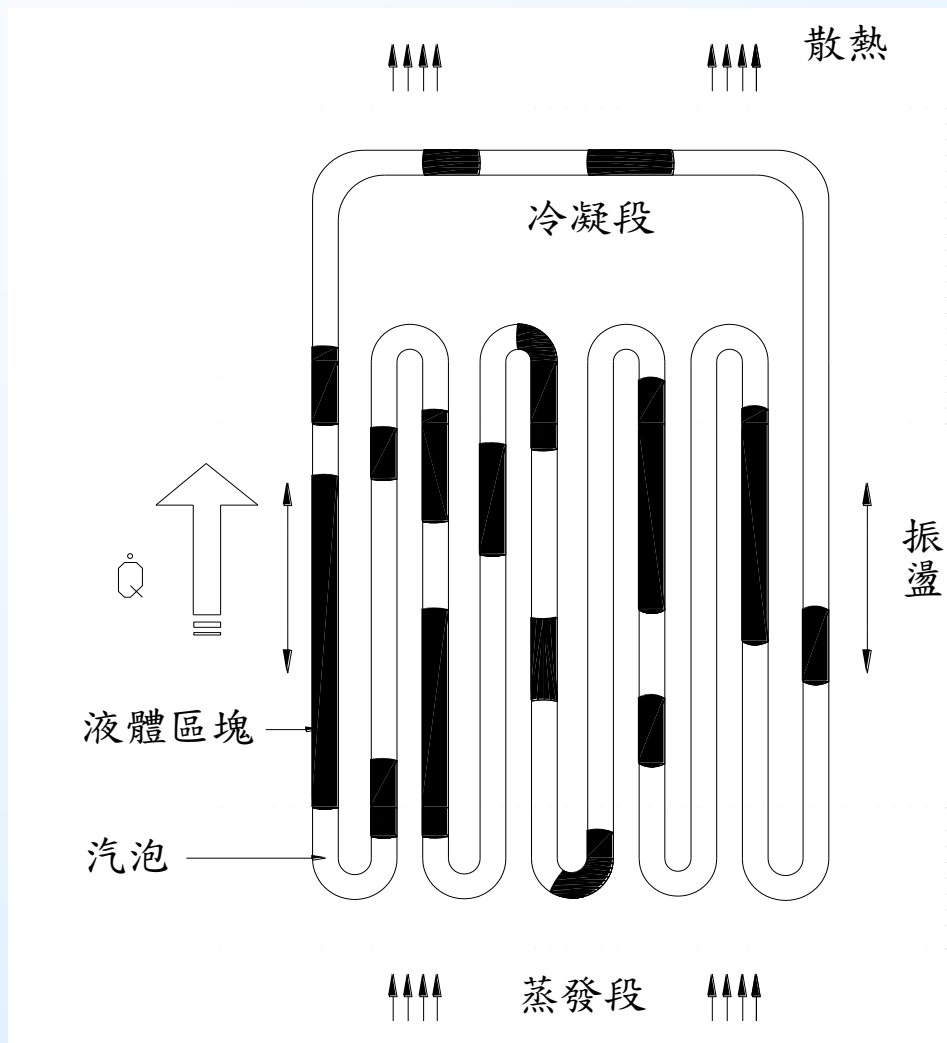


圖20-40 震盪式熱管



特殊熱管 — 蒸氣腔室(平板式熱管, Vapor Chamber)

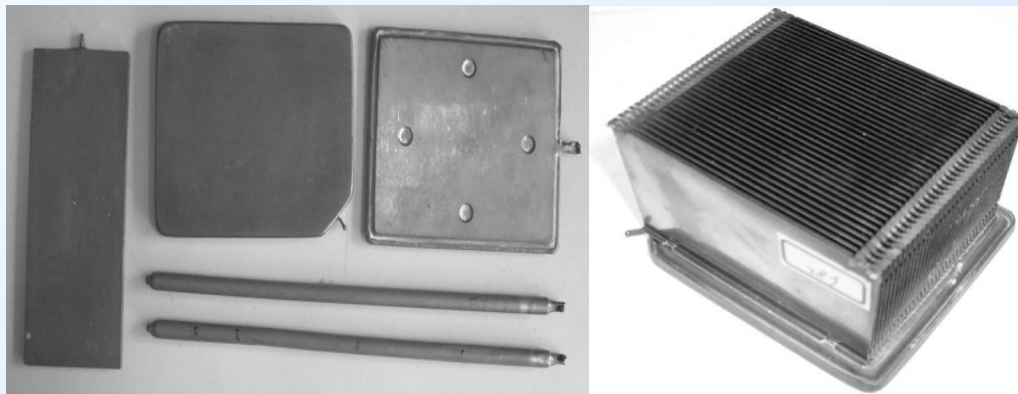
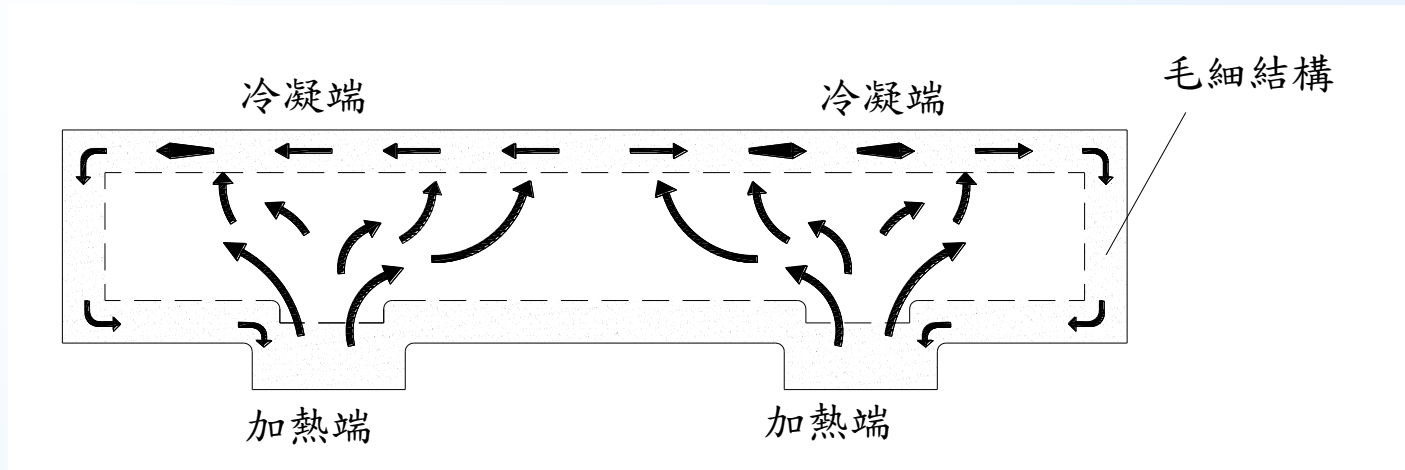


圖20-41 (a) 蒸氣腔室工作原理 (b) 熱管與蒸氣腔室, (c)與熱沉整合的產品



結 論

- 熱管熱阻基本取決於蒸發區及冷凝區之管壁與毛細結構厚度。
- 熱管熱阻與長度沒有直接之關係。
- 毛細結構厚度增加有助於最大熱傳量的增加，但也造成熱阻值增加。
另外也會影響到蒸汽的流動。
- 孔隙度增加，液體回流阻力減少，熱傳量增加；但熱阻也增加。
- 熱管若有後續之成形加工，參數必須以最終形狀做最佳化設計，
以符合性能之要求。
- 熱管類產品能有效降低散熱模組之熱阻，但其基本熱阻是存在的。



結 論

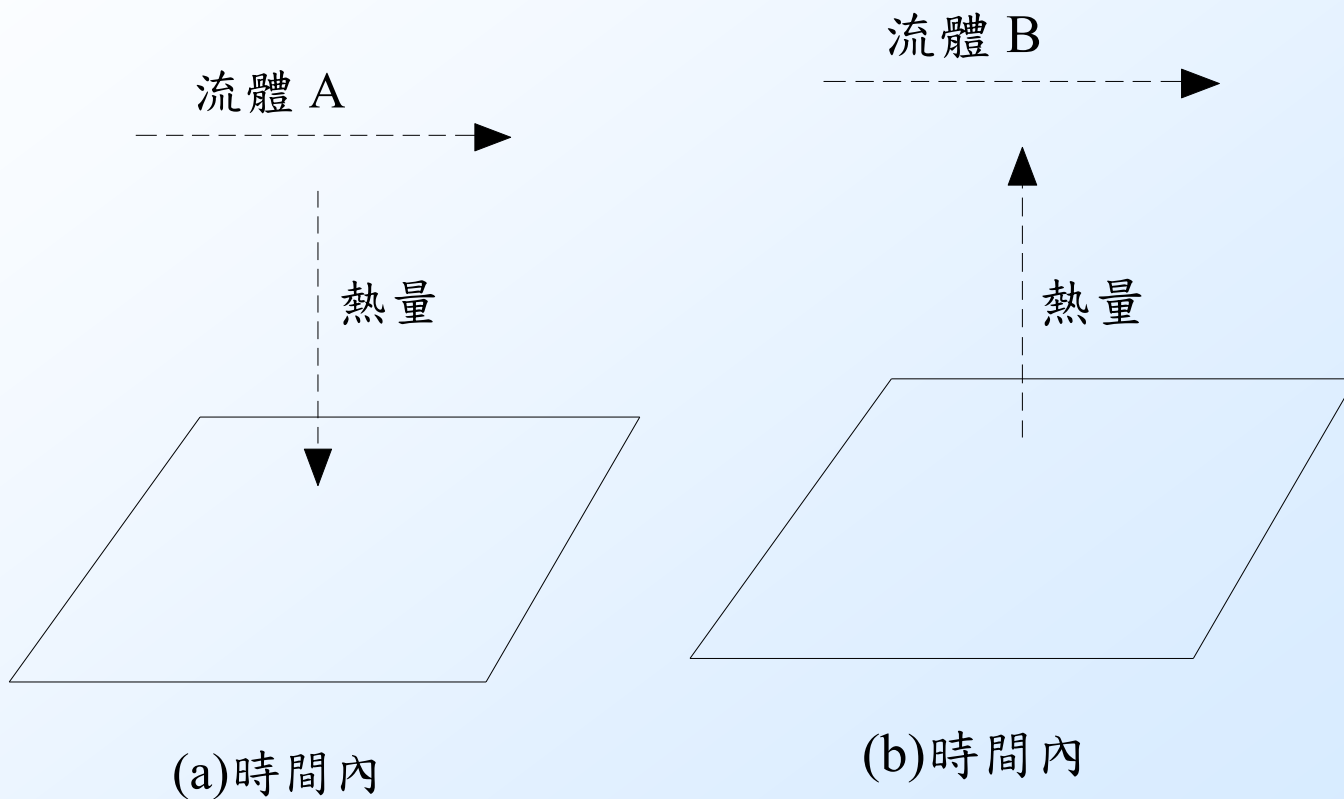
- 熱管扮演傳熱的重要組件，而非散熱之用。
- 熱管製作時，必須確認製程參數，是否能滿足設計之要求。
- 必須瞭解熱阻最大之地方，方能判斷熱管使用之有效性。
- 熱管應用有其極限，並非萬靈丹。



再生式熱交換器



再生式熱交換器工作原理





主要優缺點

優點

- 高密度，其密集度可高達 $2000\sim 15000\text{ (m}^2/\text{m}^3)$ ，典型殼管式熱交換器僅 $500\text{ m}^2/\text{m}^3$ 。由於密集度甚高，相對的熱效率也甚高，例如空氣分離與液化的應用上，熱效率常超過99%。
- 可以使用相當便宜的填料。
- 熱側與冷側以週期性逆向交替流動的方式通過熱交換器，因此熱交換器具備「自行清理」的特性。
- 接頭(header)設計比較簡單。

