



液冷技術

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The Criterion for Acceptable Cooling in a Data Center

- Manufacturers of computer servers design their equipment with a certain allowable maximum inlet temperature. This value is around 24°C (75°F).
- The air-conditioners in a data center usually supply cold air at 13°C 55°F . If the 13°C cold air enters the servers, there is no difficulty in satisfying the manufacturer's criterion. But the cold air does not always enter at all the inlet locations on the server rack.
- Often, the hot air exhausted by the rack finds its way to the inlet of the same rack or some other rack. This is how the cooling in a data center is compromised.



Acknowledgements

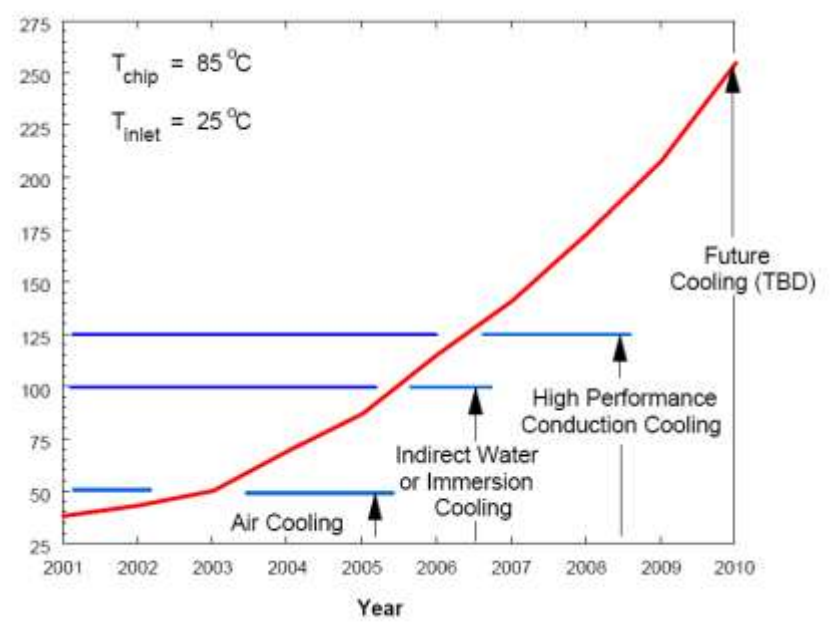
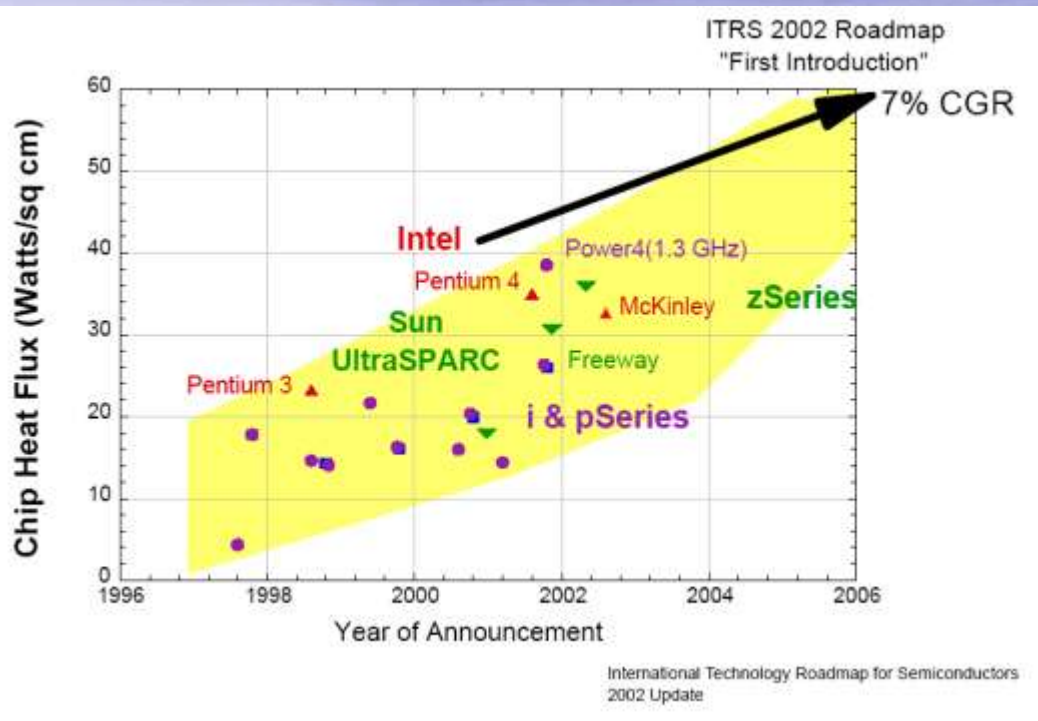
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- Technical Supports from Dr. K.S. Yang, Miss CoCo Hsu, Mr. M.C. Lu, and Dr. K.H. Chien
- Advisory support from Dr. R.J. Shyu
- Major Collaboration Professors: I.Y. Chen from Yunling Tech. Univ.



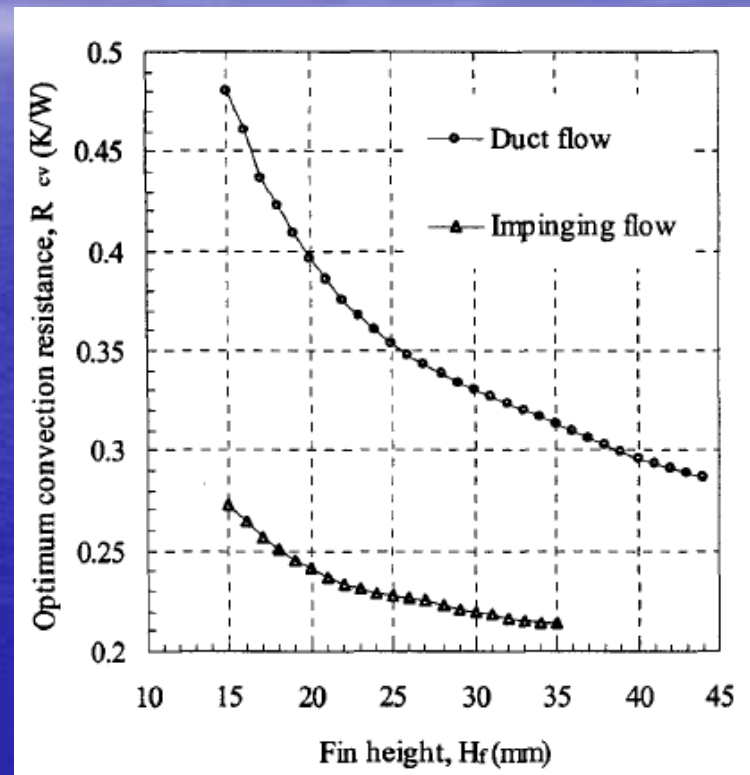
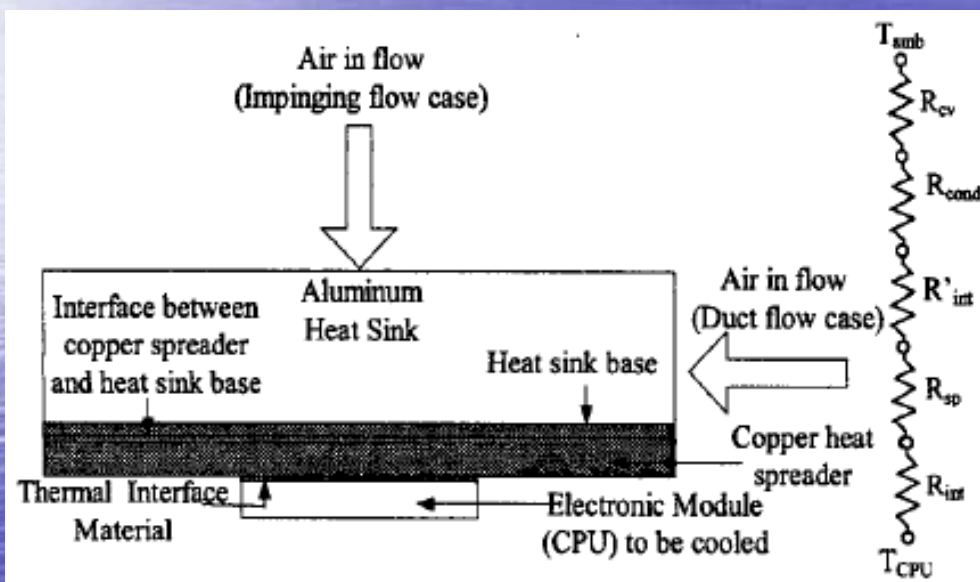
Outline

- **Background**
- **A Review of Micropump**
- **Fundamentals of Augmentation**
- **Micro-channel HXs**
- **Liquid Cooling – Maldistribution**
- **Liquid Cooling – Radiators**
- **Summary**

Limits of cooling technology



Limits of Air-cooled Design

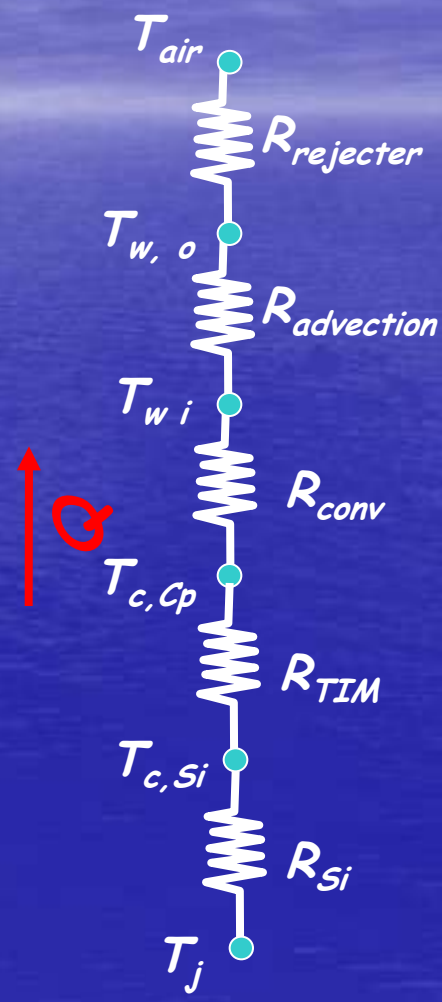
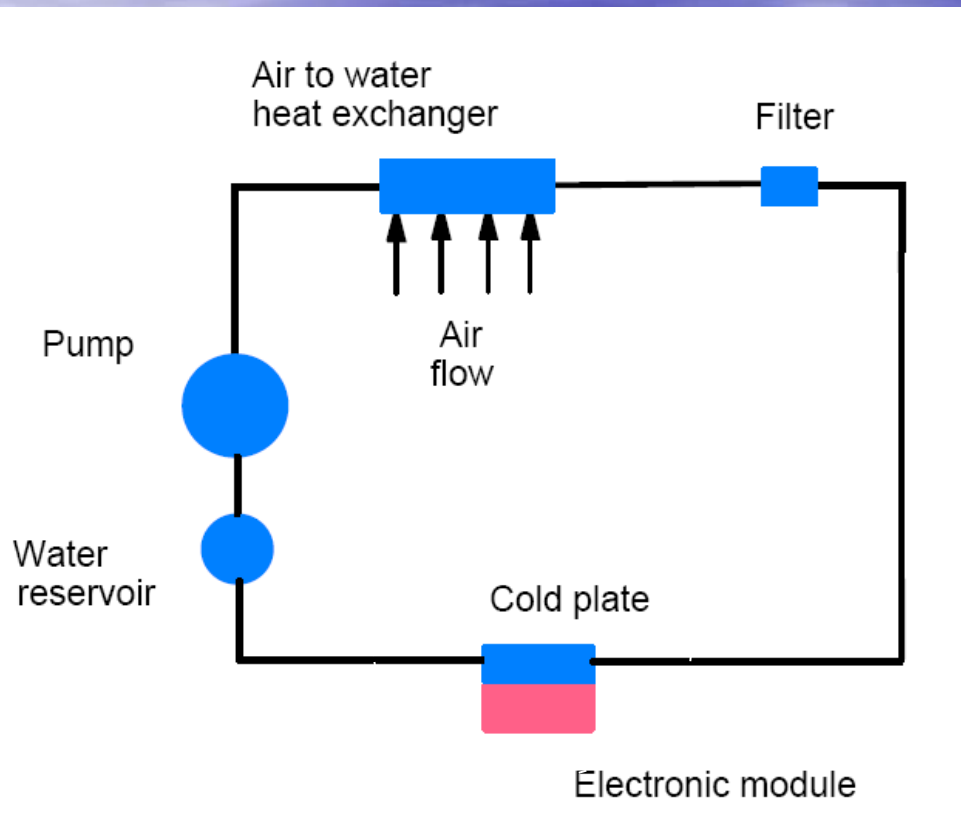




Major Components of Liquid Cooling

- Micro-pump
- Cold Plate
- Radiator – (Fin and Tube Heat Exchanger)
- Connection piping

Water-cooled System



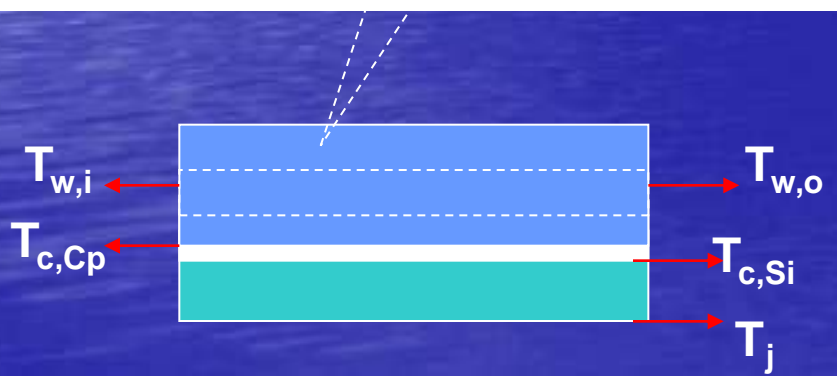
$R_{rejecter}$: Heat Transfer between water and HX

$$R_{advection} = \frac{1}{\rho \dot{V} C_p}$$

$$R_{convection} = \frac{1}{hA_{ch}}$$

$$R_{TIM} = 0.05 \sim 0.1 W / ^\circ C$$

$$R_{Si} = \frac{d}{kA}$$





Hitachi Water Cooling Laptop (Prototype Model)





A quick overview of micropump

Micropump Applications

Biotechnology

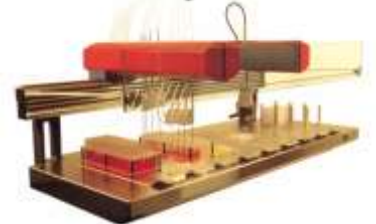


Medical Device & Diagnostic Equipment



Electronics, Semiconductors, & Microelectronics

Clinical & Analytical Laboratory Instruments



Food & Beverage



Aerospace, Aircraft, & Airline

Chemical Mixing & Processing



Environmental & Water Treatment



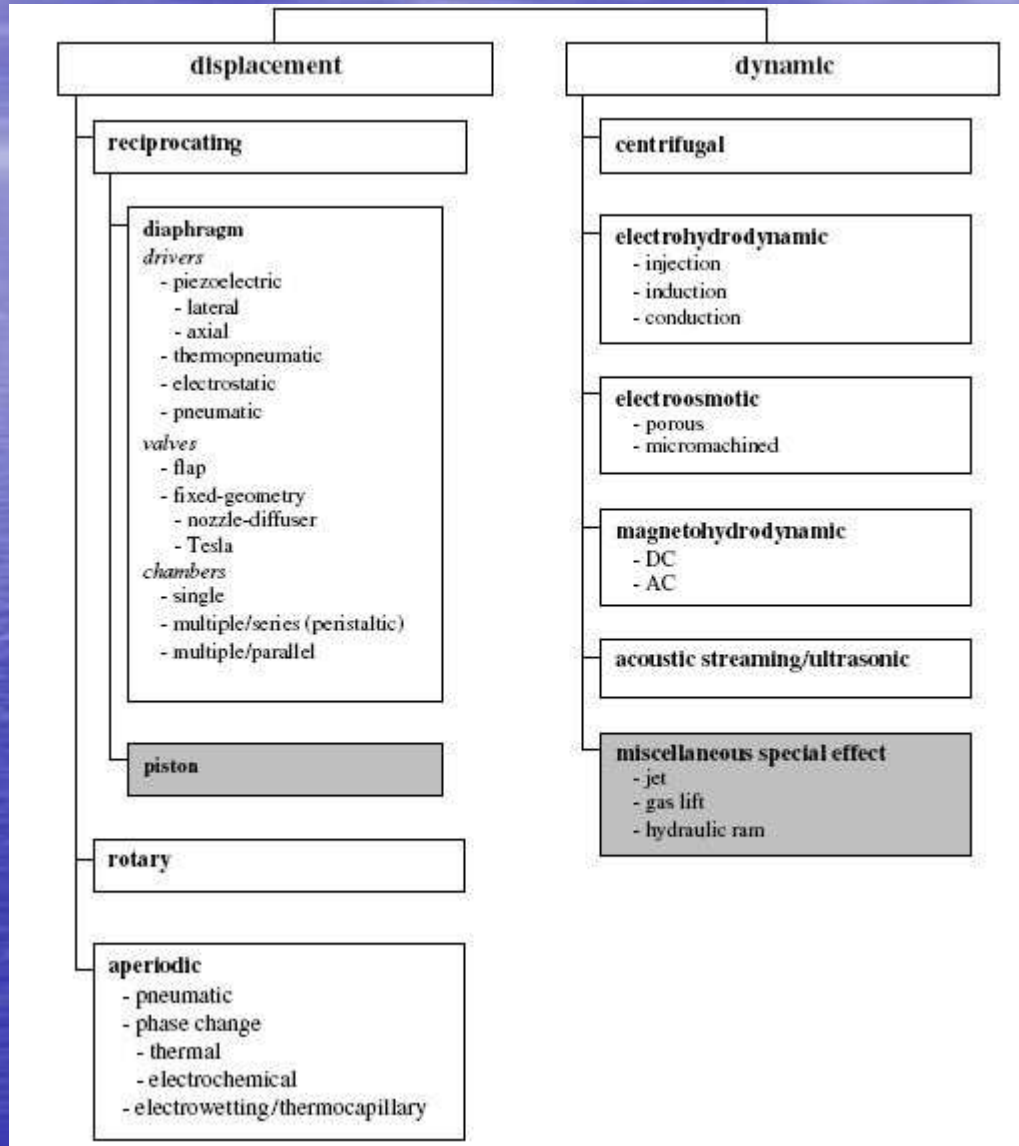
General Industry



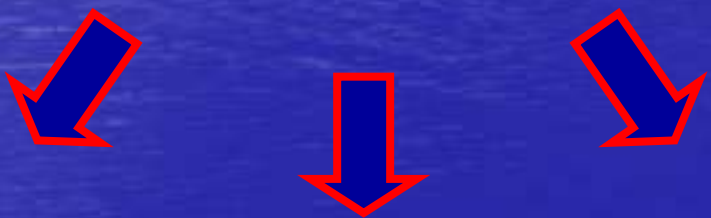
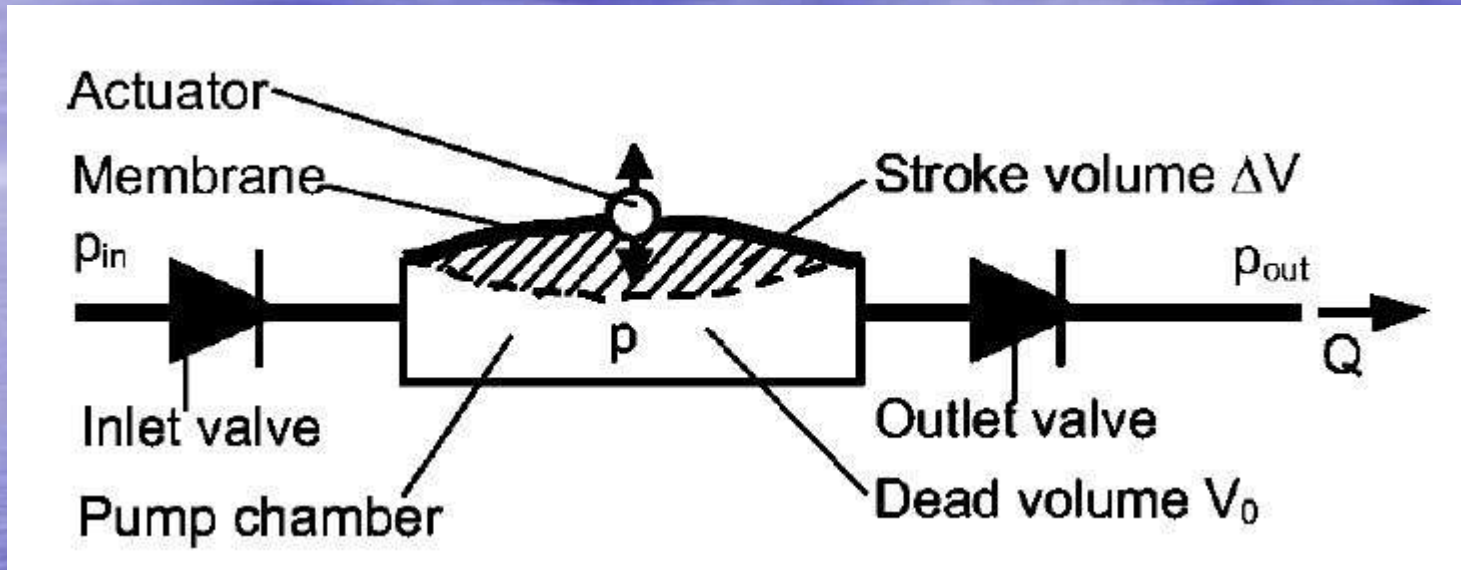
Energy & Fuels