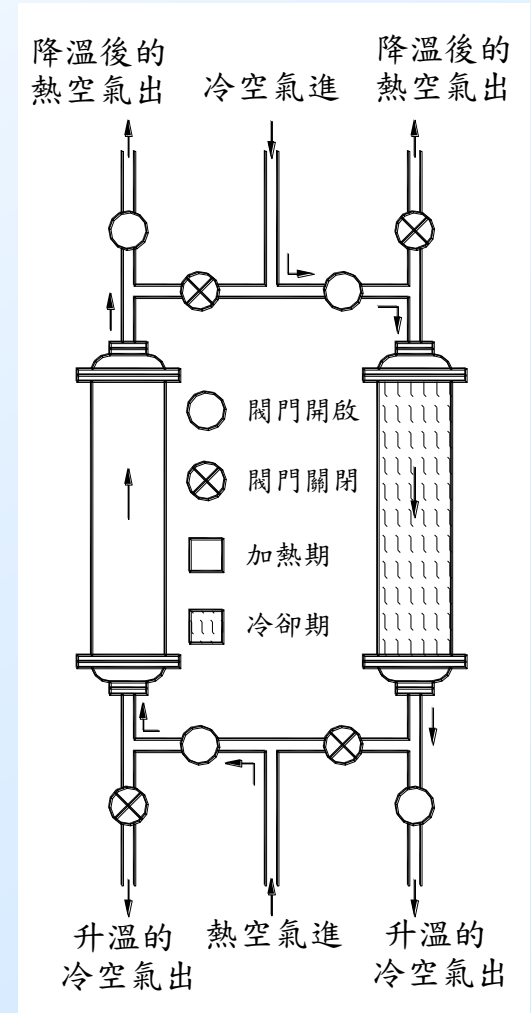
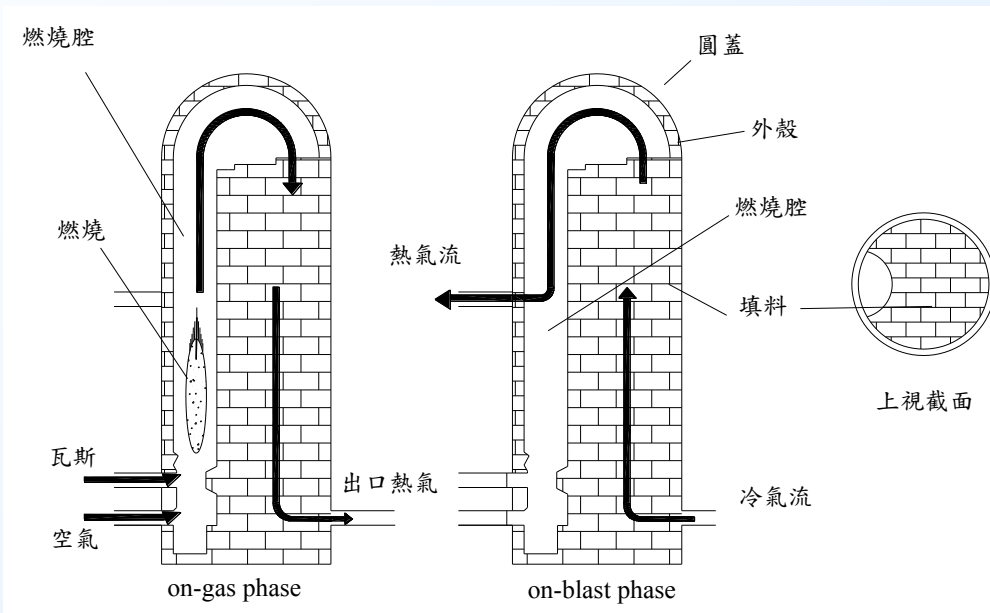
缺點：

- 熱側與冷側可能有相當大的壓差(4~7 bar)，需要使用密封的設計，以減少洩漏的問題。
- 相對於回覆式熱交換器，其流動方向來回的變化較為複雜，因此流體的流動損失也比較大。
- 由於壓降的限制，通常熱交換器的正向面積會比較龐大。
- 流體流動中的夾帶物(carryover)會增加額外的流動損失，同時亦可能造成熱側與冷側流體的互相混合，促使污染的發生。
- 熱效率高， $\varepsilon \approx 1$ ，因此熱交換器的熱容相當的大，通常設計上遠比工作流體的熱容為大，因此再生式熱交換器僅適用於氣體的應用。
- 轉輪式的再生式熱交換器需要而外的支撐設計與驅動裝置。



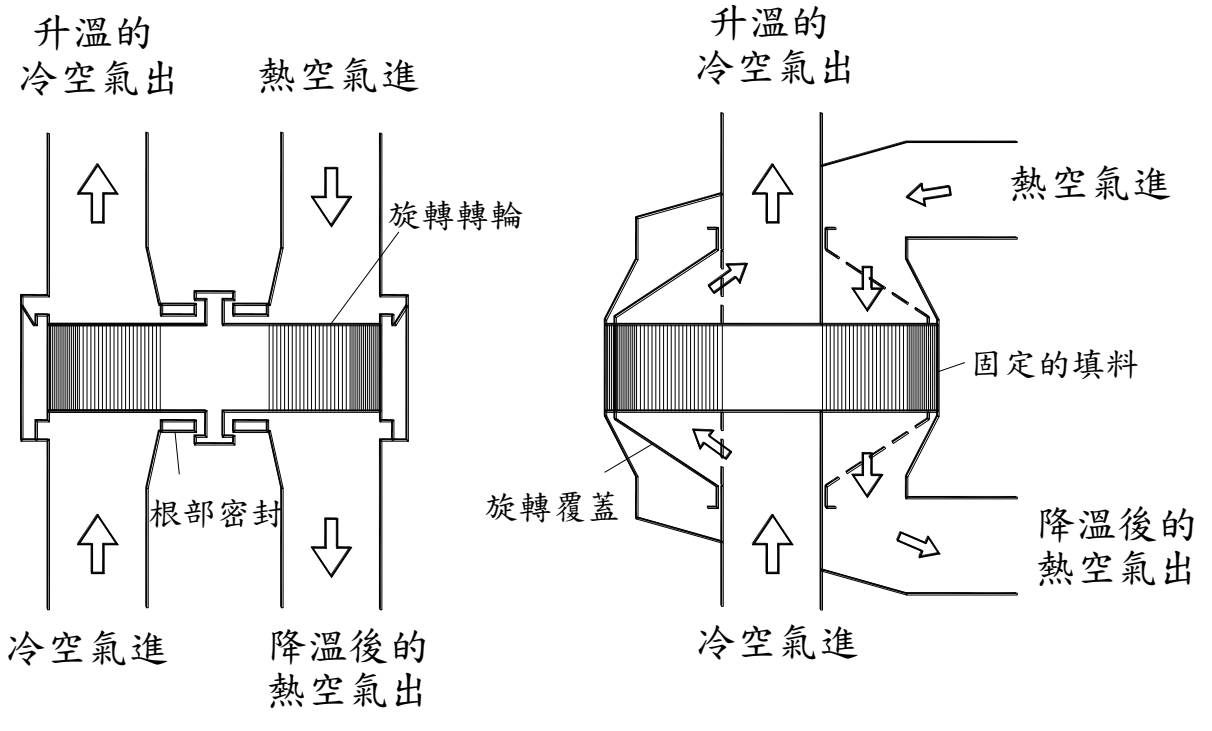
再生式熱交換器的分類

● 固定床型式(Fixed bed) 單一與多只設計





轉輪型式再生式交換器



(a) Rothemule 型式與(b) Ljungstrom 型式轉輪式再生式交換器



再生式熱交換器的填料選擇考量

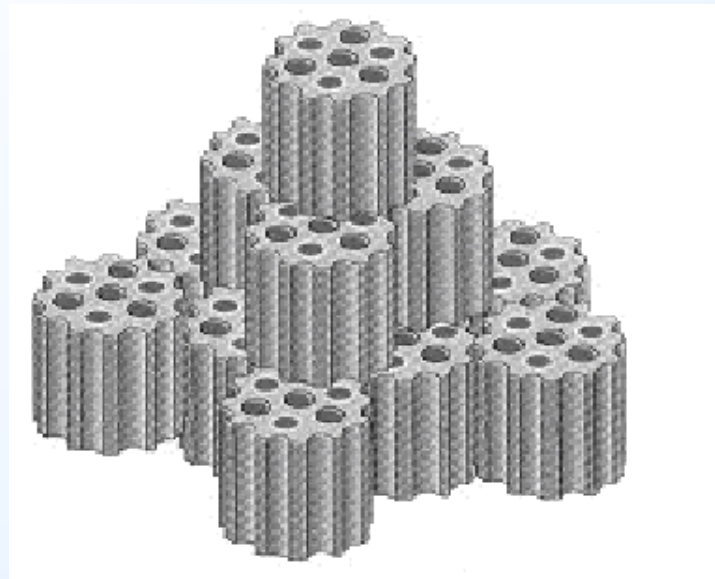
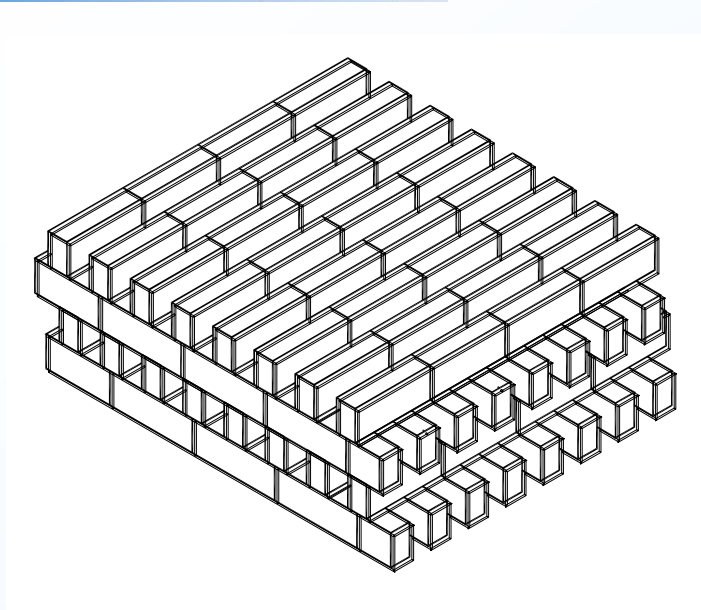
● 溫度操作的範圍

● 工作氣體內所特別含有的腐蝕物質

● 實際應用上可以忍受的壓降

- (1) 高比熱。
- (2) 高密度。
- (3) 高熱擴散係數。
- (4) 允許可逆加熱與冷卻。
- (5) 化學與幾何的穩定性。
- (6) 不可燃燒，不腐蝕，且無毒性。
- (7) 低蒸氣壓以避免污染。
- (8) 造價便宜。
- (9) 足夠的機械強度。
- (10) 合適的操作溫度。
- (11) 可長期操作。

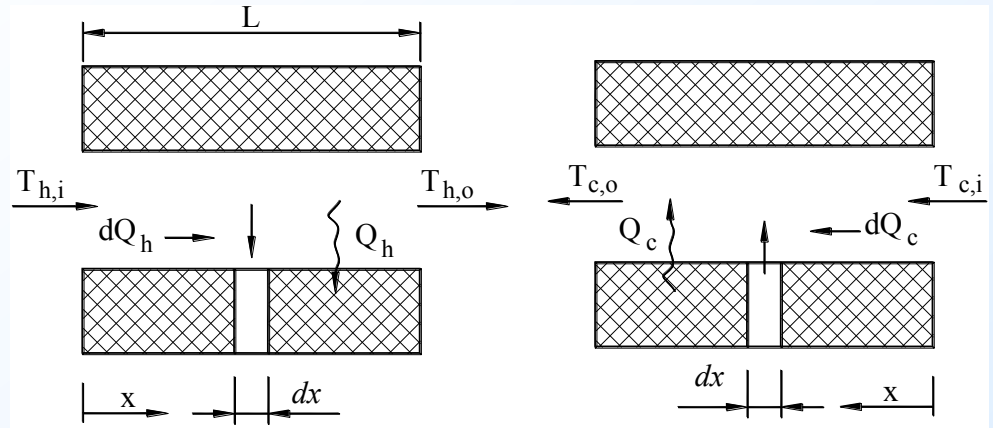
填料大致可分為三類，即(A)非金屬類；(B)金屬類；(C) PCM (phase change materials)



(a) basket weave 型式與(b) hot blast stove 填料的offset的安排型式



熱流分析



- 填料的熱傳導係數在沿流動方向甚小(≈ 0)而與流動方向垂直時則甚大($\approx \infty$)。
- 填料的比熱為定值。
- 熱側與冷側流體在冷熱週期切換時無混合現象。
- 流體與填料間的對流熱傳係數為定值(熱側與冷側可以不同)。
- 流動為逆向流動。
- 進入熱交換器的入口溫度為均勻分佈且不隨時間變動。
- 冷熱切換週期固定，且熱損失甚小可忽略。
- 流體洩漏的效應甚小可忽略。
- 熱流特性可以一維尺度來描述(1-dimensional)。



$$\begin{aligned}
 dQ &= h_h A (T_h - T_m) = m_h c_{p,h} \frac{\partial T_h}{\partial t} + m_h c_{p,h} u \frac{\partial T_h}{\partial x} \\
 &= m_h c_{p,h} \frac{\partial T_h}{\partial t} + m_h c_{p,h} \frac{L}{t} \frac{\partial T_h}{\partial x} = \frac{m_h}{t} t c_{p,h} \frac{\partial T_h}{\partial t} + \frac{m_h}{t} c_{p,h} L \frac{\partial T_h}{\partial x} \\
 &= i \left(u \frac{\partial T}{\partial t} + \frac{\partial T}{\partial x} \right) \quad \left(1 \frac{\partial T}{\partial t} + \frac{\partial T_h}{\partial x} \right)
 \end{aligned} \tag{18-3}$$