Classification of micropumps



General structure of a micro check-valve pump



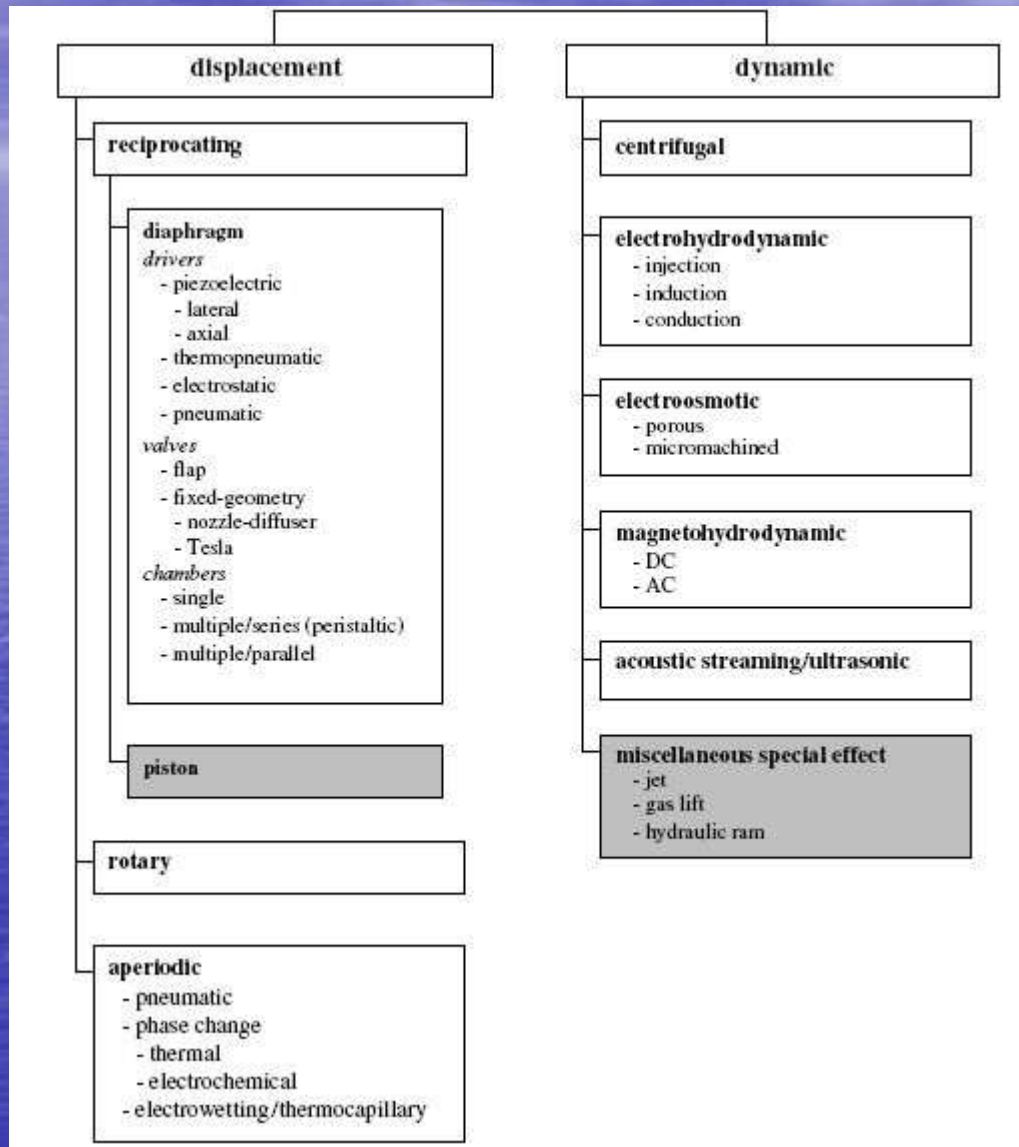
Micro channel

Micro actuator

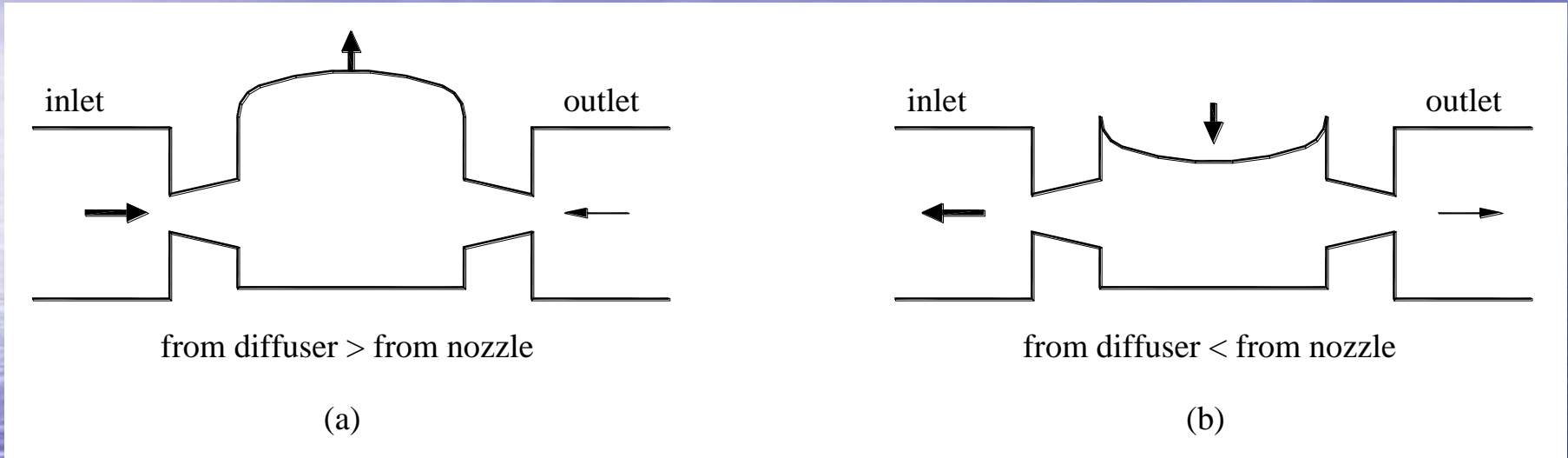
Micro valve



Classification of micropumps

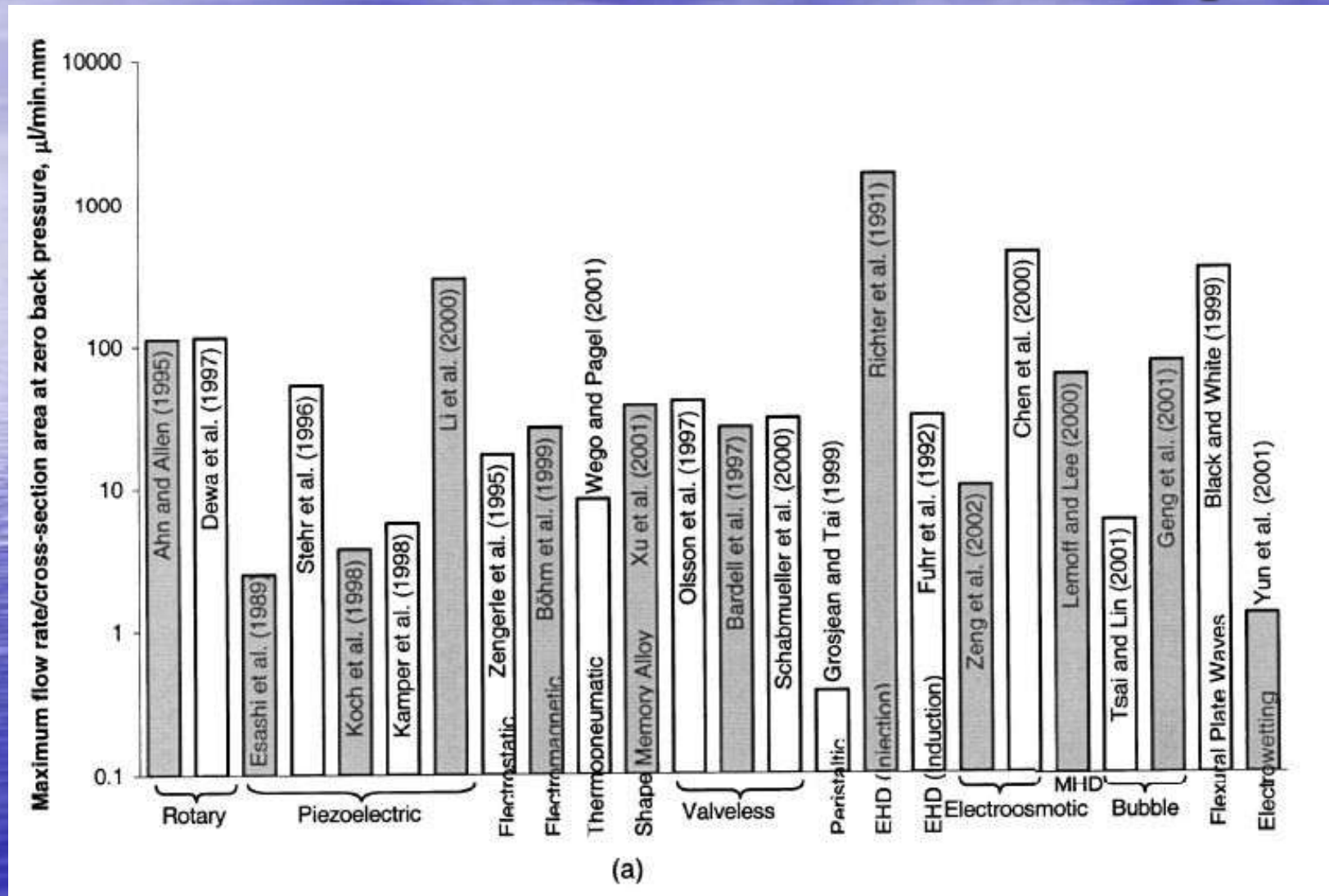


Micro valve – valve less

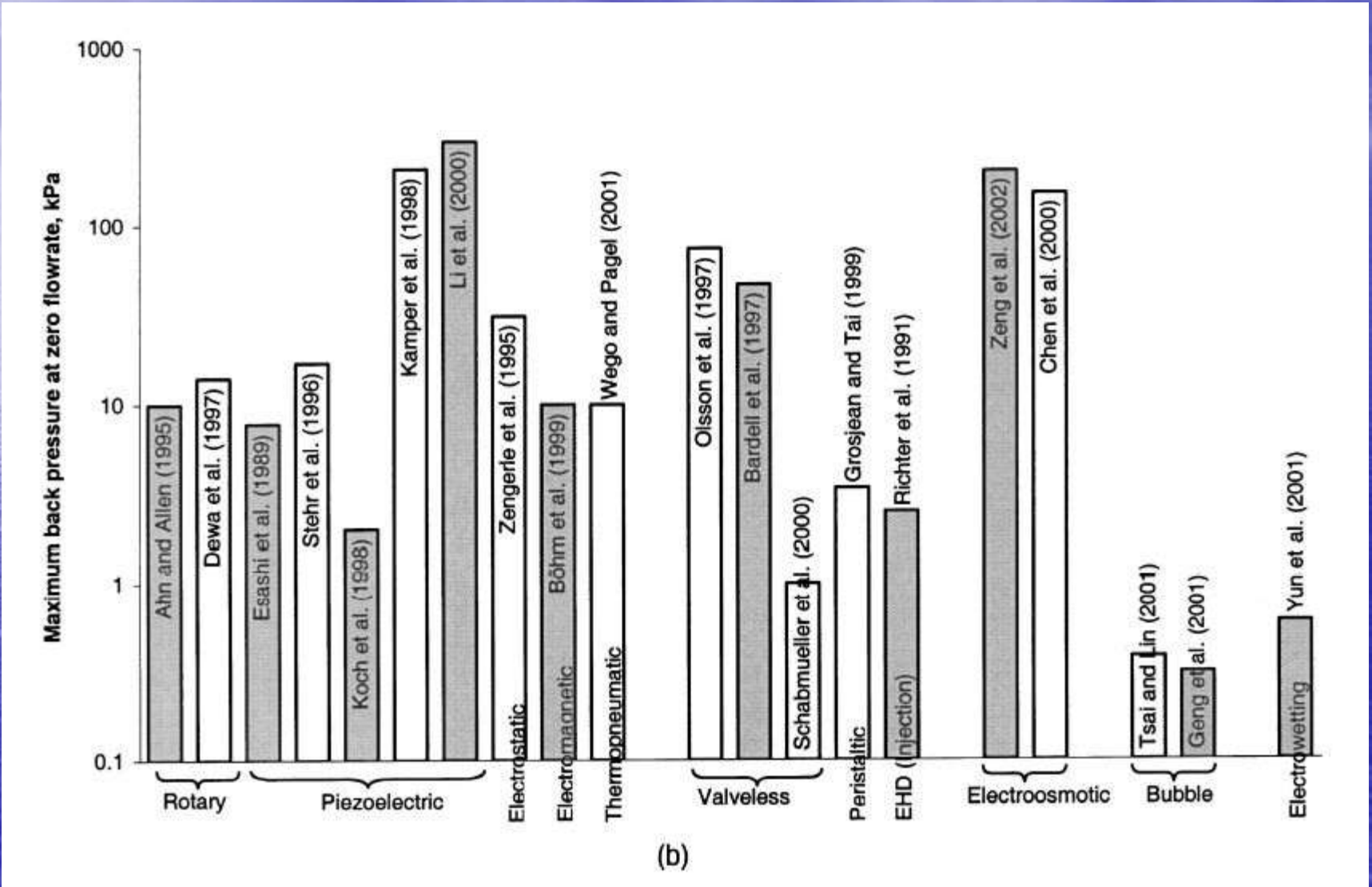


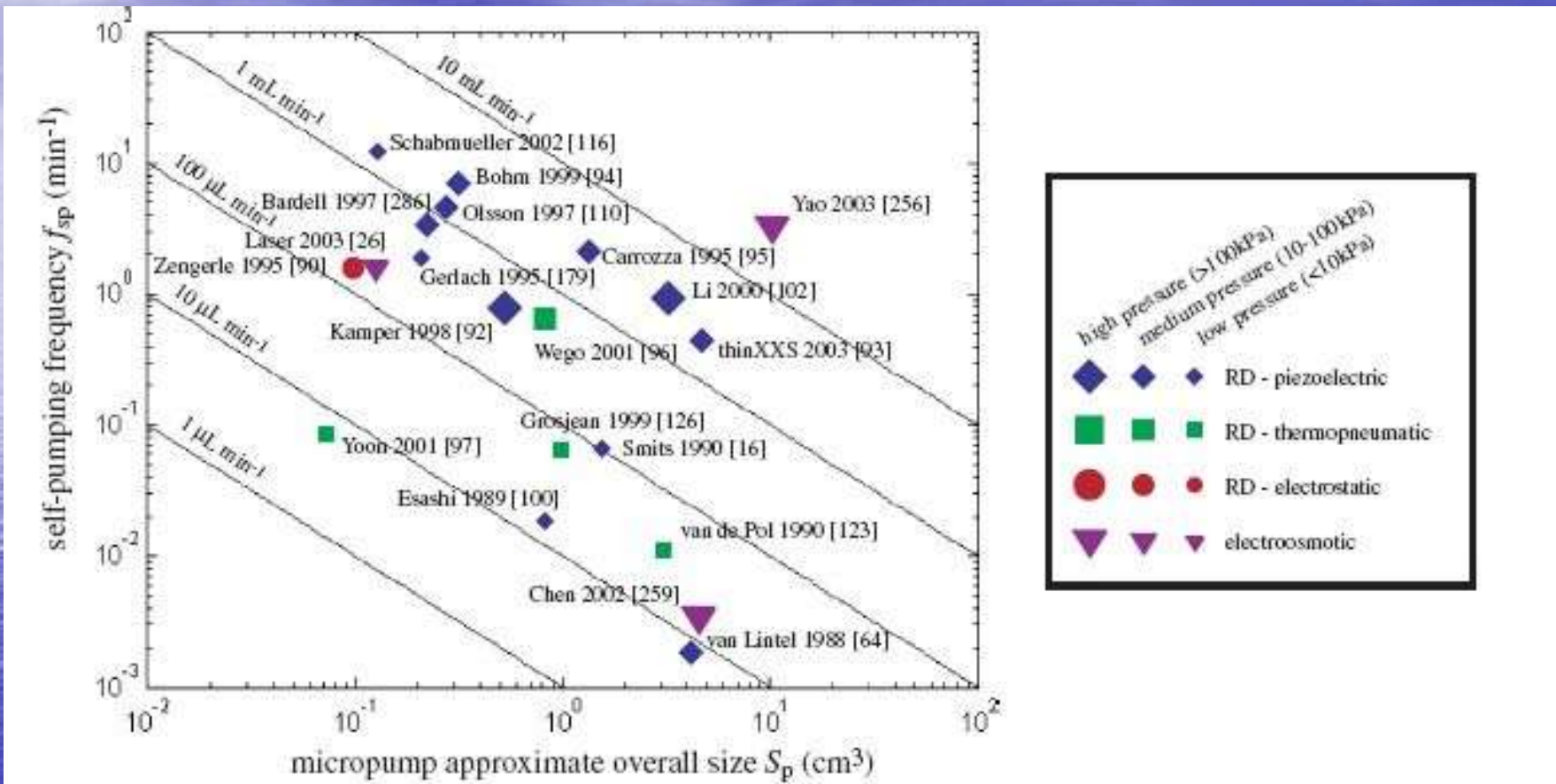
Operation of the diffuser-based pump: (a) Supply mode (b) Pump mode

Maximum flow rate per unit cross-sectional area at zero back pressure

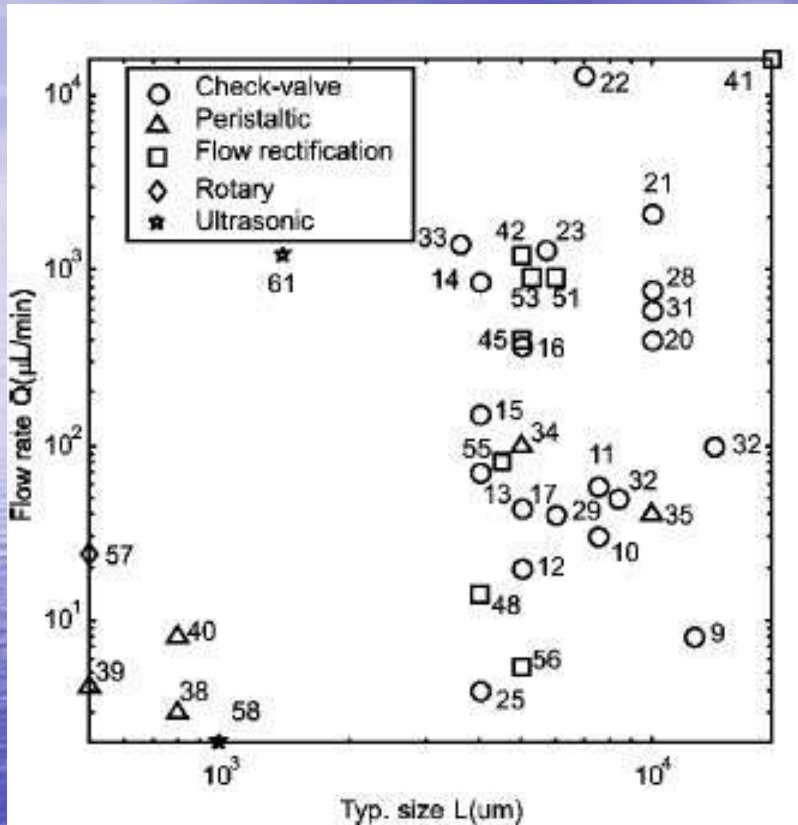


maximum back pressure at zero flow rate of various micropumps

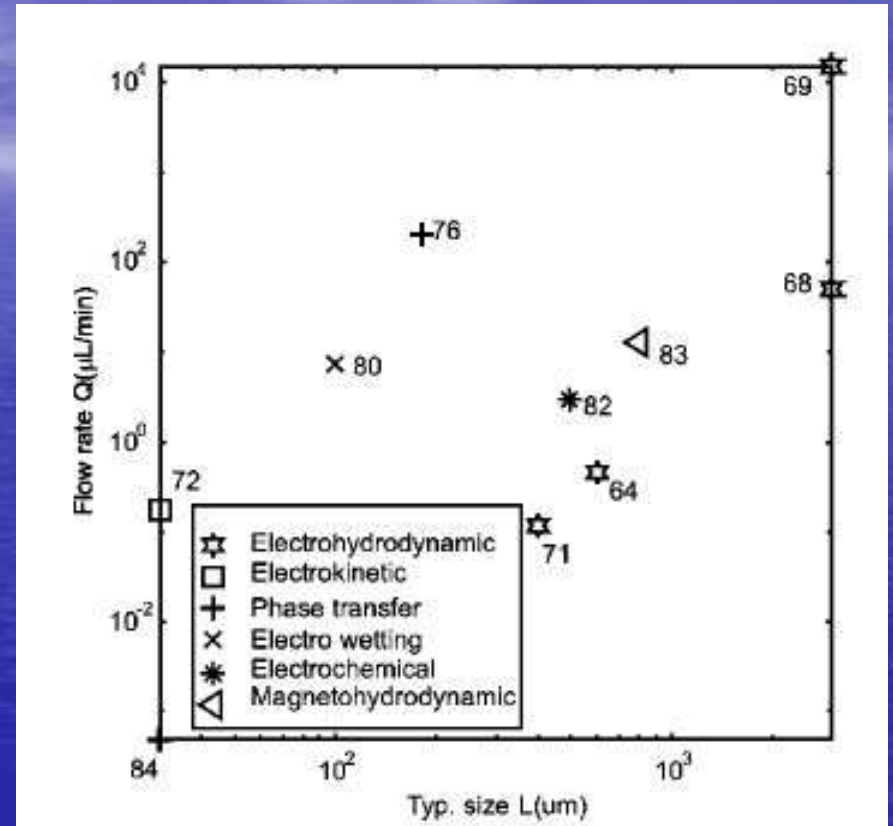




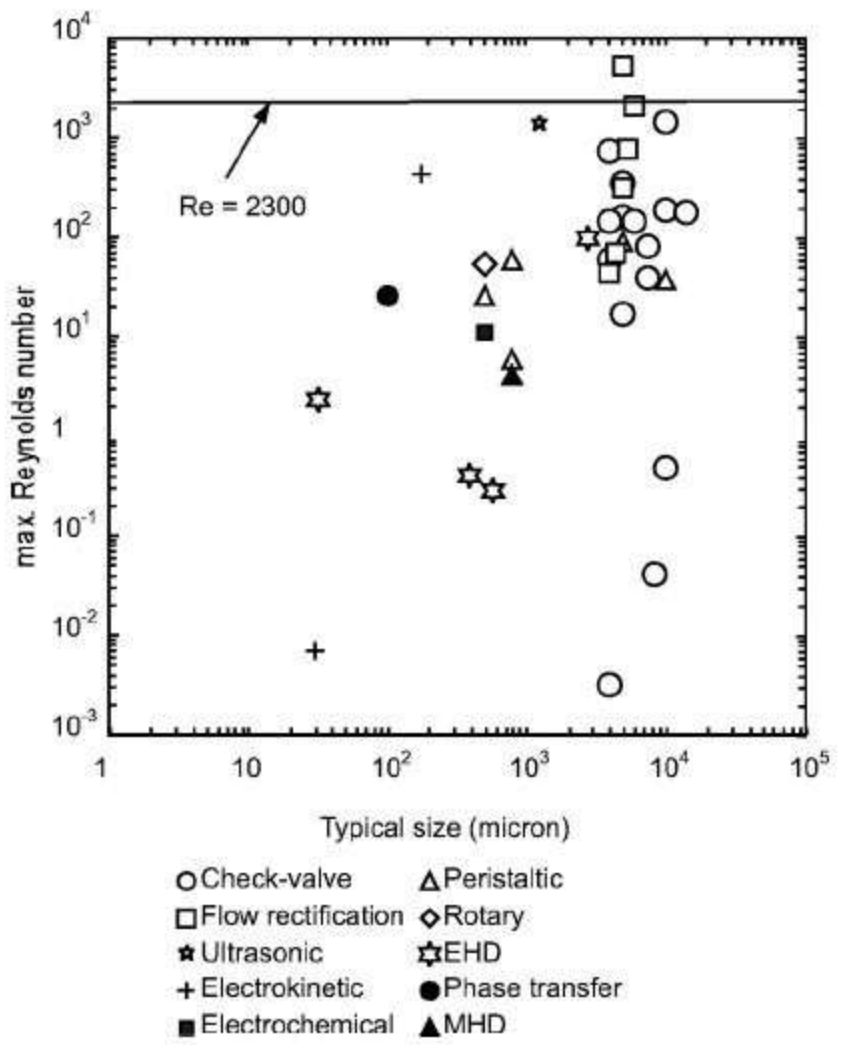
Comparison of several reported micropumps based on maximum flow rate, Q_{max} , maximum pressure p_{max} , and package size S_p . Self-pumping frequency is here defined as $f_{sp} = Q_{max}/S_p$.