18-1與18-3為再生式熱交換器的基本設計方程式

對再生式熱交換器而言，其最重要的設計參數為熱效率比 (thermal ratio) η_{reg} ，此參數與第二章介紹的有效度 ϵ 意義很類似，熱效率比定義如下：

$$\eta_{reg,c} = \frac{T_{c,out} - T_{c,in}}{T_{h,in} - T_{c,in}} \tag{18-4}$$

同樣的，熱側的熱效率比 (thermal ratio) 定義如下：

$$\eta_{reg,h} = \frac{T_{h,in} - T_{c,out}}{T_{h,in} - T_{c,in}} \tag{18-5}$$



對逆流再生式熱交換器而言，可以證明 η_{reg} 與 Λ 、 Π 有關(就如同 ε -NTU 法中的 ε 與 NTU 、 C^* 有關)；其中 Λ 為「縮減長度」(reduced length)，與 ε -NTU 法中的 NTU 類似，代表再生式熱交換器的熱尺寸(thermal size)，定義如下：

$$\Lambda = \frac{hA}{i_r} \quad (18-6)$$

而 Π 為「縮減週期」(reduced period)，定義如下：

$$\Pi \equiv \frac{hA \left(P - \frac{L}{u} \right)}{M_m c_{p,m}} \quad (18-7)$$

其中 P 為週期， L/u 代表前次週期中的流體滯流於填料內的時間，對一般的應用而言， $P \gg L/u$ ，所以 $\Pi \approx \frac{hAP}{M_m c_{p,m}}$



熱傳分析

【加熱週期熱傳量】 = 【儲存在再生式熱交換器中的熱傳量】，所以：

$$dQ = M_m c_{p,m} \frac{\partial T_{m,h}}{\partial t} \quad (18-1)$$

其中下標 m 代表填料而 h 代表加熱週期，同時【加熱週期熱傳量】 = 【熱側流體的顯熱變化】 + 【儲存在通過熱側流體的熱傳量】，所以：

$$dQ = h_h A (T_h - T_m) = \rho_h c_{p,h} V \frac{\partial T_h}{\partial t} + \rho_h c_{p,h} V u \frac{\partial T_h}{\partial x} = m_h c_{p,h} \frac{\partial T_h}{\partial t} + \dot{m}_h c_{p,h} \frac{\partial T_h}{\partial x} \quad (18-2)$$

式18-2中右式的第一項代表流體的儲存熱量的效應，而第二項代表流體被加熱的顯熱效應



- 如果再生式熱交換器的 $\Lambda_c = \Lambda_h$ 且 $\Pi_c = \Pi_h$ ，則稱之為「對稱」(symmetric)，此時熱側與冷側的熱效率比相同，即 $\eta_{reg,c} = \eta_{reg,h}$

如果考慮加熱週期與冷卻週期間的能量平衡，可得：

$$\dot{m}_h c_{p,h} (T_{h,in} - T_{h,out}) = \dot{m}_c c_{p,c} (T_{c,out} - T_{c,in}) \quad (18-8)$$

上式兩側同時除上 $T_{h,in} - T_{c,in}$ 後，可得到下式：

$$\dot{m}_h c_{p,h} \eta_{reg,h} = \dot{m}_c c_{p,c} \eta_{reg,c} \quad (18-9)$$

若式18-9中的 $\dot{m}_h c_{p,h} P_h = \dot{m}_c c_{p,c} P_c$ ，則稱此一再生式熱交換器為「平衡」(balanced)，如同對稱式的再生式熱交換器，此時熱側與冷側的熱效率比相同，即 $\eta_{reg,c} = \eta_{reg,h}$ 。



換句話說，如果一再生式熱交換為平衡($\eta_{reg,c} = \eta_{reg,h}$)，則

$$\rightarrow \frac{\Pi_h}{\Pi_c} = \frac{\Lambda_h}{\Lambda_c} = k \quad (18-11)$$

其中 k 稱之為平衡比(balance ratio)；當然，如果 $k = 1$ 就代表是對稱；式18-10可以適度改寫與修正成下式：

$$\frac{\Pi_h}{\Lambda_h} \times \frac{\Lambda_c}{\Pi_c} = \gamma \quad (18-12)$$

同樣的，如果 $\gamma = 1$ ，則代表再生式熱交換器為平衡；若 $\gamma \neq 1$ ，此一再生式熱交換器則稱之為非平衡式(unbalanced)。



(1) $\Pi \rightarrow 0$;

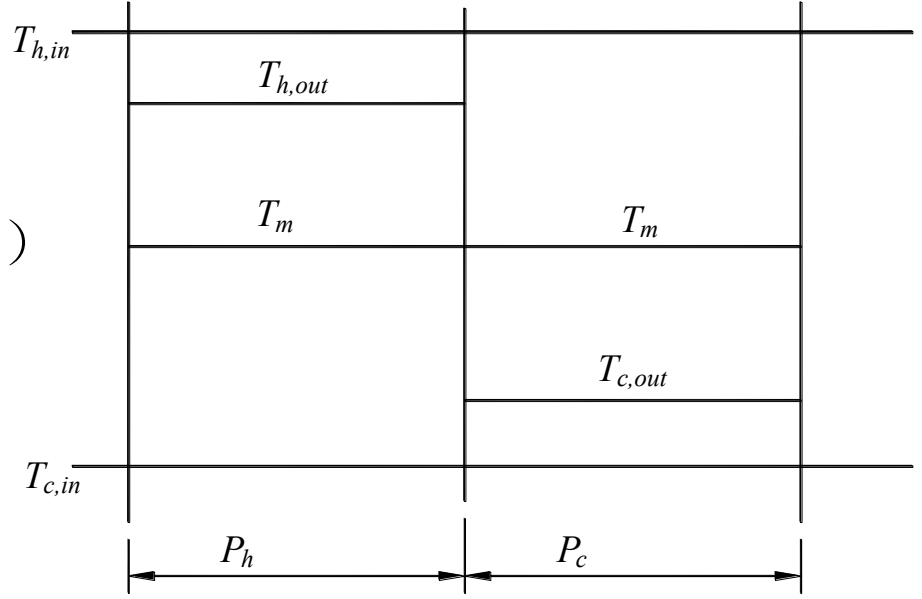
由式18-7可知填料的質量 M_m 很大或週期 P 甚短，因此填料的溫度變化也

就很小

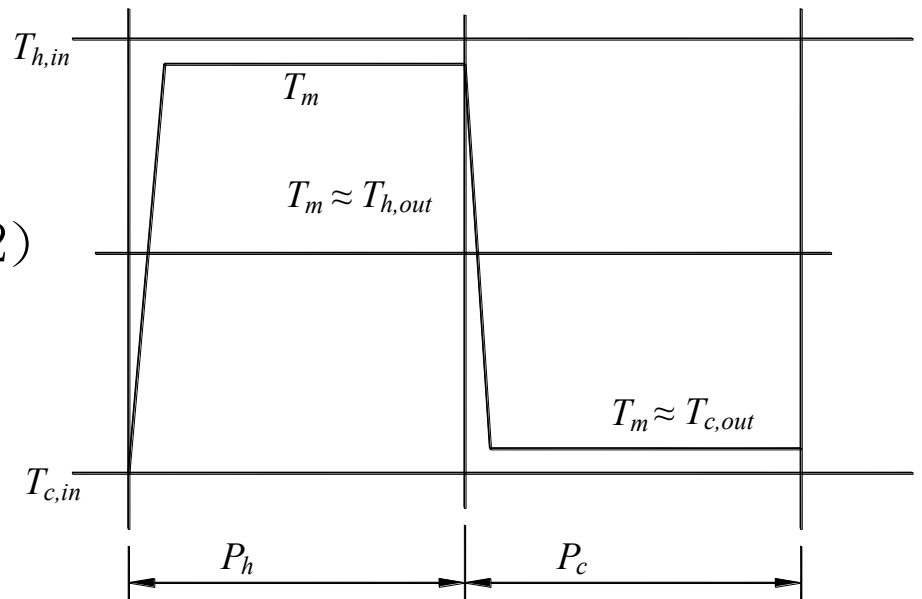
(2) $\Pi \rightarrow \infty$

即週期 P 非常大或填料質量 M_m 很小，因此填料的溫度 T_m 在加熱週期時會非常接近熱側流體的出口溫度

案例(1)

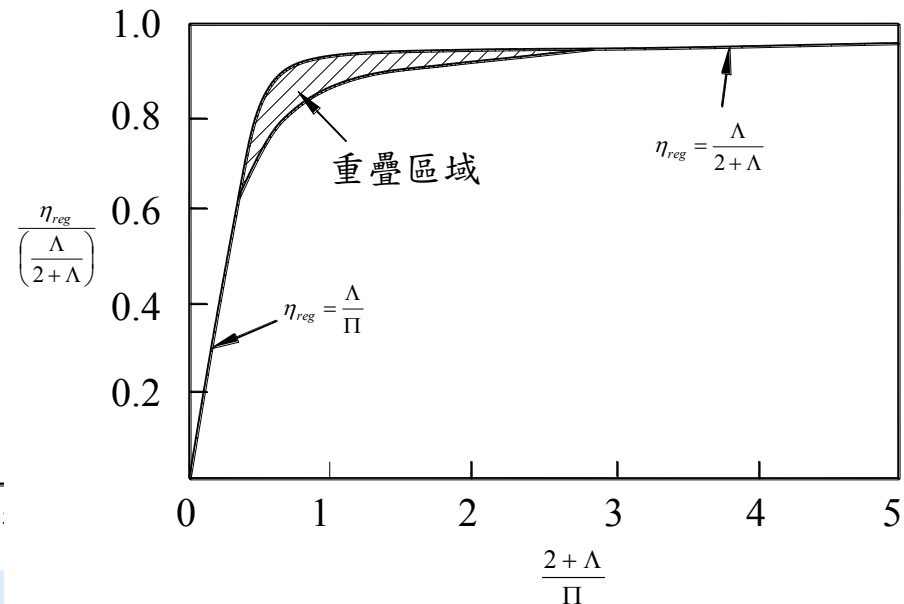
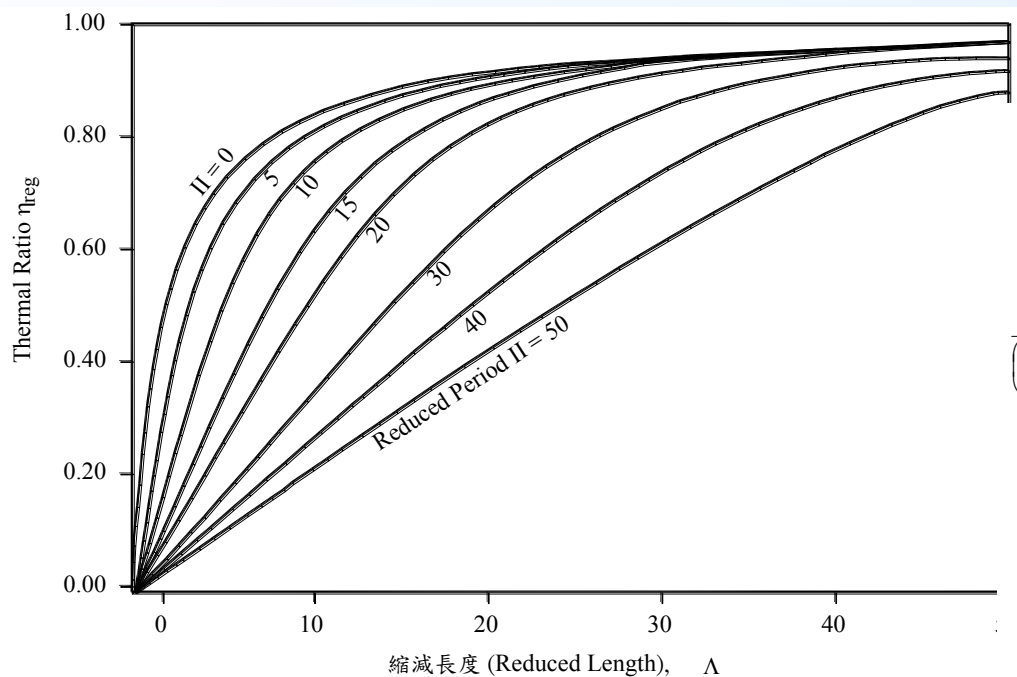


案例(2)





若僅僅先考慮對稱式型式的再生式交換器，即 $\Lambda_c = \Lambda_h = \Lambda$ 且 $\Pi_c = \Pi_h = \Pi$ ，根據Hausen (1983)的研究，此一對稱型再生式熱交換器的熱效率比與 Λ 及 Π 之間的關聯可由圖18-10的結果來表示，同時，圖18-10的結果，可經由另一階段的無因次的新座標轉換處理後，得到如圖18-11所示的類似「相似圖」(similarity)，此圖在使用上更為簡易





計算流程（平衡型）

- (1) 首先算出非對稱再生式熱交換器的 Λ_c 、 Λ_h 、 Π_c 、 Π_h 。
- (2) 計算平衡式再生式熱交換器的縮減長度與縮減週期的和諧平均值 Λ_H 、 Π_H (Harmonic means)：

$$\frac{2}{\Lambda_H} = \frac{1}{\Pi_H} \left(\frac{\Pi_c}{\Lambda_c} + \frac{\Pi_h}{\Lambda_h} \right) \quad (18-24)$$

$$\frac{2}{\Pi_H} = \frac{1}{\Pi_c} + \frac{1}{\Pi_h} \quad (18-25)$$

- (3) 利用計算的 Λ_H 、 Π_H 值，再參考圖18-10或圖18-11找出其對應的熱效率比。



計算流程（非平衡型）

非平衡型的再生式熱交換器的熱效率比值，再經過若干的處理，也可以利用平衡型的計算流程來計算，詳細的非平衡式運算流程說明如下：

(1) 首先算出非平衡再生式熱交換器的 Λ_c 、 Λ_h 、 Π_c 、 Π_h 。

(2) 由式18-11，計算 $\gamma = \frac{\Pi_h}{\Lambda_h} \times \frac{\Lambda_c}{\Pi_c}$ 。