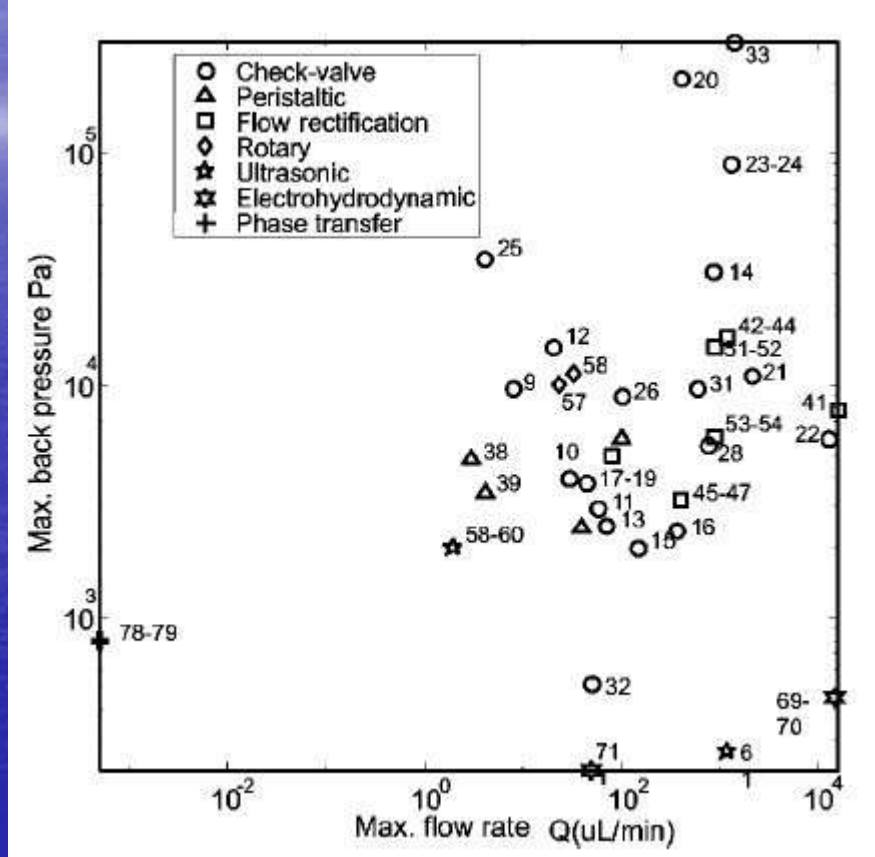
Flow rate versus typical size for mechanical pumps



Flow rate versus typical size for non-mechanical pumps



Maximum Reynolds number versus typical size



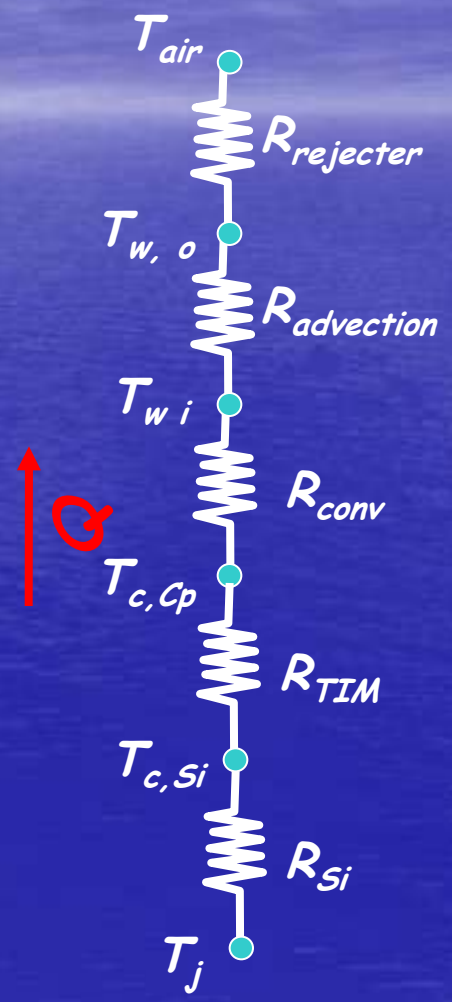
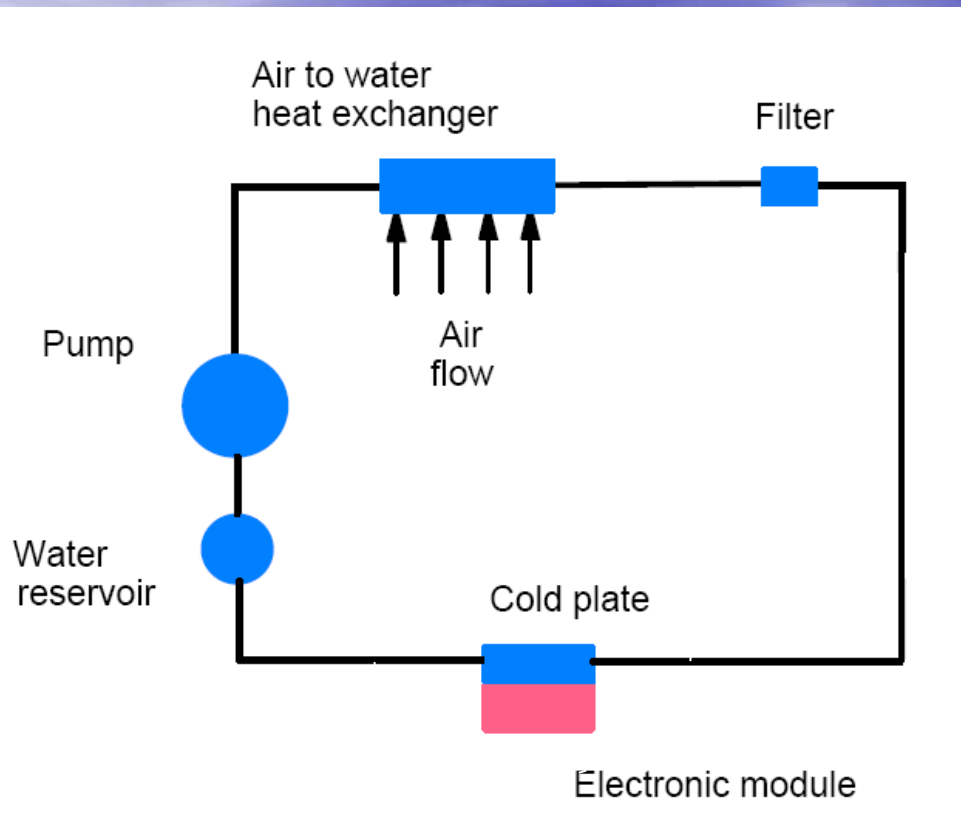
Maximum back pressure versus maximum flow rate



Summary of micropumps

- Reciprocating displacement micropumps, the most widely reported micropumps, have been produced with a wide variety of chamber configurations, valve types, drivers and constructions.
- Piezoelectrically driven reciprocating displacement micropumps have been the subject of particular attention and are now available commercially.
- A periodic displacement pumping based on localized phase change, electrowetting and other mechanisms are effective for transporting finite quantities of fluid in a generally unidirectional manner.
- Dynamic micropumps based on electromagnetic fields—electrohydrodynamic, electroosmotic and magnetohydrodynamic micropumps—are a subject of increasing interest.

A Typical Water-cooled System



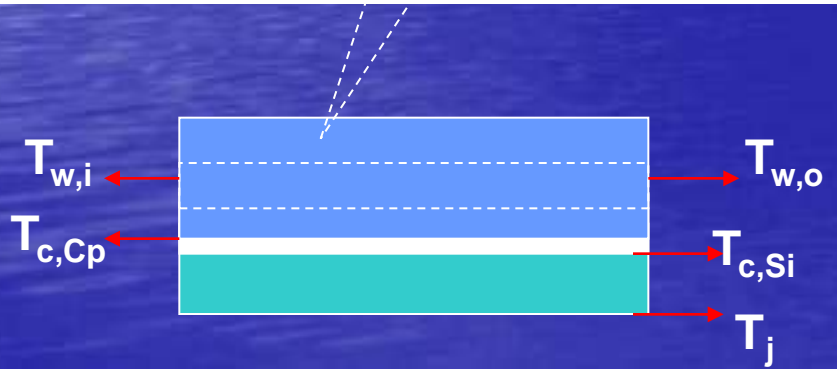
$R_{rejecter}$: Heat Transfer between water and HX

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$$R_{Si} = \frac{d}{kA}$$





Heat Transfer in Cold-plate



Fundamentals:

Heat Transfer Augmentation of Cold-Plate

- Heat Transfer Augmentation ($Q = UA\Delta T_m$)

Q : heat transfer rate,

U: overall heat transfer Coeff.

A: Area

ΔT_m : mean temperature difference

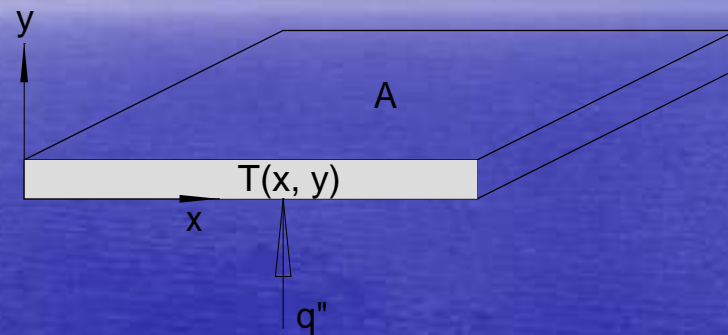
- Increase A
- Increase U
- Reduce ΔP at fixed Q



Fundamentals:

□ **Conduction** $q = -kA \frac{dT}{dy}$

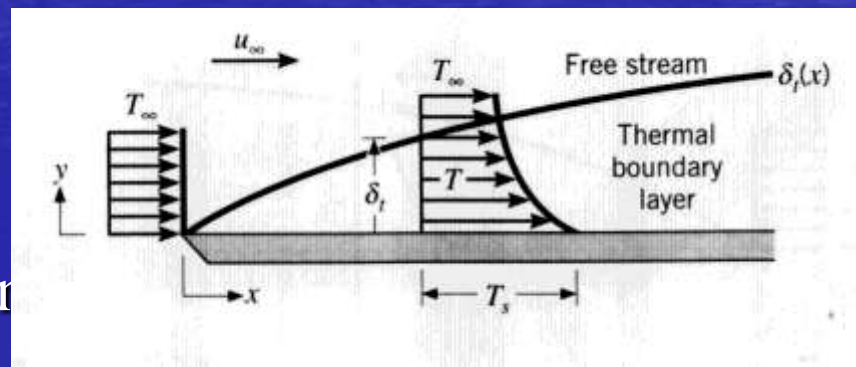
- Increase A
- Use high k materials
- Increase temperature gradient?
 $dT/dy = (T_2 - T_1)/(y_2 - y_1)$



□ **Convection**

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

- Increase A
- Increase h (convective heat transfer coeff.)





Ways of Enhancements

- Increase h without an appreciable area (A) increase.
- Increase A without appreciably changing h .
- Increase of both h and A .

In essence, increase the effective hA .

However, one should notice the constraint of pumping power.



Performance evaluation criteria of enhanced surfaces

- Reduced Heat Transfer area for fixed pumping power
- Reduced pumping power at fixed heat duty
- Reduced Effective temperature difference at fixed heat duty and pumping power
- Increased Heat Transfer Duty

Note: Be aware of the constraints of enhancements

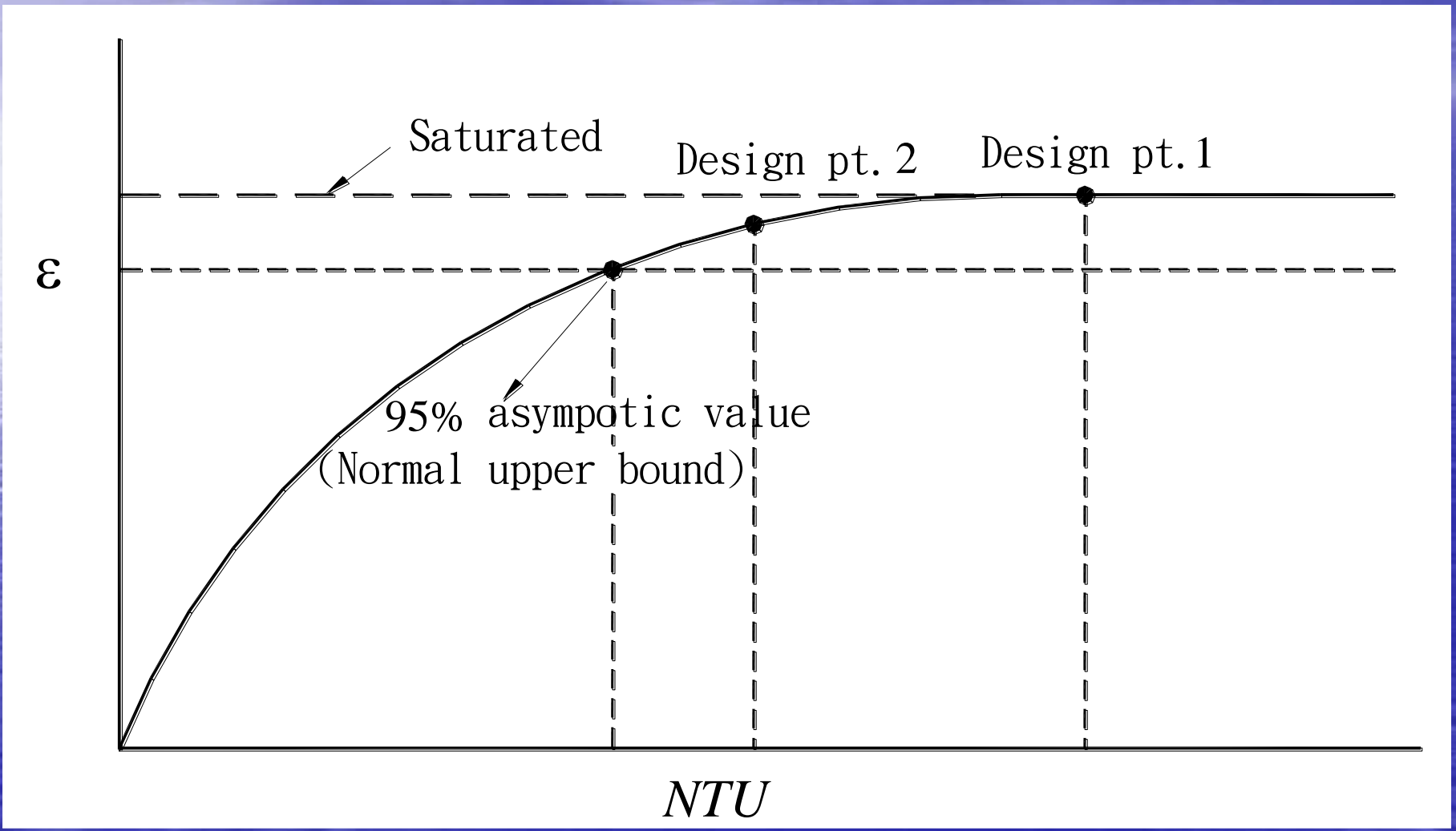
For further details, check the details of Webb & Kim's book.



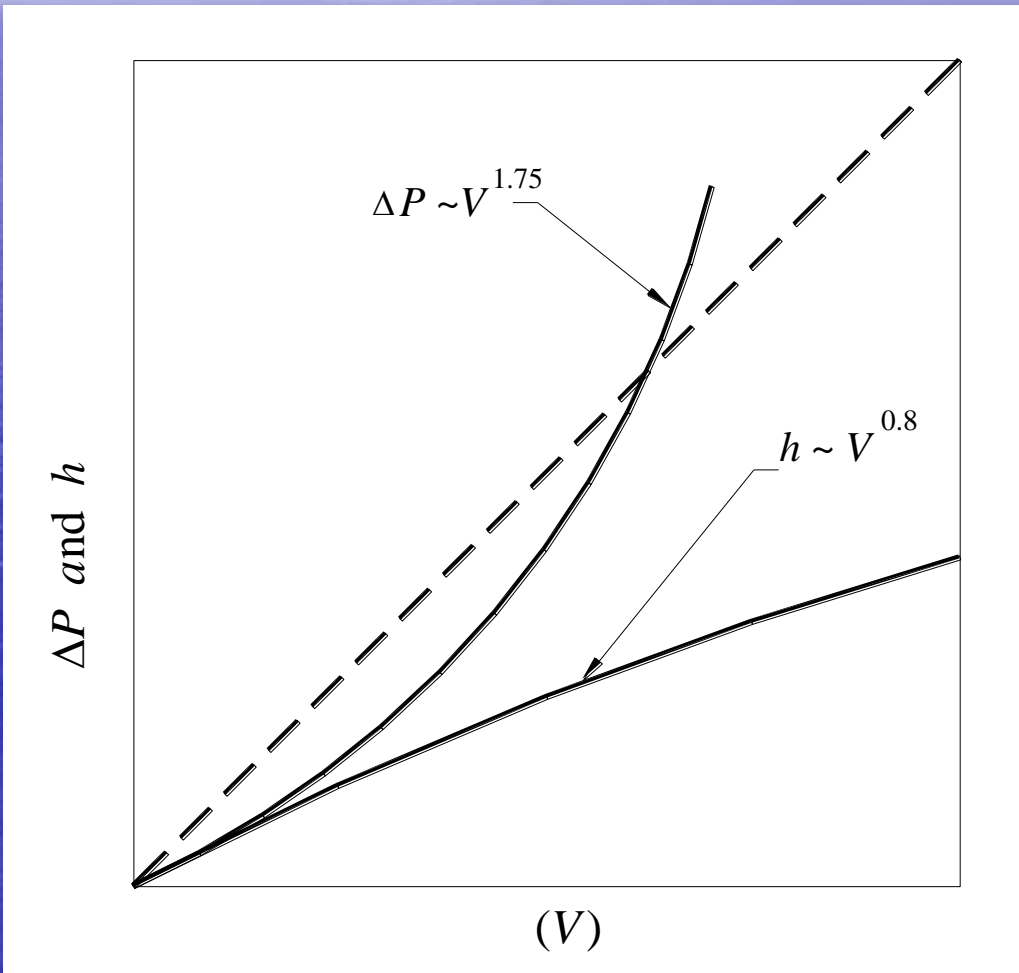
Think Before Jump to the enhancements

- Do we really need enhancement?
- When do we need the enhancement?
- Where should we place enhancements?
- Is it cost-effective?

- Why enhancement? Do we really need enhancement?



Taking into consideration about the influence of pressure drop (pumping power)

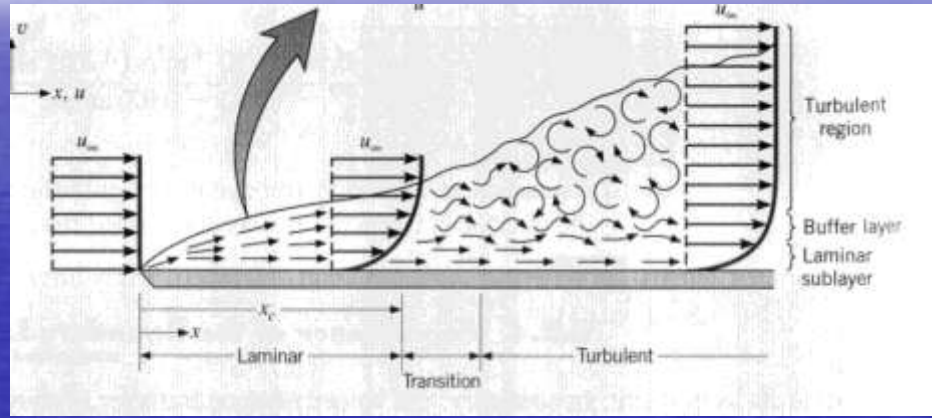


Enhancements depends on flow pattern

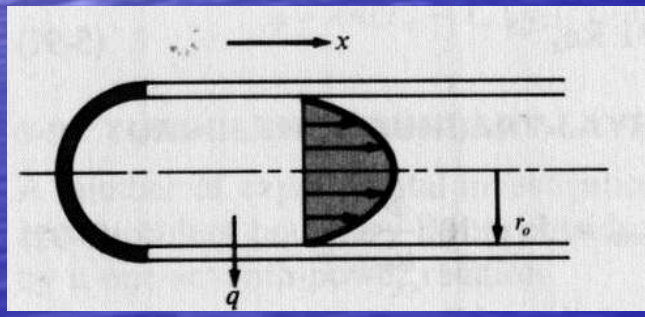
$$q = hA(T_s - T_\infty) = -kA \left. \frac{dT}{dy} \right|_{y=0}$$

□ $Re_D = \rho u D / \mu$

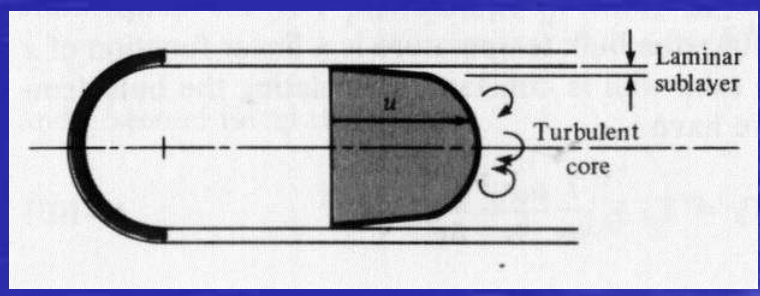
For smaller channel flow, the flow pattern is mostly operated at laminar flow



$Re_D < 2,300$ laminar flow



$Re_D > 2,300$ turbulent flow



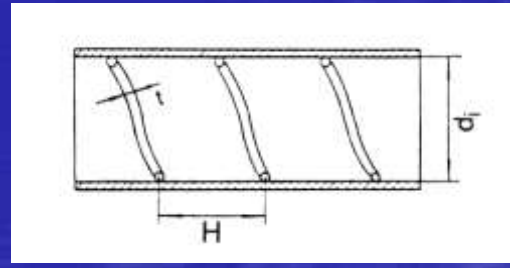
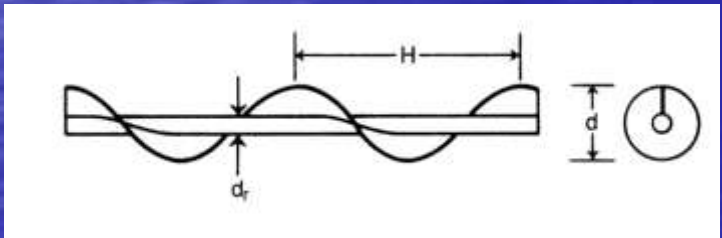
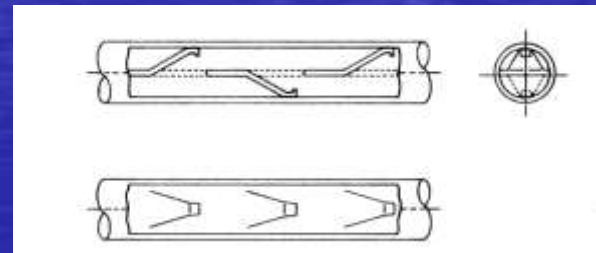
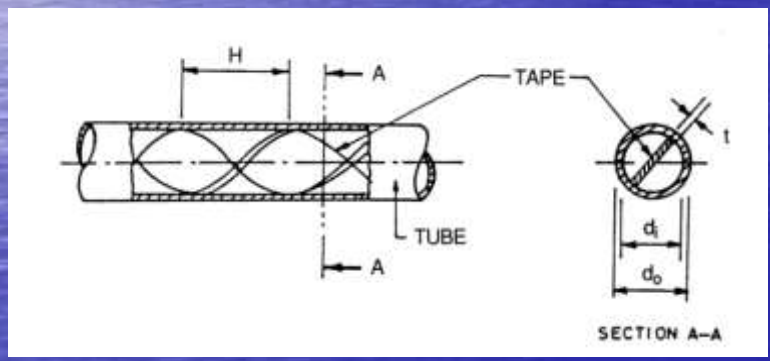
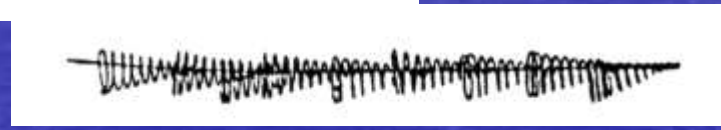
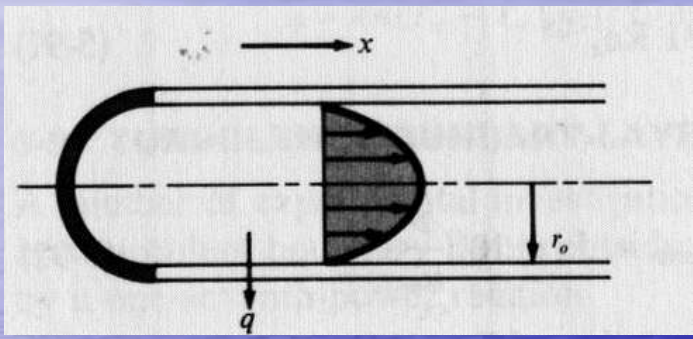


Common ways of external cooling

- Forced air cooling
- Forced liquid cooling (water or dielectric fluid)
- Forced convective boiling cooling
- Immersion cooling
- Thermoelectrics
- Refrigeration

Among them, smaller and smaller dimension are employed with heat flux being increased.

Augmentation of single-phase flow – Laminar Flow



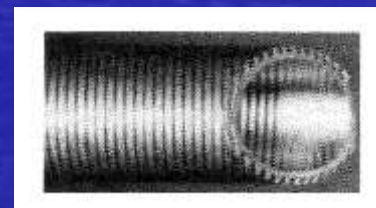
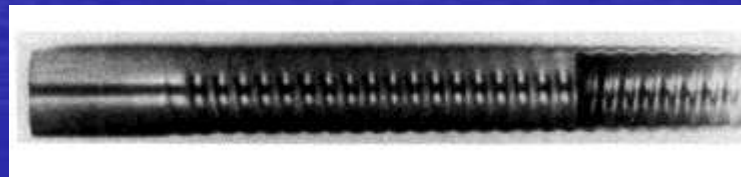
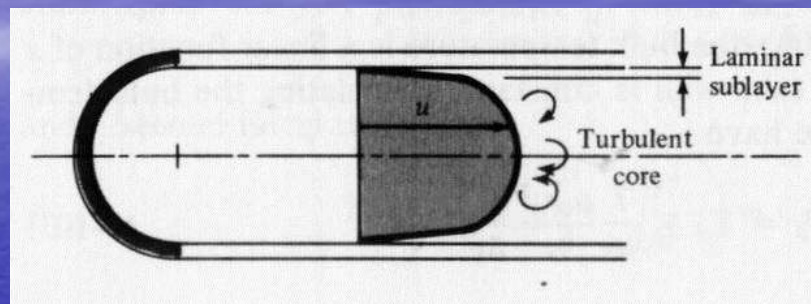
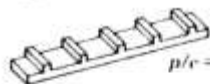
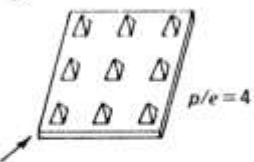
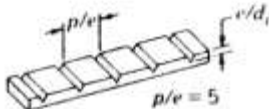
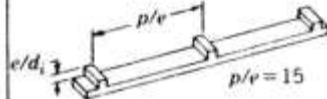
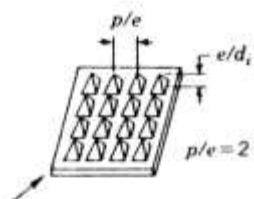
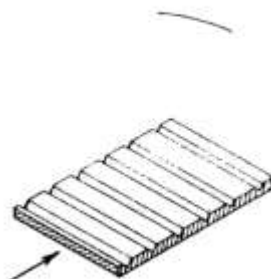
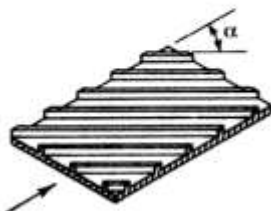


Augmentation in turbulent flow

Three dimensional roughness ("uniform roughness")

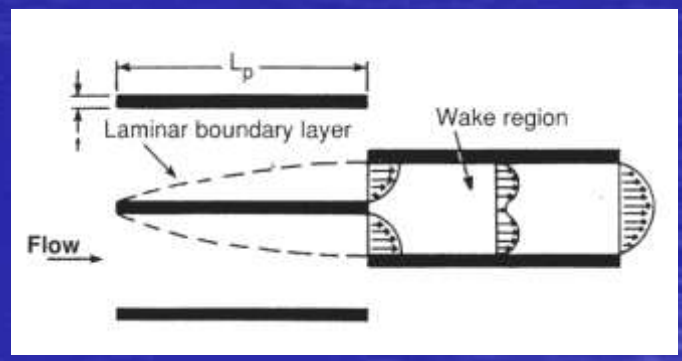
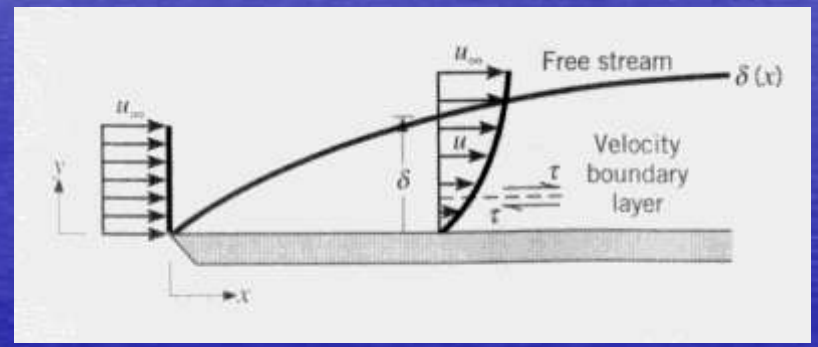
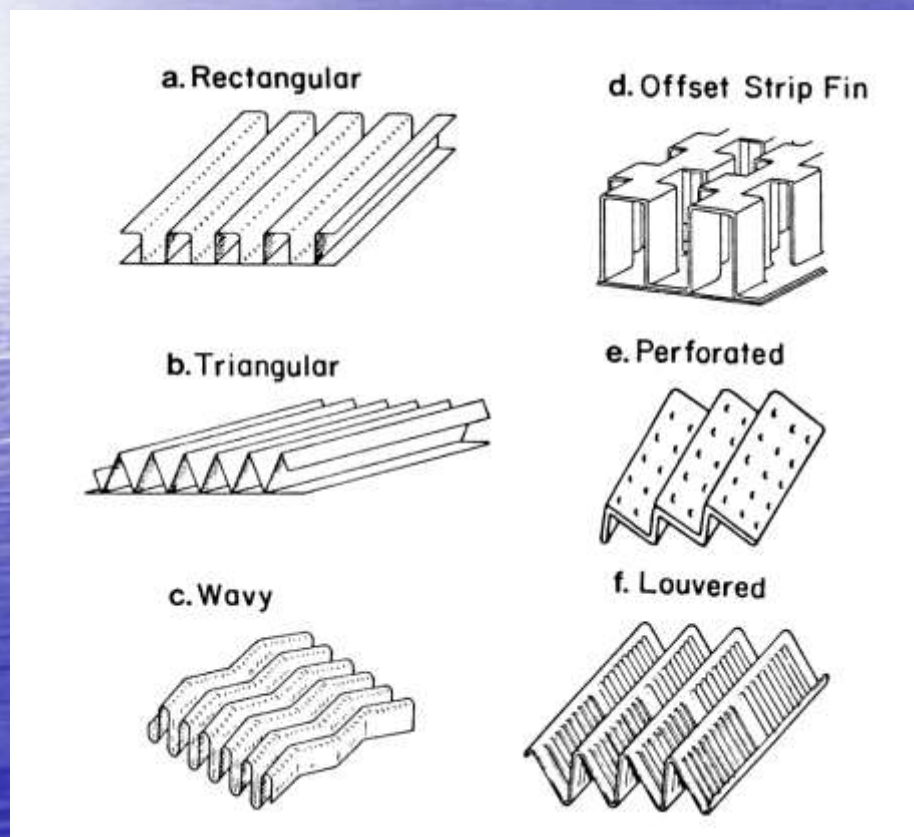
Ridge type two-dimensional roughness ("repeated ribs")