(3) 由式18-25，計算 Π_H 。

(4) 由式18-24，計算 Λ_H 。

(5) 利用計算的 Λ_H 、 Π_H 值，再參考圖18-11或圖18-12找出其對應的平衡熱效率比 $\eta_{reg,H}$ 。

(6) 計算 F ，即
$$F = \frac{\eta_{reg,H}(1-\gamma^2)}{2\gamma(1-\eta_{reg,H})} \quad (18-26)$$

(7) 由下式計算熱週期的熱效率比 $\eta_{reg,h}$ ，即：

$$\eta_{reg,h} = \frac{1-e^F}{\gamma-e^F} \quad (18-27)$$

(8) 由下式計算冷週期的熱效率比 $\eta_{reg,c}$ ，即：

$$\eta_{reg,c} = \gamma\eta_{reg,h} \quad (18-28)$$



案例計算

● 根據下面三種(對稱、平衡與非平衡)型式的再生式熱交換器的資料，計算其相對的熱效率。

(a) $\Lambda_c = \Lambda_h = \Lambda = 10$, $\Pi_c = \Pi_h = \Pi = 5$

(b) $\Lambda_c = 20$, $\Lambda_h = 10$, $\Pi_c = 10$, $\Pi_h = 5$

(c) $\Lambda_c = 10$, $\Lambda_h = 10$, $\Pi_c = 5$, $\Pi_h = 2.5$



(a) 對稱式熱交換器

由於 $\frac{2+\Lambda}{\Pi} = \frac{2+10}{5} = 2.4 > 1.7$ ，由式18-22可知

$$\eta_{reg} \approx \frac{\Lambda}{2+\Lambda} = \frac{10}{2+10} = 0.867$$

(b) 平衡式交換器

由於 $\frac{\Pi_h}{\Lambda_h} = \frac{5}{10} = \frac{\Pi_c}{\Lambda_c} = \frac{10}{20} = 0.5 = k$ ，故為平衡式，計算平衡式再生式熱

交換器的縮減長度與縮減週期的和諧平均值 Λ_H 、 Π_H (由式18-24與式18-25)：

$$\frac{2}{\Pi_H} = \frac{1}{\Pi_c} + \frac{1}{\Pi_h} = \frac{1}{10} + \frac{1}{5} \rightarrow \Pi_H = 6.67$$

$$\frac{2}{\Lambda_H} = \frac{1}{\Pi_H} \left(\frac{\Pi_c}{\Lambda_c} + \frac{\Pi_h}{\Lambda_h} \right) = \frac{1}{6.67} \left(\frac{10}{20} + \frac{5}{10} \right) = 6.67$$

$$\therefore \Lambda_H = 3.33$$

$$\frac{2+\Lambda_H}{\Pi_H} = \frac{2+3.33}{6.67} = 0.8 < 1.7$$

由圖18-10估算可得 $\eta_{reg} \approx 0.59$



(c) 非平衡式交換器

由式 18-11，計算 $\gamma = \frac{\Pi_h}{\Lambda_h} \times \frac{\Lambda_c}{\Pi_c} = \frac{2.5}{10} \times \frac{10}{5} = 0.5$

由式 18-25，計算 Π_H ， $\frac{2}{\Pi_H} = \frac{1}{\Pi_c} + \frac{1}{\Pi_h} = \frac{1}{5} + \frac{1}{2.5} \rightarrow \Pi_H = 0.8333$

由式 18-24， $\frac{2}{\Lambda_H} = \frac{1}{\Pi_H} \left(\frac{\Pi_c}{\Lambda_c} + \frac{\Pi_h}{\Lambda_h} \right) = \frac{1}{0.8333} \left(\frac{5}{10} + \frac{2.5}{10} \right) = 0.9$

$\therefore \Lambda_H = 2.22$

$\frac{2 + \Lambda_H}{\Pi_H} = \frac{2 + 2.22}{0.8333} = 5.07 > 1.7$

由式 18-22 可知 $\eta_{reg} \approx \frac{\Lambda_H}{2 + \Lambda_H} = \frac{2.22}{2 + 2.22} = 0.526$

由式 18-26 計算 F ，即 $F = \frac{\eta_{reg,H} (1 - \gamma^2)}{2\gamma(1 - \eta_{reg,H})} = \frac{0.526 \times (1 - 0.5^2)}{2 \times 0.5 \times (1 - 0.526)} = 0.833$



修正 Λ - Π 法

再生式熱交換器的性能可以一整體熱交換係數 K 來表示，即：

$$E = KA(\bar{T}_h - \bar{T}_c)(P_h + P_c) \quad (18-29)$$

其中 E 代表冷熱週期的總熱量

$\bar{T}_h - \bar{T}_c$ 代表加熱週期與冷卻週期的整體平均溫差 (spatial and chronological average)，此一溫差可由進出口的對數平均溫差來表示：

$$\bar{T}_h - \bar{T}_c = \frac{(T_{h,in} - \bar{T}_{c,out}) - (\bar{T}_{h,out} - T_{c,in})}{\ln \left(\frac{T_{h,in} - \bar{T}_{c,out}}{\bar{T}_{h,out} - T_{c,in}} \right)}$$



週期的能量平衡可由下式表示：

$$E = i_{c,out} - i_{c,in} = i_{h,in} - i_{h,out} \quad (18-31)$$

若定義

$$Y = \frac{i_{c,out} - i_{c,in}}{i_{h,in} - i_{h,out}} \quad (18-32)$$

則式18-31可改寫成

$$\bar{T}_{h,out} = T_{h,in} - Y(\bar{T}_{c,out} - T_{c,in}) \quad (18-33)$$

另外若考慮加熱週期熱側的能量平衡(P_h)，

$$E = KA(\bar{T}_h - \bar{T}_c)(P_h + P_c) = i_{h,in} - i_{h,out} \quad (18-34)$$

同樣的，定義

$$X = \frac{KA(P_h + P_c)}{i_{h,in} - i_{h,out}} \quad (18-35)$$

將式18-35與18-30帶入式18-34整理後，可得：

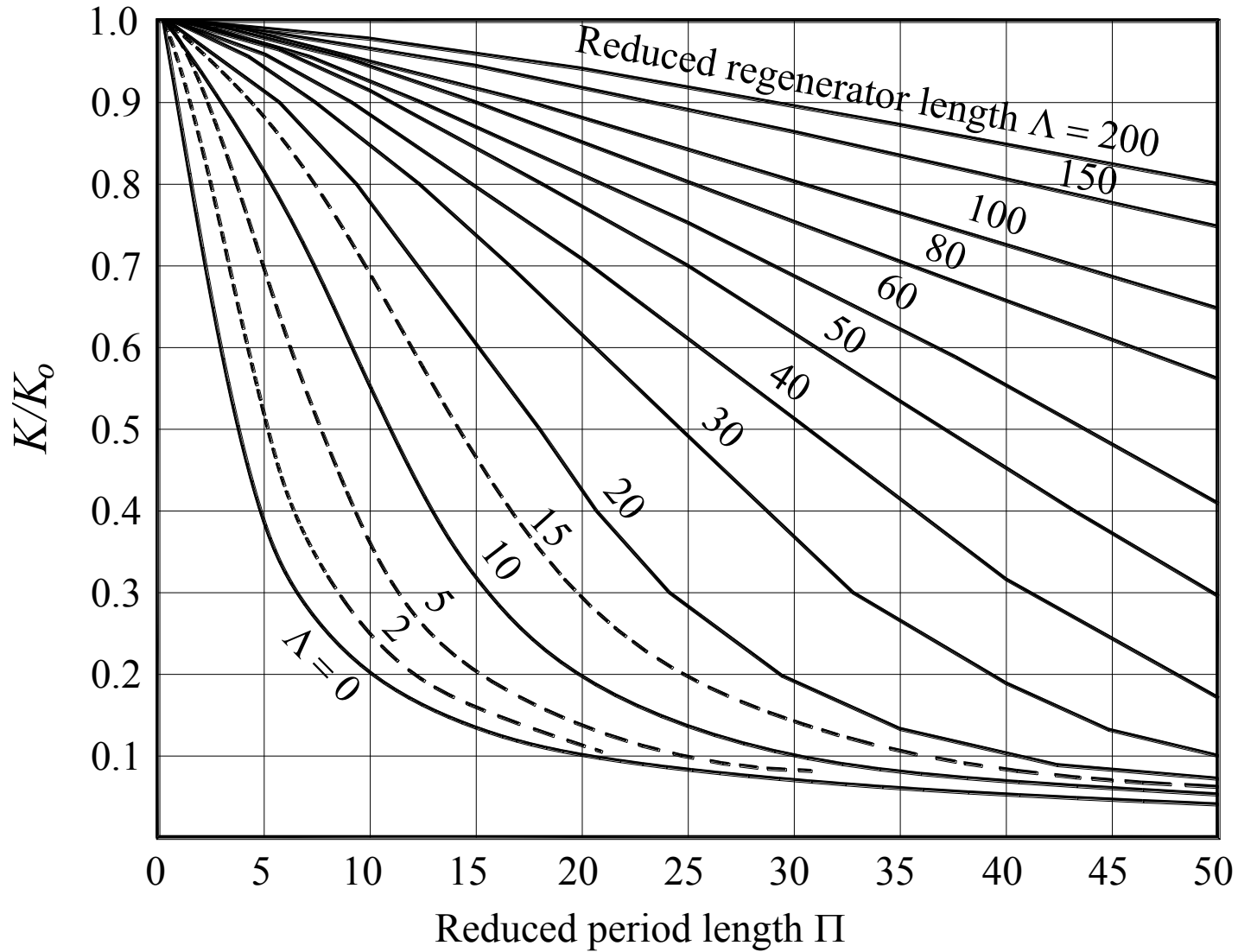
$$X \left(\frac{(T_{h,in} - \bar{T}_{c,out}) - (\bar{T}_{h,out} - T_{c,in})}{\ln \left(\frac{T_{h,in} - \bar{T}_{c,out}}{\bar{T}_{h,out} - T_{c,in}} \right)} \right) = T_{h,in} - T_{h,out} \quad (18-36)$$



$$\bar{T}_{c,out} = T_{c,in} + \frac{\left(1 - e^{X\left(1-\frac{1}{Y}\right)}\right) (T_{h,in} - T_{c,in})}{\left(1 - Ye^{X\left(1-\frac{1}{Y}\right)}\right)} \quad (18-37)$$

在上面的計算方法中，最重要的整體熱交換係數 K 的計算必須涵蓋加熱週期與冷卻週期間的熱傳性能，根據Hausen (1942)的研究，若氣體與熱交換器本體的溫度為線性變化， K_0 可表示如下：

$$\frac{1}{K_0} = \left(\frac{1}{h_h P_h} + \frac{1}{h_c P_c} \right) (P_h + P_c) \quad (18-38)$$



K/K_0 與 Λ 、 Π 間的關係圖



K/K_0 與 Λ 、 Π 間的關係如圖18-12所示，或可由下式來計算：

$$\frac{K}{K_0} = \frac{2\eta_{reg,h}}{\Lambda_H (1 - \eta_{reg,h})} \quad (18-39)$$

若是非平衡式的再生式熱交換器，Razelos (1979)的研究顯示 K/K_0 與 $\eta_{reg,h}$ 、 γ 有關，即：

$$\frac{K}{K_0} = \frac{4\gamma}{\Lambda_H (1 - \gamma^2)} \ln \left(\frac{1 - \gamma\eta_{reg,h}}{1 - \eta_{reg,h}} \right) \quad (18-40)$$

因此，一旦 K/K_0 算出，再生式熱交換器真正的熱傳量為：

$$E = \frac{K}{K_0} K_0 A (\bar{T}_h - \bar{T}_c) (P_h + P_c) \quad (18-41)$$



轉輪式再生式熱交換器

- 習慣上，轉輪式熱交換器的熱流設計方法以 NTU_0 法較為常用，此一設計法係由Coppage 與 London (1952) 發展而來，此法的基本假設與固定床型式相同
- 此法計算的觀念與先前介紹的 ε - NTU 法類似，轉輪式熱交換器的熱傳性能同樣的可以用有效度 ε 來表示



$\varepsilon = Q / Q_{max}$ ，如果 $C_c < C_h$ ，則

$$\varepsilon = \frac{Q}{Q_{max}} = \frac{C_c (T_{c,out} - T_{c,in})}{C_c (T_{h,in} - T_{c,out})} = \frac{T_{c,out} - T_{c,in}}{T_{h,in} - T_{c,out}} \quad (18-54)$$

同樣的，如果 $C_c > C_h$ ，則

$$\varepsilon = \frac{Q}{Q_{max}} = \frac{C_h (T_{h,in} - T_{h,out})}{C_h (T_{h,in} - T_{c,out})} = \frac{T_{h,in} - T_{h,out}}{T_{h,in} - T_{c,out}} \quad (18-55)$$