Groove type two-dimensional roughness



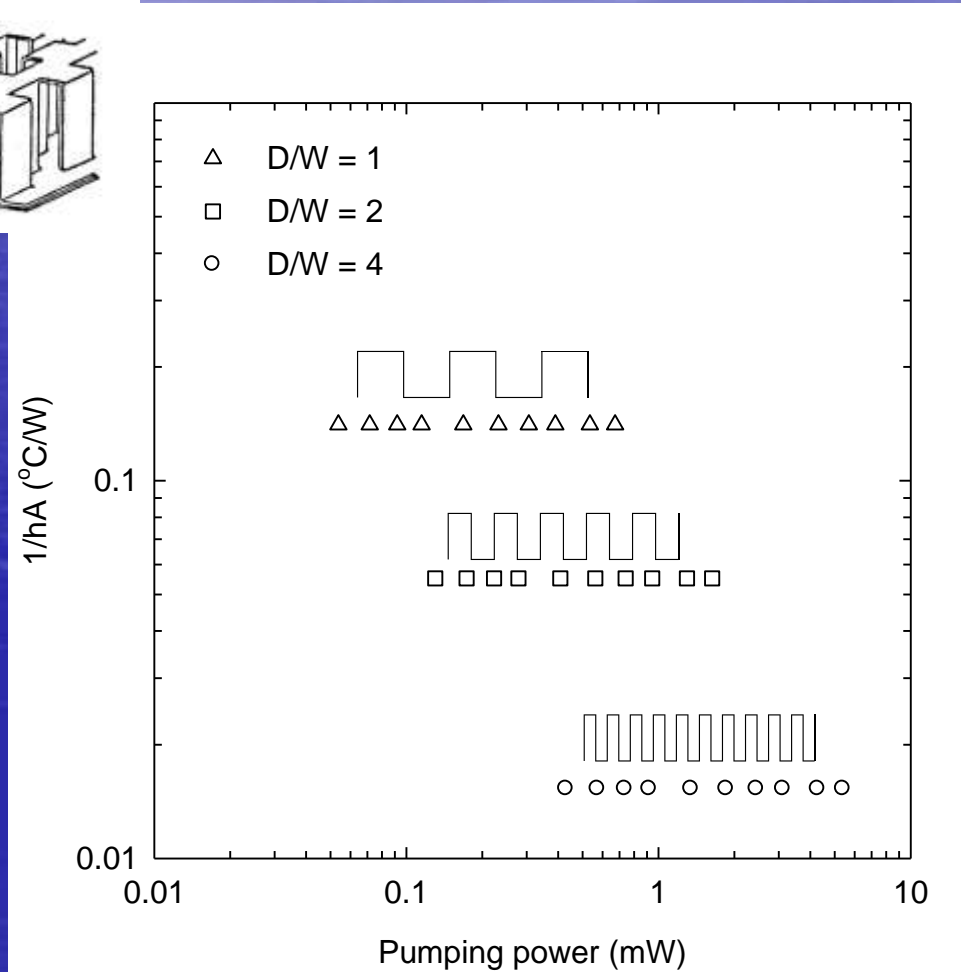
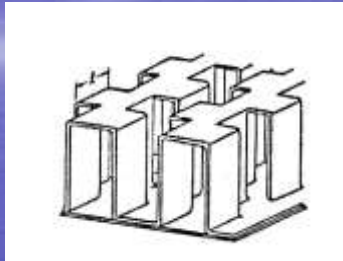
Augmentation – with the presence of fins

- ❑ 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

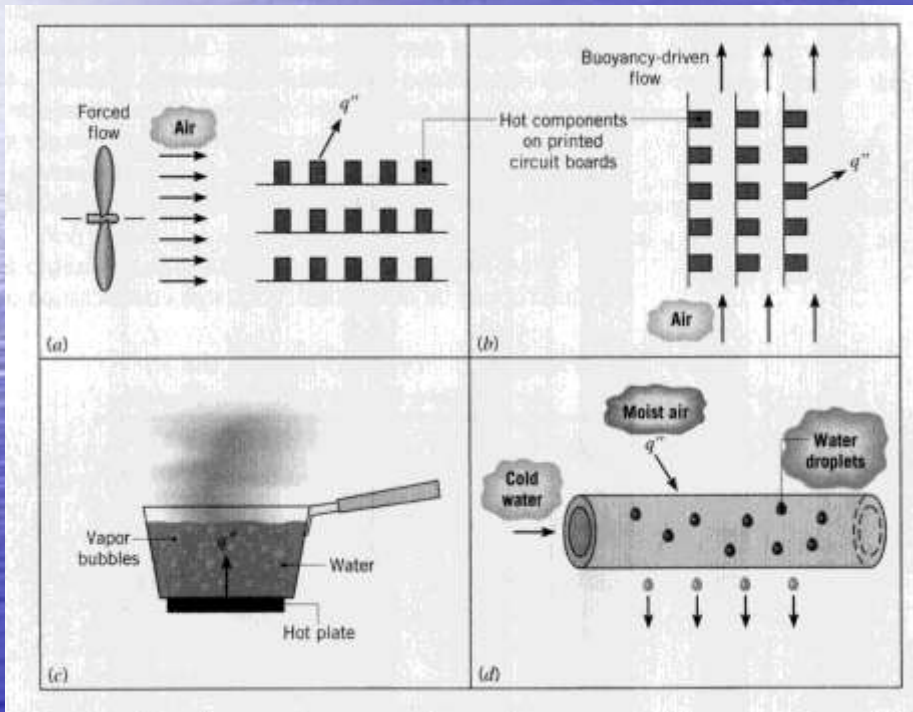


TABLE 1.1 Typical values of the convection heat transfer coefficient

Process	h (W/m ² · 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

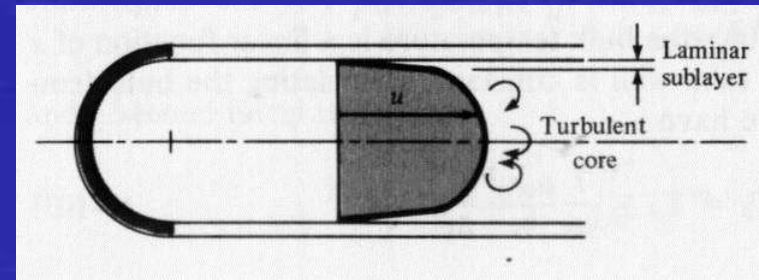
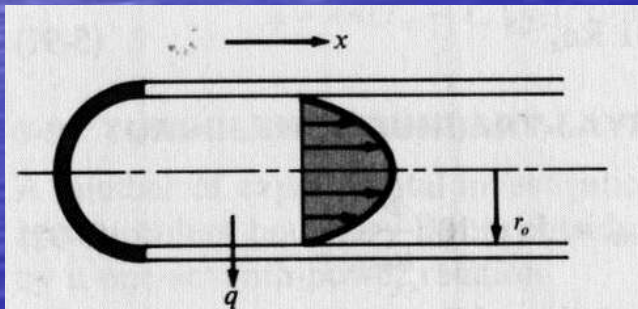
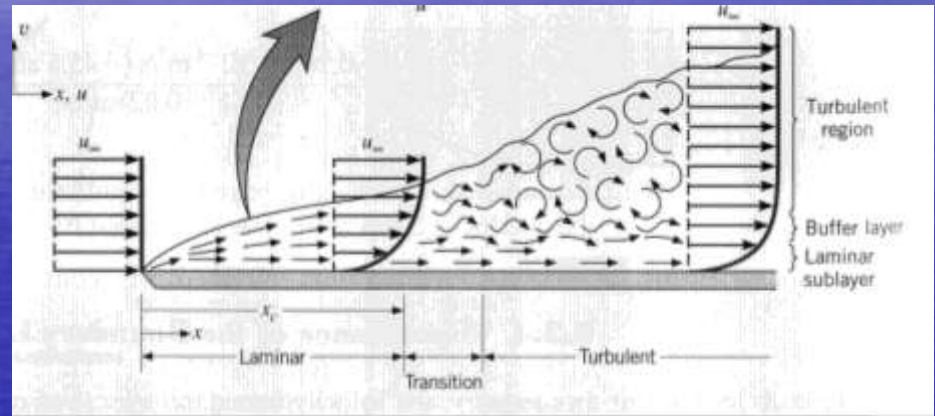
$$q = hA(T_s - T_\infty) = -kA \frac{dT}{dy} \Big|_{y=0}$$

□ $Re_D = \rho u D / \mu$

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

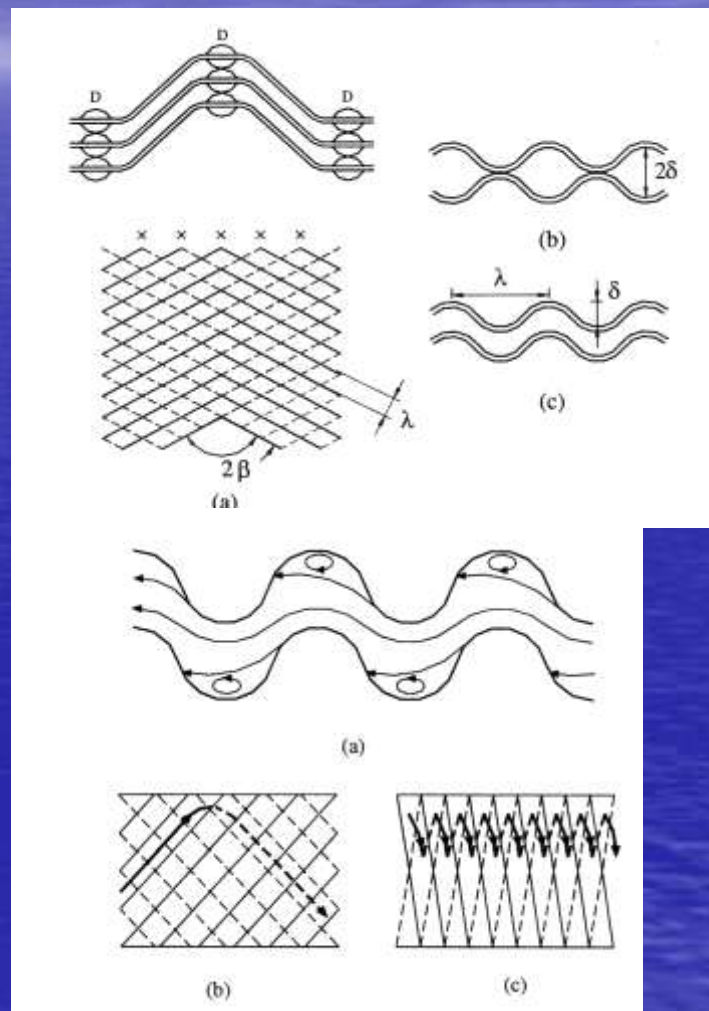
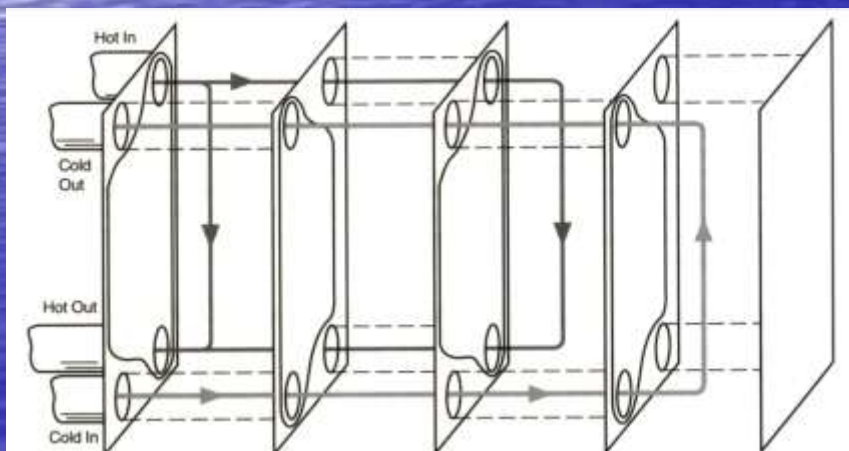
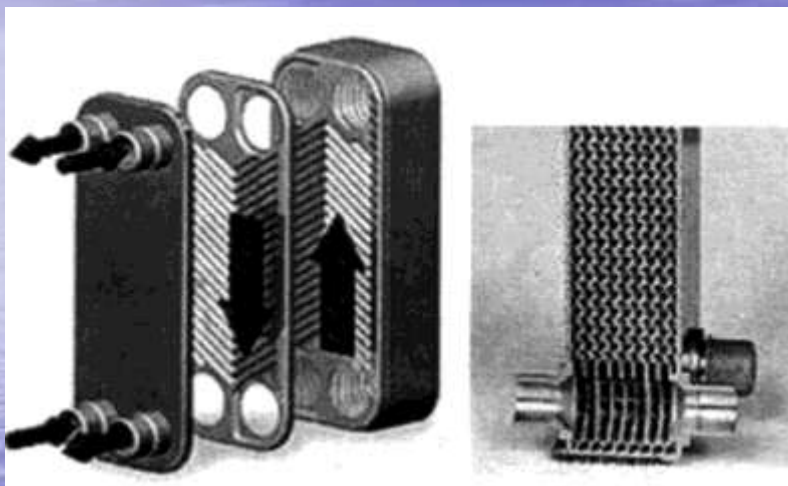
$Re_D < 2,300$ laminar flow