可以證明 $\varepsilon = \varepsilon(C^*, NTU, \text{流動形式})$ ，同樣的，對轉輪式熱交換器，一樣可證明如下的類似關係式

$$\varepsilon = \varepsilon\left(C^*, C_r^*, (hA)^*, NTU_o\right) \quad (18-56)$$

$$C^* = C_{\min}/C_{\max} \quad (18-57)$$

$$C_r^* = \frac{C_r}{C_{\min}} = \frac{M_m c_{p,m} \omega}{(\dot{m} c_p)_{\min}} \quad (18-58)$$

其中 ω 為轉輪的轉速(例如同常給定為 rpm 但計算上必須轉換成標準 SI 單位，即每秒幾轉)

$$(hA)^* = \frac{(hA)_{C_{\min}}}{(hA)_{C_{\max}}} \quad (18-59)$$

$$NTU_o = \frac{1}{C_{\min}} \left[\frac{1}{\frac{1}{(hA)_h} + \frac{1}{(hA)_c}} \right] \quad (18-60)$$



當 $C_r/C_{\min} \rightarrow \infty$ ，此時的轉輪式熱交換器的性能趨近於逆向流動的熱交換器，對逆流式的回復式熱交換器的有效度 (ε_{cf} ，注意 ε_{cf} 不是轉輪式熱交換器的有效度！)：

$$\varepsilon_{cf} = \begin{cases} \frac{1 - e^{-NTU_0(1-C^*)}}{1 - C^* e^{-NTU_0(1-C^*)}} & \text{if } C^* \neq 1 \\ \frac{NTU_0}{1 + NTU_0} & \text{if } C^* = 1 \end{cases} \quad (18-61)$$

通常，若 C_r/C_{\min} 的值為有限大小，Kays and London (1984) 根據 Lambertson (1958) 的研究，提出轉輪式熱交換器與逆流式回復式熱交換器間的經驗關係方程式如下：

$$\varepsilon = \varepsilon_{cf} \left(1 - \frac{1}{9(C_r^*)^{1.93}} \right) \quad (18-62)$$



故此，應用範圍通常落在該方程式的適用區內，讀者在真正的應用設計時也應特別注意；設計時要留意合理的應用範圍。一旦獲得轉輪式的有效度 ε ，則其熱交換量便可以下式算出(與第二章的回復式熱交換器設計方法完全相同)：

$$Q = \varepsilon C_{\min} (T_{h,i} - T_{c,i}) \quad (18-63)$$

式18-62僅適用於平衡式轉輪且範圍比較侷限，至於非平衡式的轉輪計算方法，Razelos (1979)則提出一等效式對稱方法，此方法計算有效度適用於 $C^* < 1$ 的條件，其有效度的計算過程如下所示：

$$(1) \quad NTU_{0,m} = \frac{2NTU_0 C^*}{1 + C^*} \quad (18-64)$$

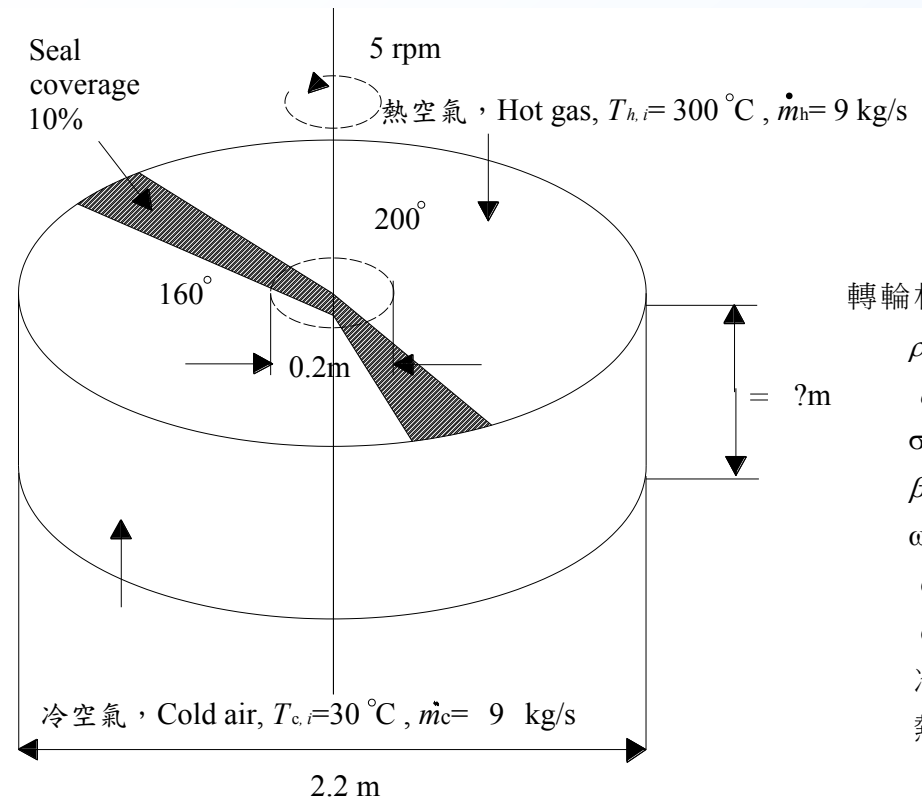
$$(2) \quad C_{r,m}^* = \frac{2C_r^* C^*}{1 + C^*} \quad (18-65)$$

$$(3) \quad \varepsilon_r = \frac{NTU_{0,m}}{1 + NTU_{0,m}} \left(1 - \frac{1}{9(C_{r,m}^*)^{1.93}} \right) \quad (18-66)$$

$$(4) \quad \varepsilon = \frac{1 - e^{-\frac{\varepsilon_r \left((C^*)^2 - 1 \right)}{2C^* (1 - \varepsilon_r)}}}{1 - C^* e^{-\frac{\varepsilon_r \left((C^*)^2 - 1 \right)}{2C^* (1 - \varepsilon_r)}}} \quad (18-67)$$



一轉輪式熱交換器用來回收高溫的廢熱空氣，此熱交換器的幾何尺寸與物理性質如下圖所示



轉輪材質與幾何尺寸如下：

$$\rho_m = 6000 \text{ kg/m}^3$$

$$c_{p,m} = 500 \text{ kJ/kg}\cdot\text{K}$$

$$\sigma \text{ (轉輪孔隙率, porosity)} = 0.75$$

$$\beta \text{ (轉輪密集度, packing density)} = 1300 \text{ m}^2/\text{m}^3$$

$$\omega \text{ (轉輪轉速)} = 5 \text{ rpm}$$

$$d \text{ (轉輪直徑)} = 2.2 \text{ m}$$

$$d_r \text{ (轉輪中心轉軸直徑)} = 0.2 \text{ m}$$

冷熱兩側因隔熱所產生的無效區域(seal coverage) = 10% = 0.1

熱交換器其他的操作條件如下：

	熱側	冷側
T_{in} (進口溫度)	400°C	30°C
氣體通過轉輪所佔的角度	200°	160°
c_p (比熱)	1120 J/kg·K	1006 J/kg·K
\dot{m}	9 kg/s	7 kg/s
h	50 W/m ² ·K	40 W/m ² ·K

根據上述的情況，如果設計希望達到0.9有效度的熱回收，則轉輪的厚度應為何？



先假設一個轉輪的厚度 $L = 0.5 \text{ m}$ ，然後再看看最後計算的有效度是否可達到 0.9，如果不夠，則可重新假設一個比較大的值，否則就假設小一點的值再試試看。在 $L = 0.5 \text{ m}$ 時的計算過程如下：

$$(a) C_h = i_{r,s} \times 1120 = 10080 \text{ W/K}$$

$$C_c = i_{r,s} \times 1006 = 7042 \text{ W/K}$$

$$\therefore C_{\min} = C_c = 7042 \text{ W/K}, C_{\max} = C_h = 10080 \text{ W/K}$$

$$C^* = \frac{C_{\min}}{C_{\max}} = \frac{7042}{10080} = 0.69861$$

(b) 由於熱側面積與冷側面積的比值轉輪在熱側的角度與冷側角度的

$$\text{比值，所以 } (hA)^* = \frac{(hA)_{C_{\min}}}{(hA)_{C_{\max}}} = \frac{40 \times 160^\circ}{50 \times 200^\circ} = 0.64$$

(c) 轉輪的重量(必須扣除轉輪中心轉軸部份的無效熱傳區域)

$$\begin{aligned} M_m &= \rho_m V_m = \rho_m \times \left[\frac{\pi}{4} (d^2 - d_r^2) L \times (1 - \text{孔隙率, Prosimity}) \right] \\ &= 8000 \times \left[\frac{\pi}{4} (2.2^2 - 0.2^2) \times 0.5 \times (1 - 0.75) \right] = 3769.9 \text{ kg} \end{aligned}$$

(d)

$$\begin{aligned} C_r^* &= \frac{C_r}{C_{\min}} = \frac{M_m c_{p,m} \omega}{(i_{r,s})_{\min}} \\ &= \frac{3769.9 \times 500 \times \frac{5}{60} \text{ (rpm必須換為以秒計算的標準SI單位)}}{7042} \\ &= 22.306 \end{aligned}$$



(e) 計算熱交換的有效總面積 A

$A =$ 轉輪截面積 \times 轉輪厚度 $(L) \times$ 密集度 $(\beta) \times (1 - \text{隔熱所產生的無效區域})$

$$= \frac{\pi}{4} (2.2^2 - 0.2^2) \times 0.5 \times 1300 (1 - 0.1) = 2205.4 \text{ m}^2$$

$$\therefore \text{加熱側的總面積 } A_h = A \times 200^\circ / 360^\circ = 1225.2 \text{ m}^2$$

$$\text{冷卻側的總面積 } A_c = A \times 160^\circ / 360^\circ = 980.18 \text{ m}^2$$

(f) $(hA)_h = 50 \times 1225.2 = 61261 \text{ W/K}$

$(hA)_c = 40 \times 980.18 = 39207 \text{ W/K}$

(g)
$$NTU_o = \frac{1}{C_{\min}} \left[\frac{1}{\frac{1}{(hA)_h} + \frac{1}{(hA)_c}} \right] = \frac{1}{7042} \left(\frac{1}{\frac{1}{61261} + \frac{1}{39207}} \right) = 3.3949$$

(h) 由式 18-61 (由於 $C^* \neq 1$)

$$\varepsilon_{cf} = \frac{1 - e^{-NTU_o(1-C^*)}}{1 - C^* e^{-NTU_o(1-C^*)}} = \frac{1 - e^{-3.3949(1-0.69861)}}{1 - 0.69861 \times e^{-3.3949(1-0.69861)}} = 0.85534$$

(i) 由式 18-62

$$\varepsilon = \varepsilon_{cf} \left(1 - \frac{1}{9(C_r^*)^{1.93}} \right) = 0.85534 \times \left(1 - \frac{1}{9 \times 22.306^{1.93}} \right) = 0.8551$$



計算上也可以利用 Razelos (1979) 式 18-64~18-67 的算法如下：

$$(j) \text{ 由式 18-62, } NTU_{0,m} = \frac{2NTU_0 C^*}{1+C^*} = \frac{2 \times 3.3949 \times 0.69861}{1+0.69861} = 2.7925$$

$$(k) C_{r,m}^* = \frac{2C_r^* C^*}{1+C^*} = \frac{2 \times 22.306 \times 0.69861}{1+0.69861} = 18.348$$

$$(l) \quad \varepsilon_r = \frac{NTU_{0,m}}{1+NTU_{0,m}} \left(1 - \frac{1}{9(C_{r,m}^*)^{1.93}} \right) = \frac{2.7925}{1+2.7925} \left(1 - \frac{1}{9 \times 18.348^{1.93}} \right) = 0.73603$$

$$(m) \quad \varepsilon = \frac{1 - e^{-\frac{\varepsilon_r((C^*)^2 - 1)}{2C^*(1-\varepsilon_r)}}}{1 - C^* e^{-\frac{\varepsilon_r((C^*)^2 - 1)}{2C^*(1-\varepsilon_r)}}} = \frac{1 - e^{-\frac{0.73603(0.69861^2 - 1)}{2 \times 0.69861(1-0.73603)}}}{1 - 0.69861 \times e^{-\frac{0.73603(0.69861^2 - 1)}{2 \times 0.69861(1-0.73603)}}} = 0.85503$$

此依計算方法與式 18-61 的計算結果幾乎相同，不過此一計算值低於設計值 0.9，因此必須重新假設厚度 L ，經過幾次的嘗試後，可算出 $L \approx 0.6418 \text{ m}$ 。順帶一提，由於熱效率 $\varepsilon = 0.9$ ，且 $C_{\min} = C_c$ ，因此由式 18-54，

$$\varepsilon = \frac{Q}{Q_{\max}} = \frac{C_c (T_{c,out} - T_{c,in})}{C_c (T_{h,in} - T_{c,out})} = \frac{T_{c,out} - T_{c,in}}{T_{h,in} - T_{c,out}}, \text{ 可算出冷熱側的出口溫度。}$$



常見之熱回收熱交換器



何謂廢熱？

- “Dumped” heat that can still be reused
- “Value” (quality) more important than quantity
- Waste heat recovery saves fuel

<i>S. No</i>	<i>Source of Waste Heat</i>	<i>Quality of Waste Heat</i>
1	Heat in flue gases	The higher the temperature, the greater the potential value for heat recovery
2	Heat in vapour streams	As above but when condensed, latent heat also recoverable
3	Convective & radiant heat lost from exterior of equipment	Low grade – if collected may be used for space heating or air preheats
4	Heat losses in cooling water	Low grade – useful gains if heat is exchanged with incoming fresh water
5	Heat losses in providing chilled water or in the disposal of chilled water	1.High grade if it can be utilized to reduce demand for refrigeration 2.Low grade if refrigeration unit used as a form of Heat pump
6	Heat stored in products leaving the process	Quality depends upon temperature
7	Heat in gaseous & liquid effluents leaving process	Poor if heavily contaminated & thus requiring alloy heat exchanger



High Temperature Heat Recovery

Table: Typical waste heat temperature at high temperature range from various sources

<i>Types of Devices</i>	<i>Temperature (°C)</i>
Nickel refining furnace	1370 – 1650
Aluminium refining furnace	650 – 760
Zinc refining furnace	760 – 1100
Copper refining furnace	760 – 815
Steel heating furnace	925 – 1050
Copper reverberatory furnace	900 – 1100
Open hearth furnace	650 – 700
Cement kiln (Dry process)	620 – 730
Glass melting furnace	1000 – 1550
Hydrogen plants	650 – 1000
Solid waste incinerators	650 – 1000
Fume incinerators	650 – 1450



Medium Temperature Heat Recovery

Table: Typical waste heat temperature at medium temperature range from various sources

<i>Types of Devices</i>	<i>Temperature (°C)</i>
Steam boiler exhaust	230 – 480
Gas turbine exhaust	370 – 540
Reciprocating engine exhaust	315 – 600
Reciprocating engine exhaust (turbo charged)	230 – 370
Heat treatment furnace	425 – 650
Drying & baking ovens	230 – 600
Catalytic crackers	425 – 650
Annealing furnace cooling systems	425 – 650



Source	Temperature °C
Process steam condensate	55-88
Cooling water from: Furnace doors	32-55
Bearings	32-88
Welding machines	32-88
Injection molding machines	32-88
Annealing furnaces	66-230
Forming dies	27-88
Air compressors	27-50
Pumps	27-88
Internal combustion engines	66-120
Air conditioning and refrigeration condensers	32-43
Liquid still condensers	32-88
Drying, baking and curing ovens	93-230
Hot processed liquids	32-232
Hot processed solids	93-232

Table: Typical waste heat temperature at low temperature range from various sources



常見之熱回收熱交換器 - recuperator

Commercial Waste Heat Recovery

✓ Recuperators

- Heat exchange between flue gases and the air through metallic/ceramic walls
- Ducts/tubes carry combustion air for preheating
- Waste heat stream on other side

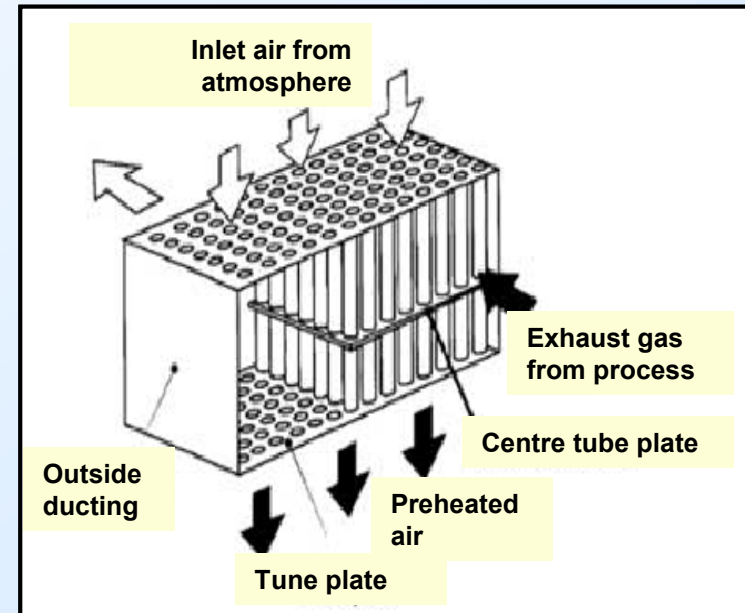


Figure 1 : Waste heat recovery using recuperator, Source: SEAV



常見之熱回收熱交換器 - recuperator

Commercial Waste Heat Recovery

- ✓ **Metallic radiation recuperators**
 - Simplest recuperator
 - Two metal tubes
 - Less fuel is burned per furnace load
 - Heat transfer mostly by radiation

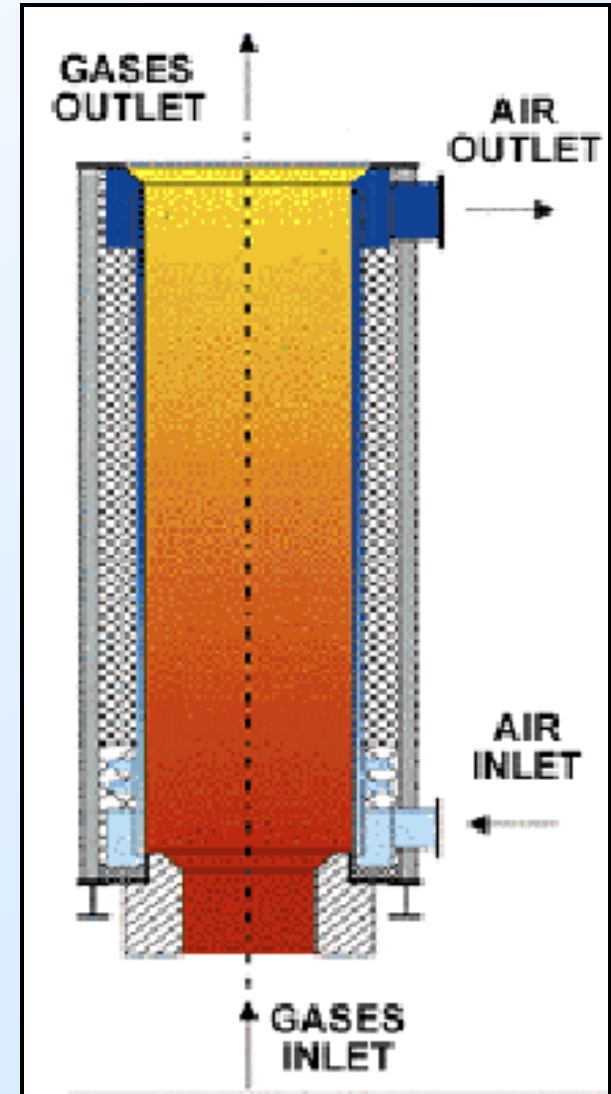


Figure 2. Metallic Radiation Recuperator (Hardtech Group)



常見之熱回收熱交換器 - recuperator

Commercial Waste Heat Recovery

✓ Convective recuperators

- Hot gas through parallel small diameter tubes
- Tubes can be baffled to allow gas to pass over them again
- Baffling increases heat exchange but more expensive exchanger is needed

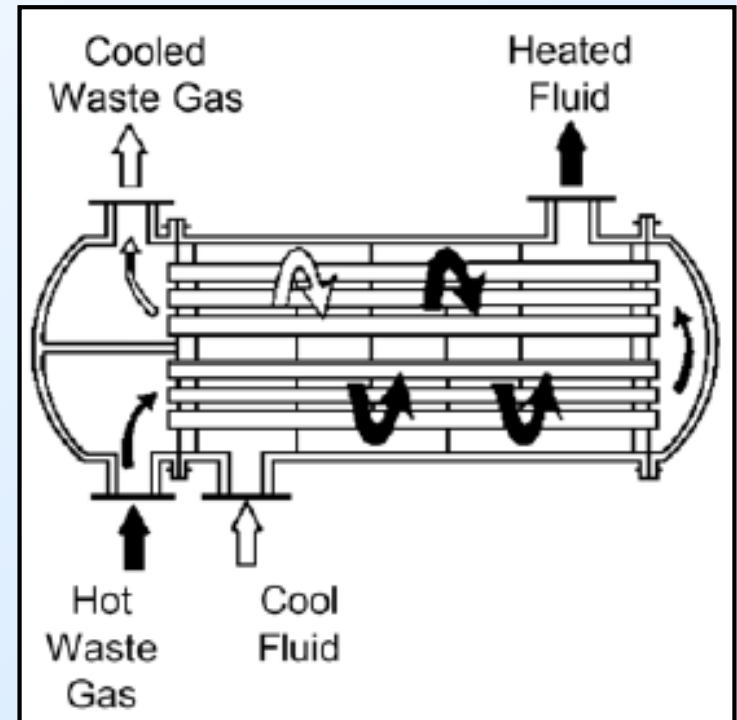


Figure 3. Convective Recuperator
(Reay, D.A., 1996)



常見之熱回收熱交換器 - recuperator

Commercial Waste Heat Recovery

- ✓ Radiation/convective hybrid recuperators
- Combinations of radiation & convection
- More effective heat transfer
- More expensive but less bulky than simple metallic radiation recuperators

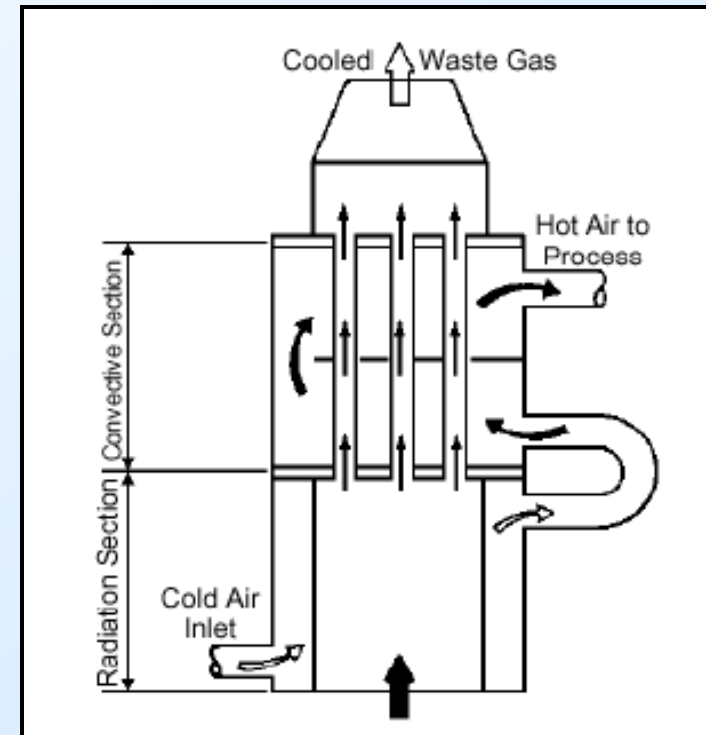


Figure 4. Hybrid Recuperator
(Reay, D.A., 1996)



常見之熱回收熱交換器 - recuperator

Commercial Waste Heat Recovery

✓ Ceramic recuperators

- Less temperature limitations:
 - Operation on gas side up to 1550 °C
 - Operation on preheated air side to 815 °C
- New designs
 - Last two years
 - Air preheat temperatures <700 °C
 - Lower leakage rates



常見之熱回收熱交換器 - Regenerator

Regenerator

- Large capacities
- Glass and steel melting furnaces
- Time between the reversals important to reduce costs
- Heat transfer in old regenerators reduced by
 - Dust & slagging on surfaces
 - heat losses from the walls

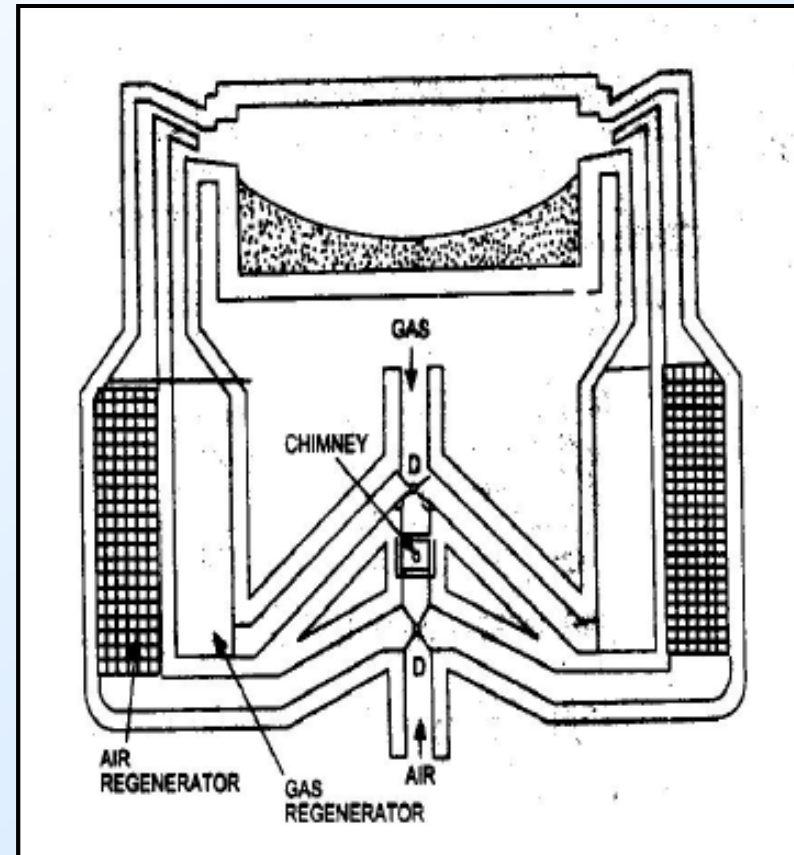


Figure 5. Regenerator
(Department of Coal, India, 1985)