$Re_D > 2,300$ turbulent flow





Apply the conventional Plate HX



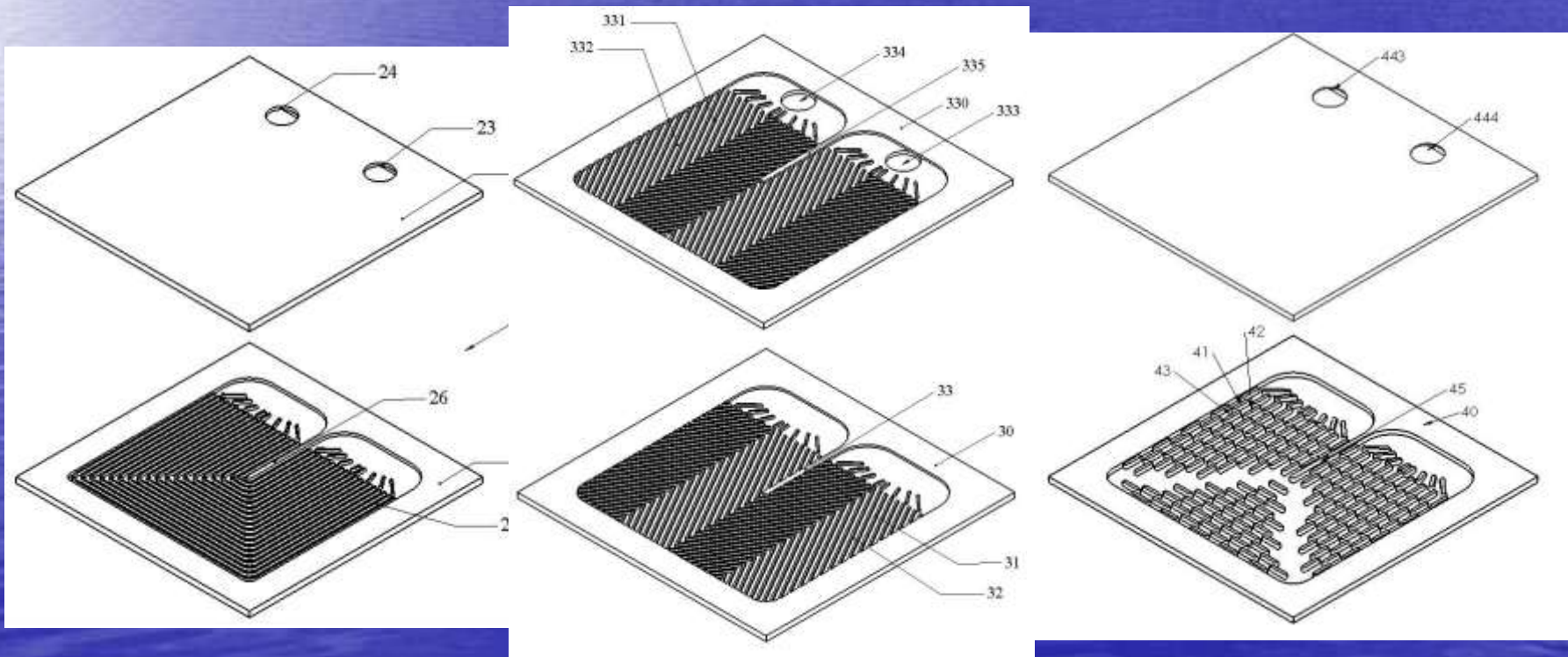
Augmentation based on plate HXs

dimension: 50 mm x 50 mm x 2 mm

U - type

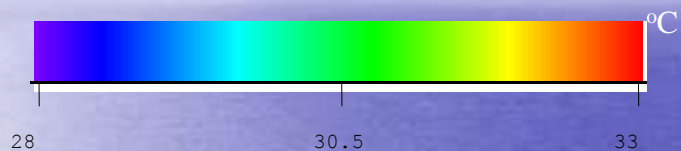
Chevron (V)

(OSF)

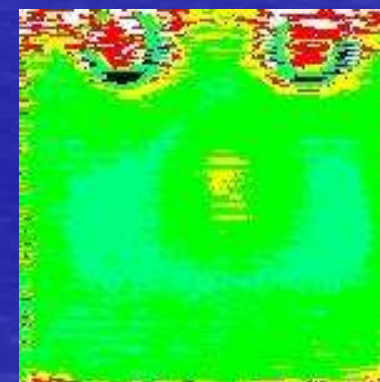




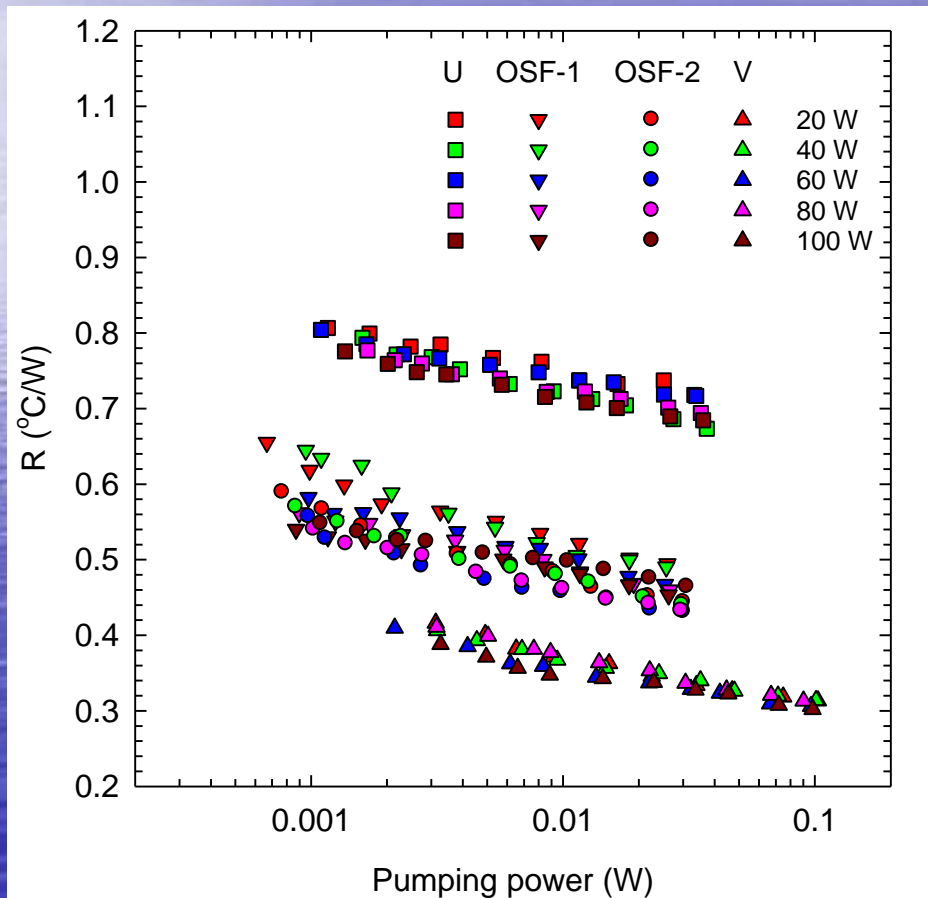
Temperature Distribution ..



Power	Flowrate (mL/min)	Inlet Temp.
20	230	30 $^{\circ}\text{C}$



A comparison of Thermal Resistance vs. pumping power

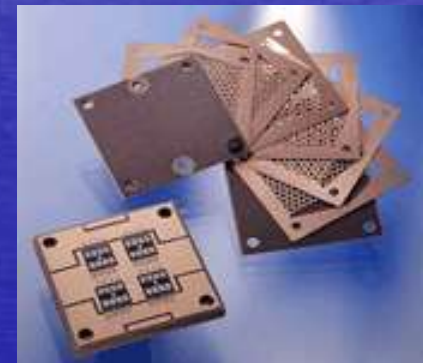
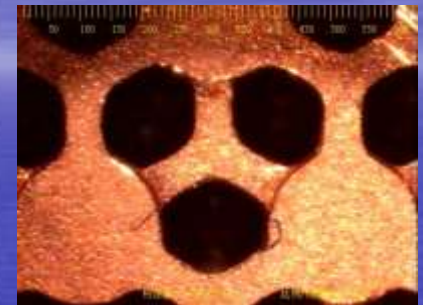
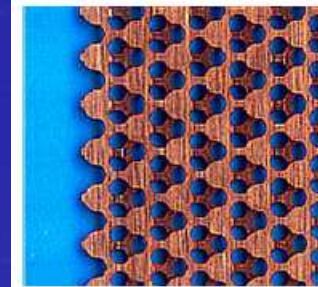
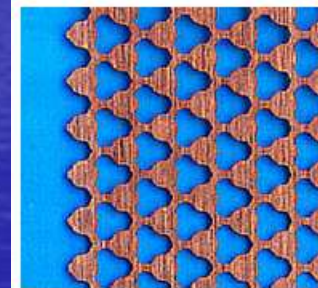
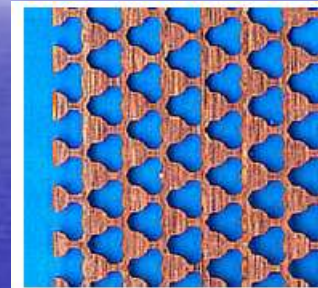
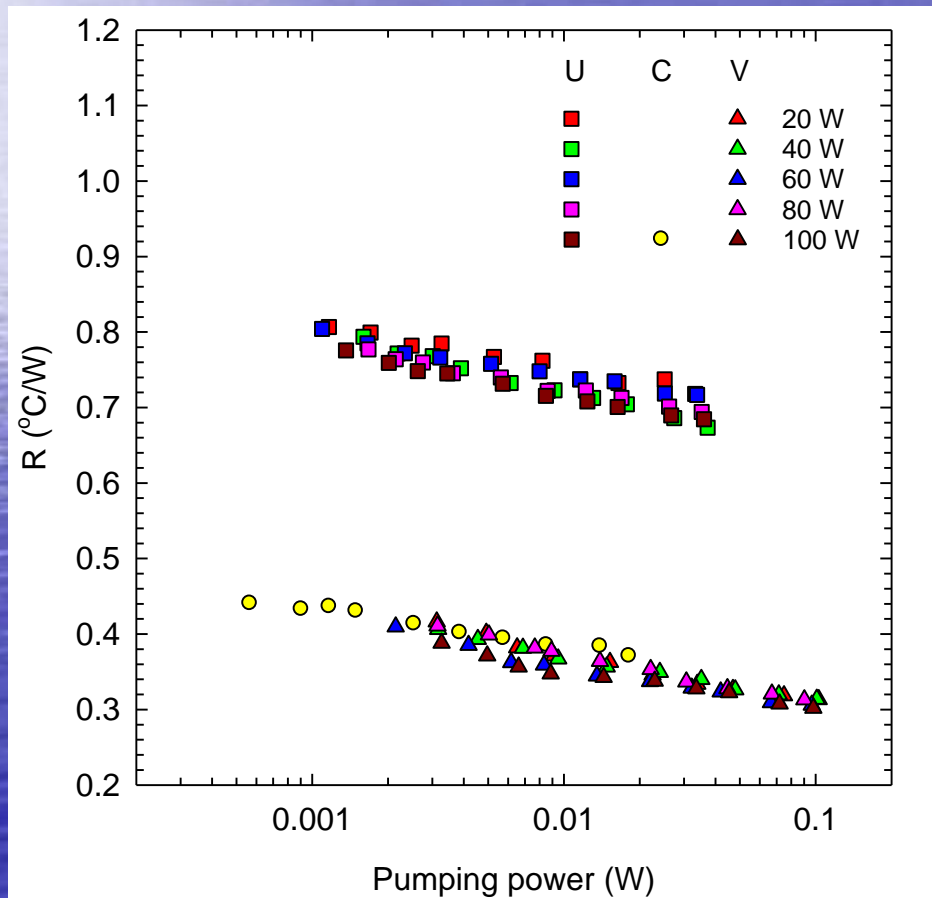


$$R = \frac{T_c - \frac{1}{2}(T_{wi} + T_{wo})}{q}$$

A comparison with some existing commercial products

Dimension: U, V: 50 x 50 x 2 mm³

C: 40 x 40 x 4 mm³

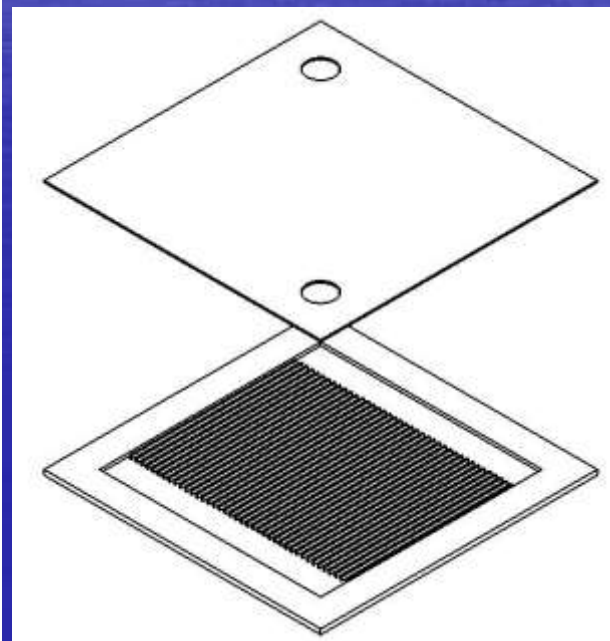
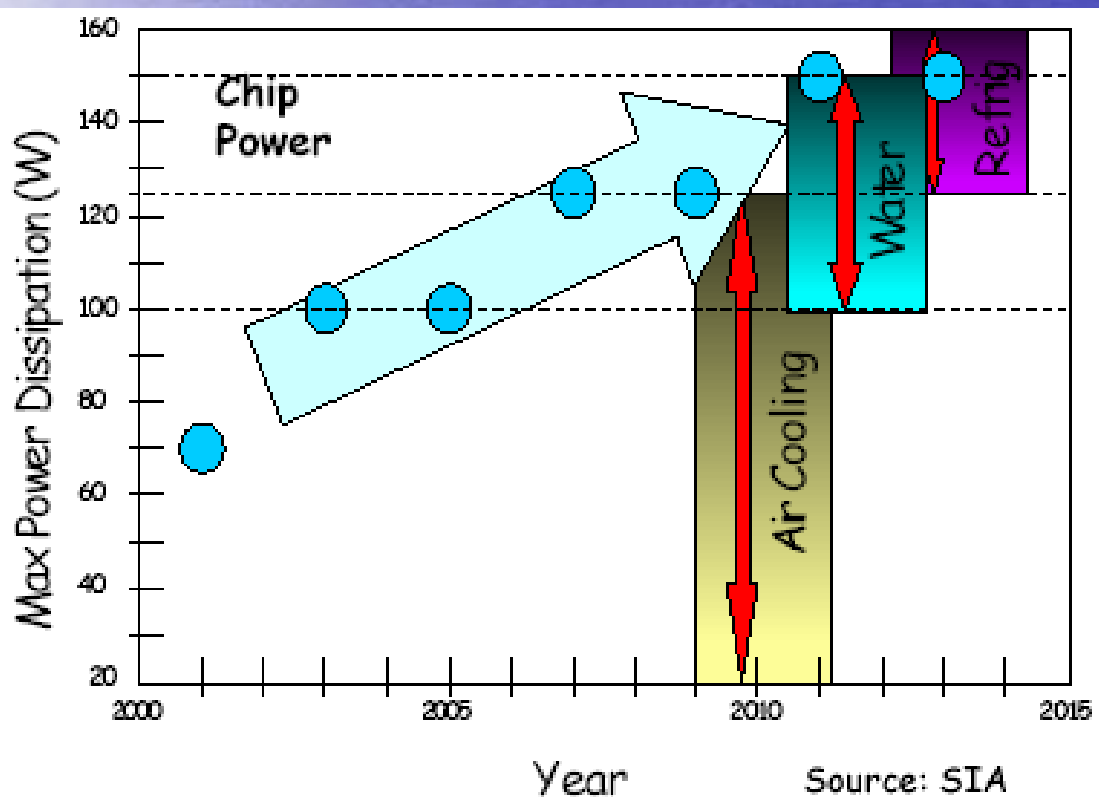




Why Microchannel?