常見之熱回收熱交換器 - regenerator

Heat Wheels

- Porous disk rotating between two side-by-side ducts
- Low to medium temperature waste heat recovery systems
- Heat transfer efficiency up to 85 %

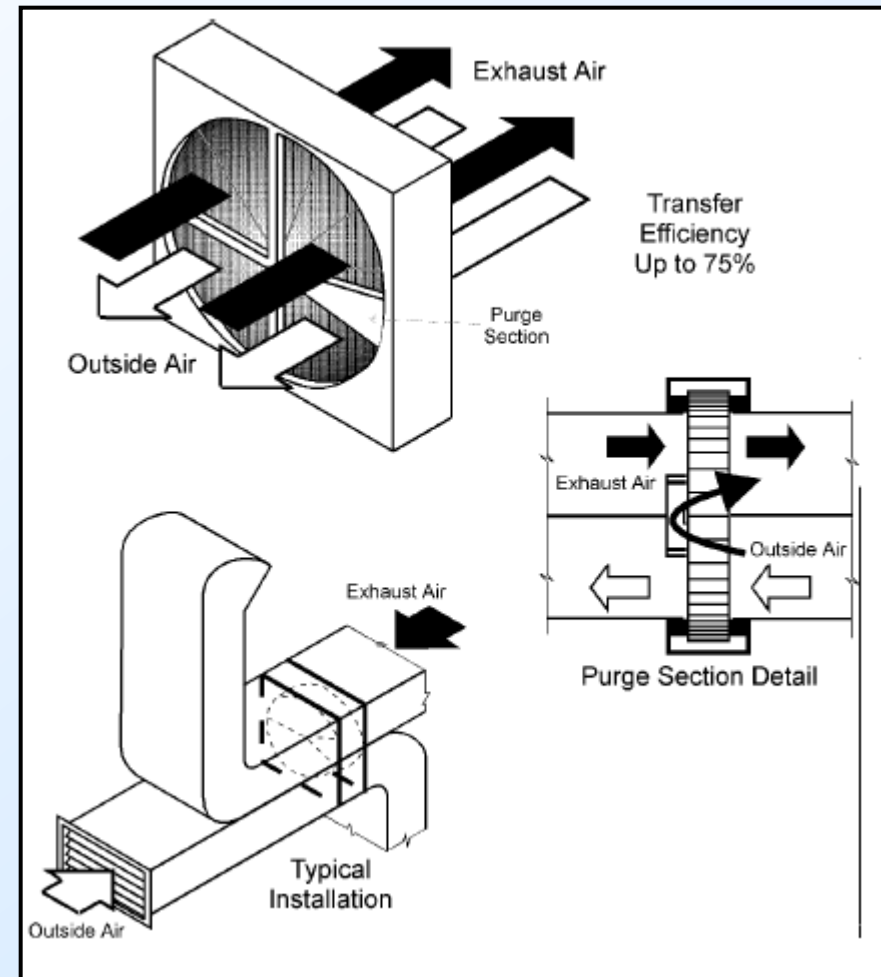


Figure 6. Heat Wheel
(SADC, 1999)



常見之熱回收熱交換器 – Heat pipe Heat Exchanger

Heat Pipe

- Transfer up to 100 times more thermal energy than copper
- Three elements:
 - sealed container
 - capillary wick structure
 - working fluid
- Works with evaporation and condensation

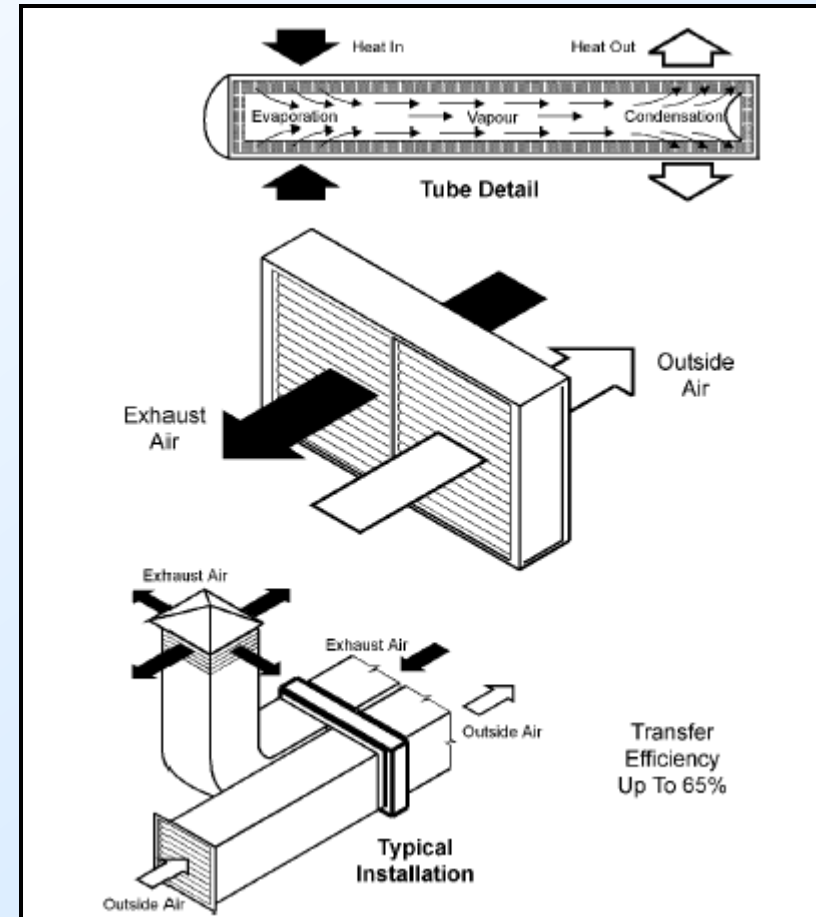


Figure 7. Heat Pipe
(SADC, 1999)



常見之熱回收熱交換器 – Heat pipe Heat Exchanger

Heat Pipe

✓ Performance and advantage

- Lightweight and compact
- No need for mechanical maintenance, input power, cooling water and lubrication systems
- Lowers the fan horsepower requirement and increases the overall thermal efficiency of the system
- Can operate at 315 °C with 60% to 80% heat recovery



常見之熱回收熱交換器 – Heat pipe Heat Exchanger

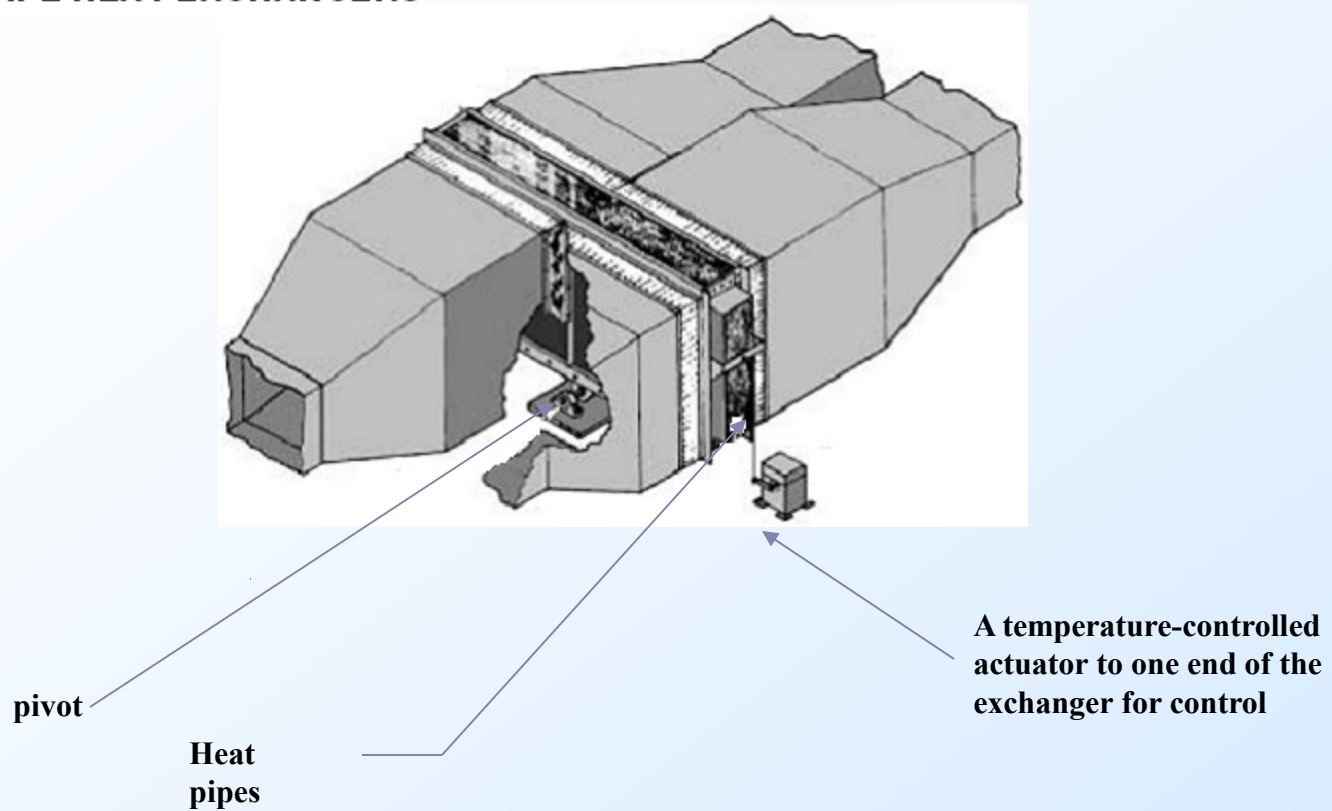
Heat Pipe

✓ Typical application

- **Process to space heating**
 - Transfers thermal energy from process exhaust for building heating
- **Process to process**
 - Transfers recovered waste thermal energy from the process to the incoming process air
- **HVAC applications**
 - Cooling and heating by recovering thermal energy



HEAT PIPE HEAT EXCHANGERS



In practice, tilt control is effected by pivoting the exchanger about the center of its base.



常見之熱回收熱交換器 – Economizer

Economizer

- Utilize the flue gas heat for pre-heating the boiler feed water
- 1% fuel savings if
 - 60 °C rise of feed water
 - 200 °C rise in combustion air temp

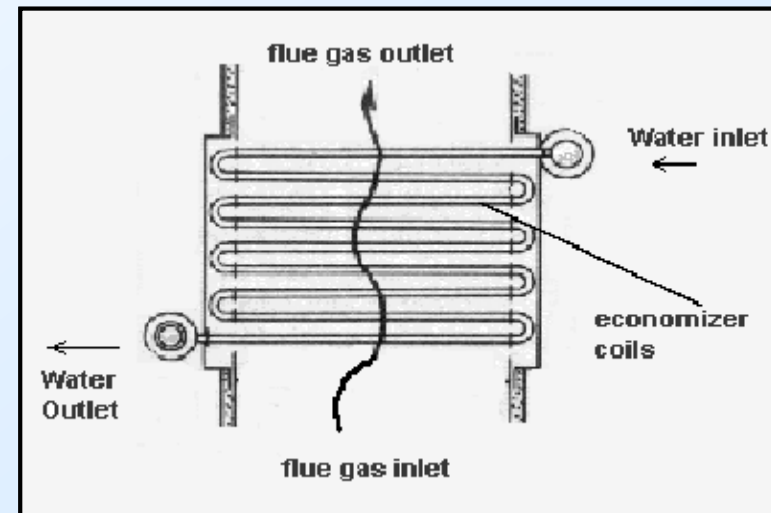


Figure 8. Economizer
(Bureau of Energy Efficiency,
2004)



常見之熱回收熱交換器 – Economizer

Economizer

✓ Shell and tube heat exchanger

- Used when the medium containing waste heat is a liquid or a vapor that heats another liquid

- Shell contains the tube bundle, and usually internal baffles to direct the fluid

- Vapor contained within the shell

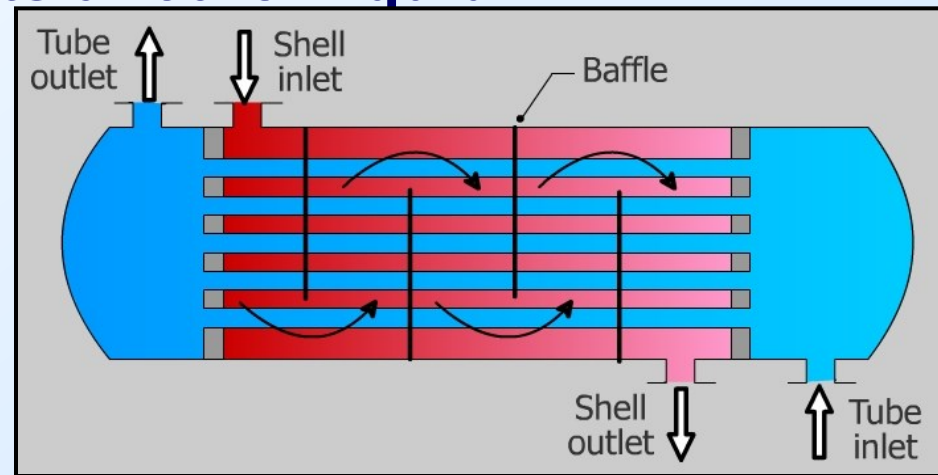


Figure 9. Shell & Tube Heat Exchanger
(King Fahad University of Petroleum & Minerals, 2003)



常見之熱回收熱交換器 – Economizer

Plate Heat Exchanger

- Parallel plates forming a thin flow pass
- Avoids high cost of heat exchange surfaces
- Corrugated plates to improve heat transfer
- When directions of hot and cold fluids are opposite, the arrangement is counter current

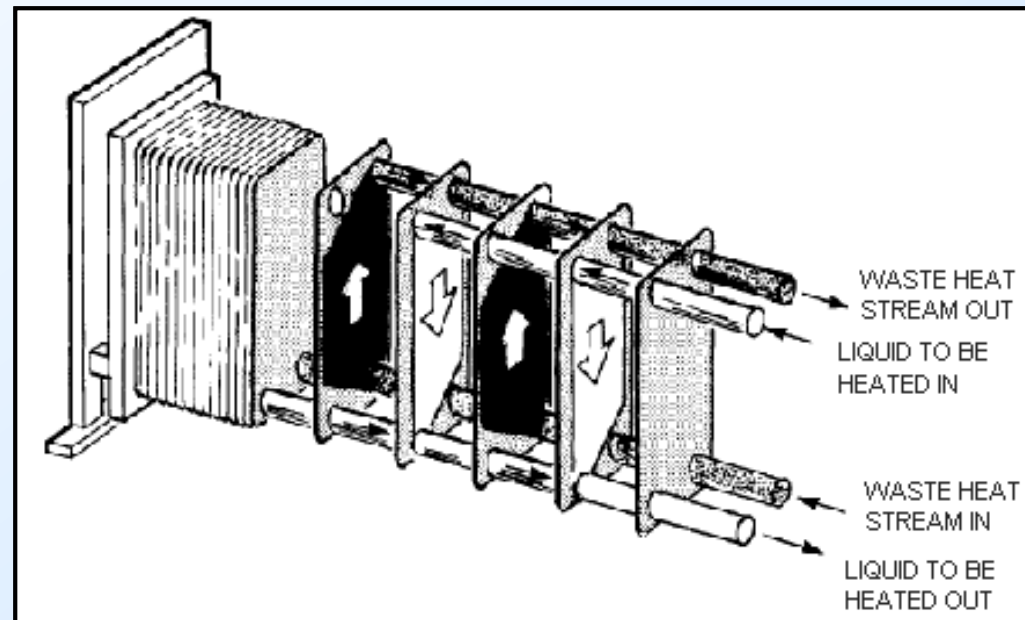


Figure 10. Plate Heat Exchanger
(Canada Agriculture and Agri-Food)



Type of Waste Heat Recovery

Plate Heat Exchanger

✓ Run around coil exchanger

- Heat transfer from hot to colder fluid via *heat transfer fluid*
- One coil in hot stream
- One coil in cold stream

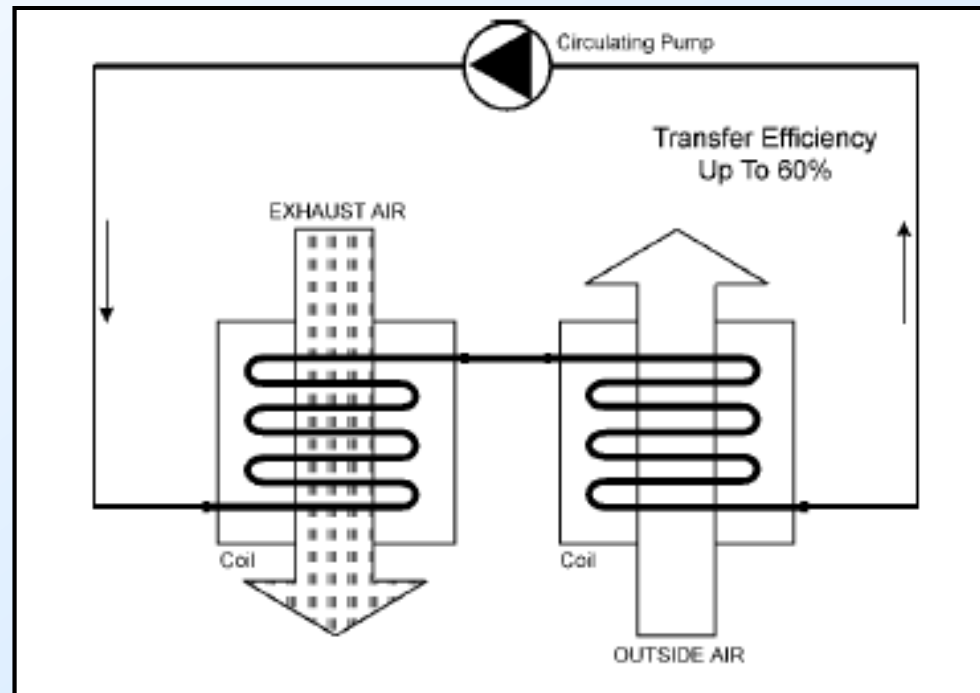


Figure 11. Run Around Coil Exchanger
(SADC , 1999)



Type of Waste Heat Recovery

Plate Heat Exchanger

✓ Waste heat boiler

- Water tube boiler: hot exhaust gases pass over parallel tubes with water
- Capacities: 25 m³ to 30,000 m³/min of exhaust gas

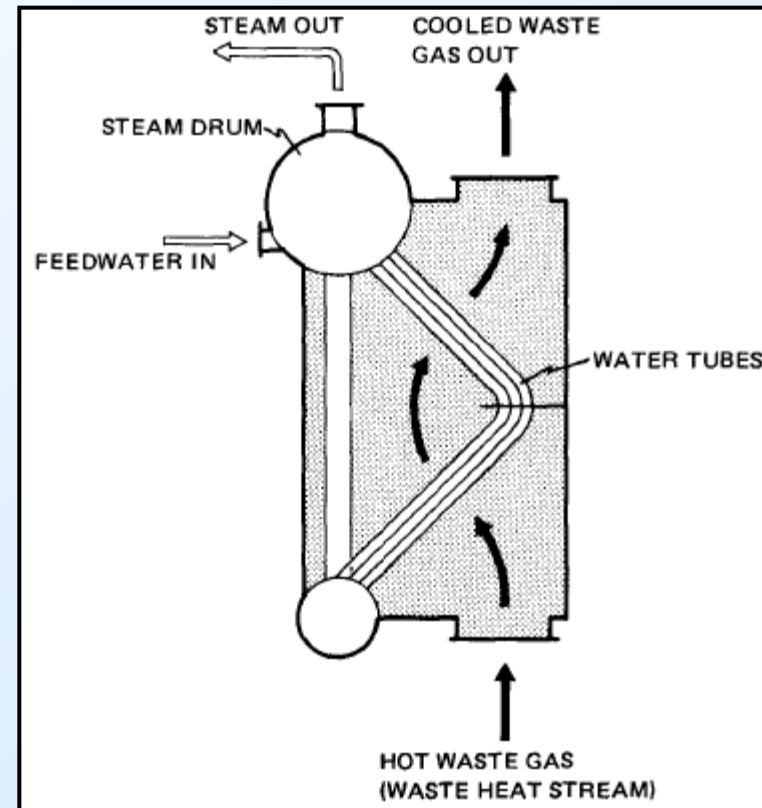


Figure 12. Two-Pass Water Tube Waste Heat Recovery Boiler
(Canada Agriculture and Agri-Food)



Type of Waste Heat Recovery

Heat Pump

The vapour compression cycle

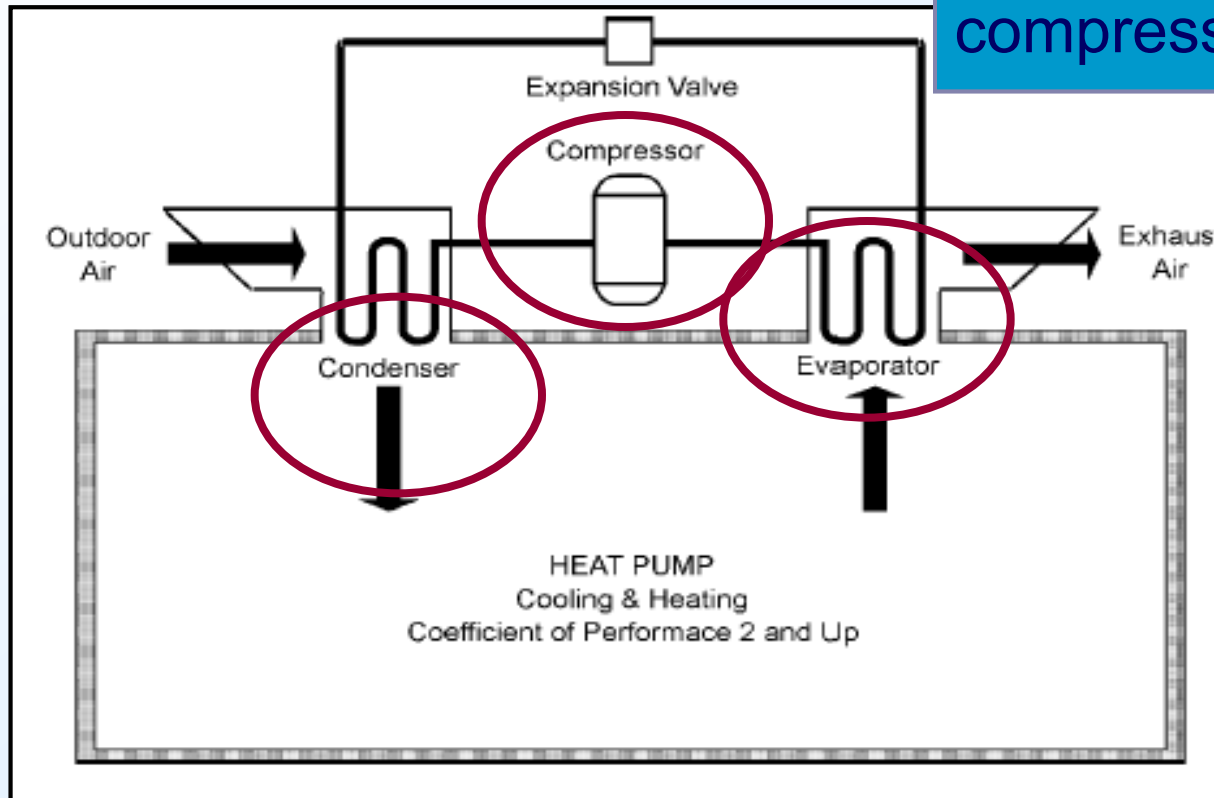


Figure 13. Heat Pump Arrangement
(SADC, 1999)



Type of Waste Heat Recovery

Heat Pump

- ✓ Developed as a space heating system
- ✓ Can upgrade heat $>2X$ the energy consumed by the device
- ✓ Most promising when heating and cooling capabilities are combined



Type of Waste Heat Recovery

Heat Pump

✓ Thermo compressor

- Compress low-pressure steam by very high-pressure steam and reuse as medium pressure steam
- Nozzle for acceleration of HP steam to a high velocity fluid.

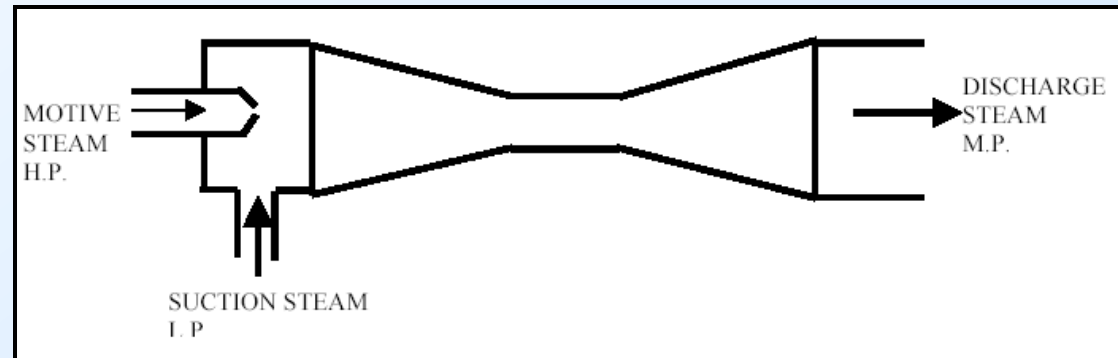


Figure: Thermo compressor



End of 3rd Talk,
Questions?