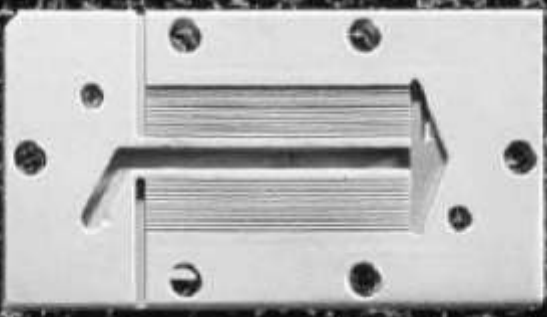
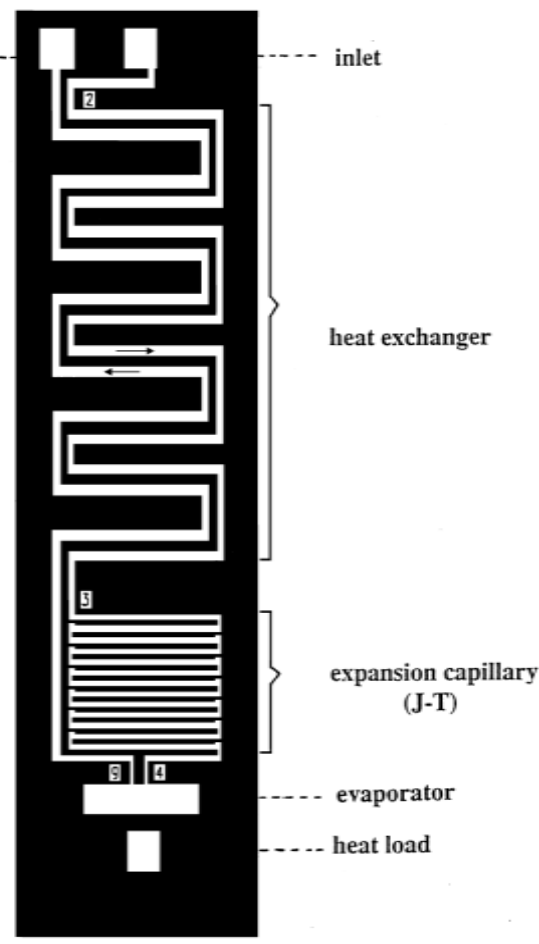
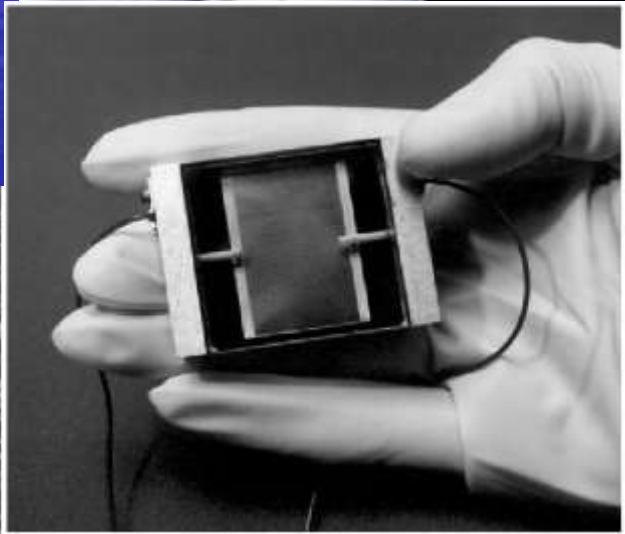
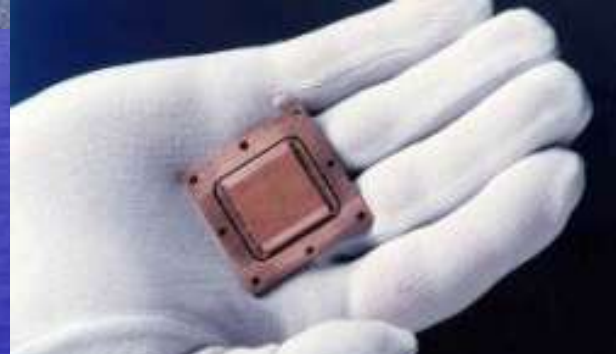
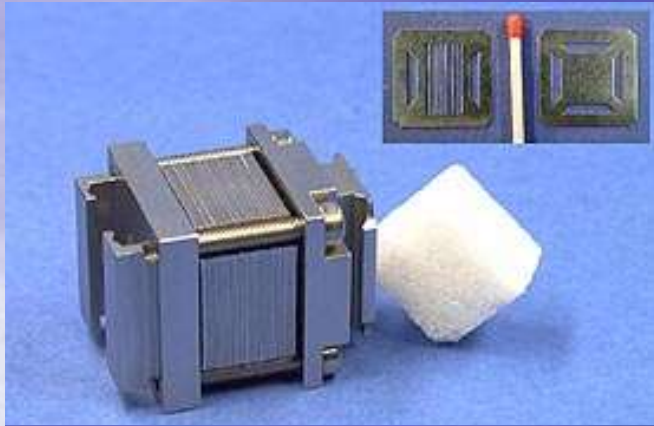


Water-cooled Micro-channel HXs Usually with micro-channels





Micro-channel HXs - Example



Why Use smaller Channel?

Consider a reduction of D_h with a factor of 10

- In this case, 10 times increase of heat transfer coefficient ($Nu = hD_h/k$)
- In this case, 10 times increase of heat surface area at the same volume)
- For the same thermal resistance, flow rate, and pressure drop constraints, the microchannel design offers 1/100 Less volume!

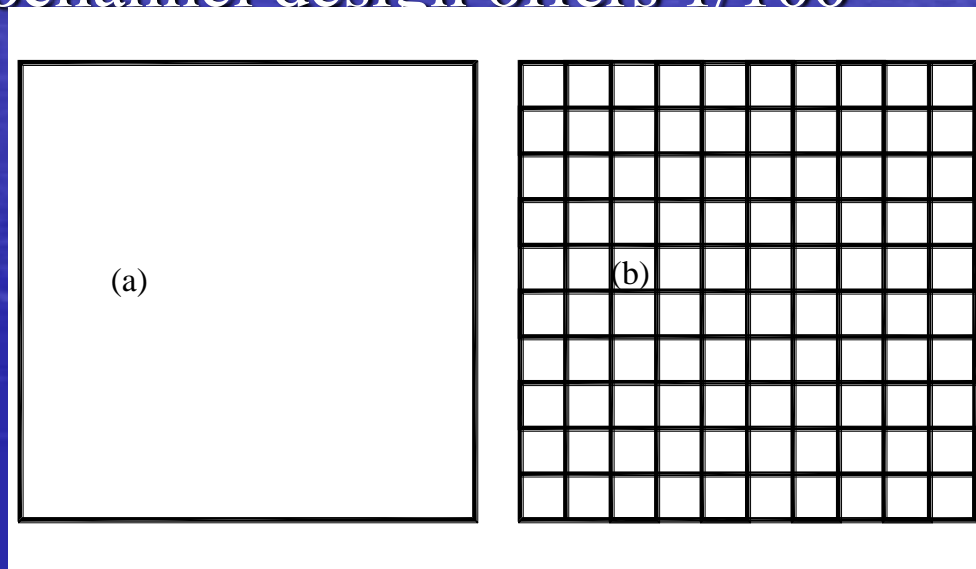
$$\Delta P = \frac{4L}{D_h} f \frac{1}{2} \rho u^2 = \frac{4L}{D_h} f \frac{\rho u D_h}{\mu} \frac{\mu}{D_h} \frac{1}{2} u$$

$$= \frac{2\mu u L}{D_h^2} f Re$$

$$\Delta P \propto \frac{L}{D_h^2} \rightarrow L \text{ must reduce to } \frac{1}{100} \text{ to}$$

maintain a constant ΔP , while the thermal

resistance $\frac{1}{hA}$ stays the same! This implies a reduction of volume to its original $\frac{1}{100}$



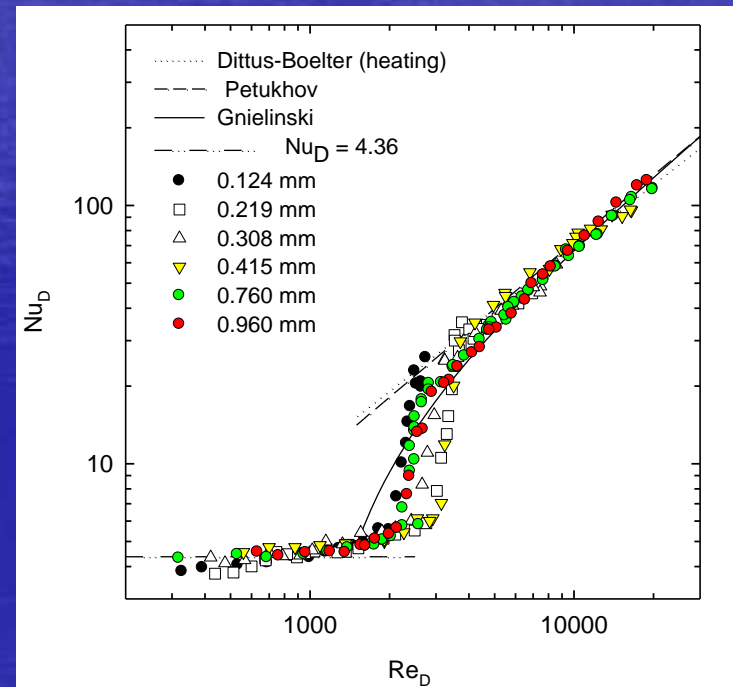
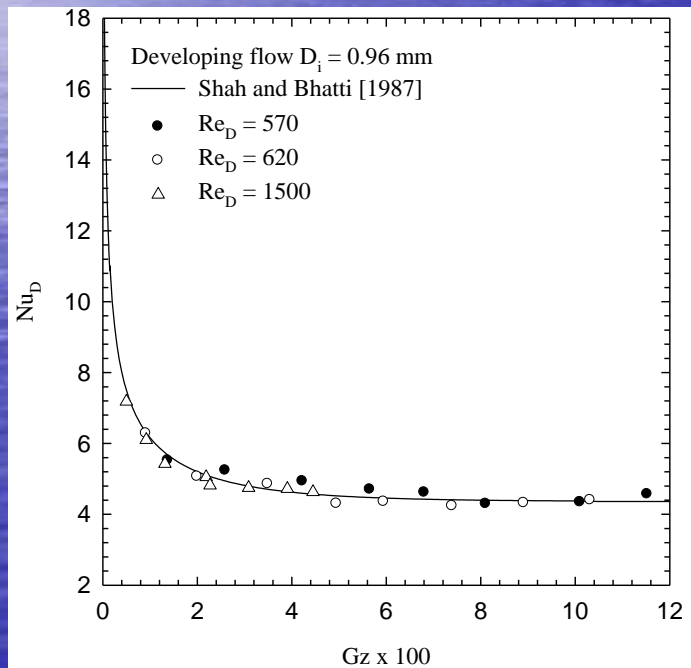
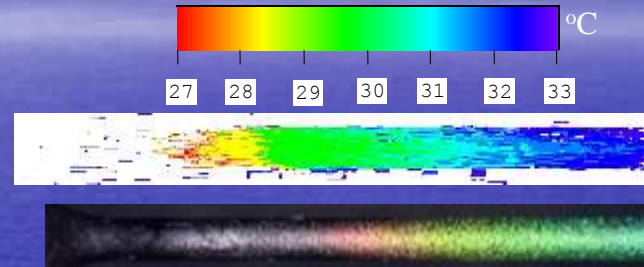


What's problem in designing Microchannels?

- Correct dimension (most crucial!), correct flow rate
- Effect of entrance
 - Hydraulic entrance length
 - Thermal entrance length
 - Simultaneously entrance effect
- Mostly Laminar flow
 - Depends on boundary conditions

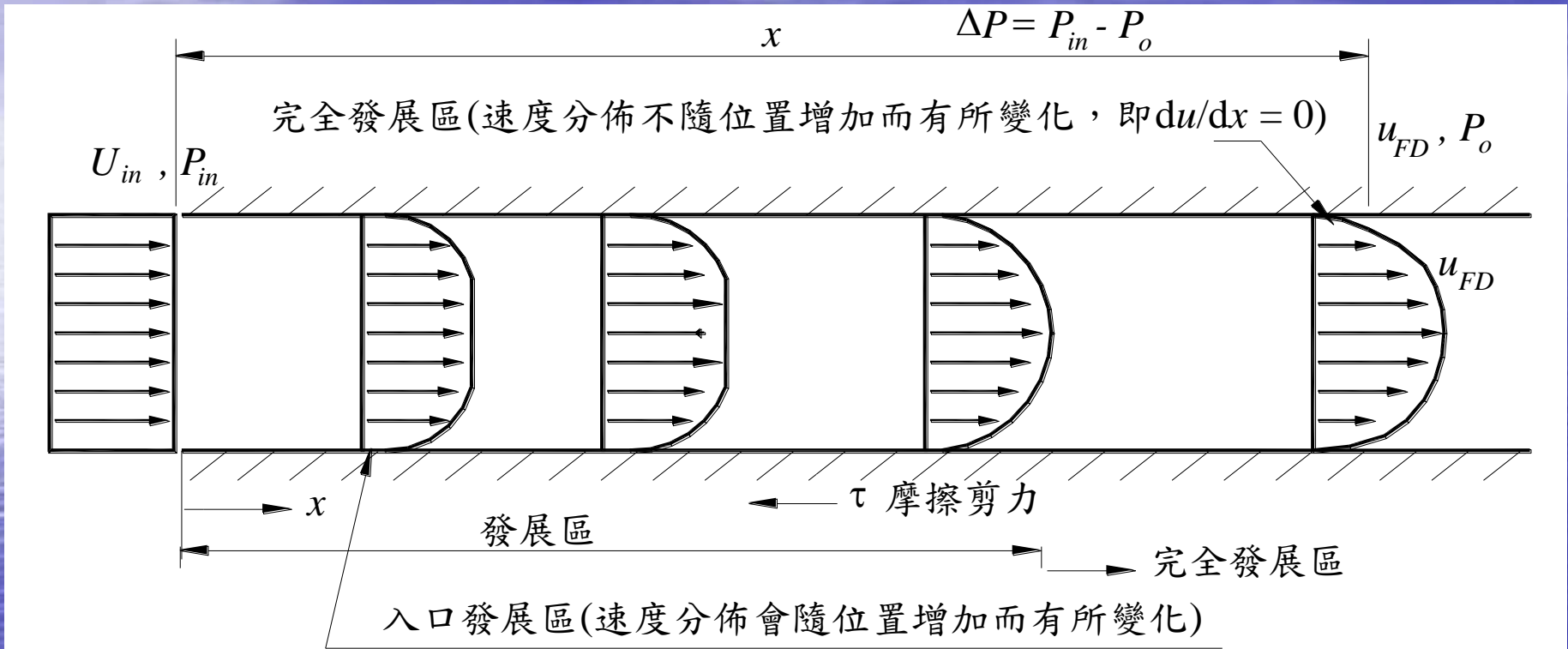
Does the heat/flow characteristics in micro channel behaves like macro channel?

- ❑ $Nu_D (= hD/k)$
- ❑ For single-phase fluid in the range of 0.1 to 1.0 mm, heat transfer behaves just like macro-channels



Effect of hydraulic entrance

$$L_{hy}^+ = \frac{L_{hy}}{D_h Re}$$



$L_{hy}^+ \approx 0.05$ (有些文獻建議 0.06) (層流流動)

$L_{hy}^+ \approx 4.4 Re^{-\frac{5}{6}}$ (紊流流動 → 資料來源: www.engineeringtoolbox.com)



Example

- For a laminar flow $Re = 1000$, and a turbulent flow condition with $Re = 10000$.

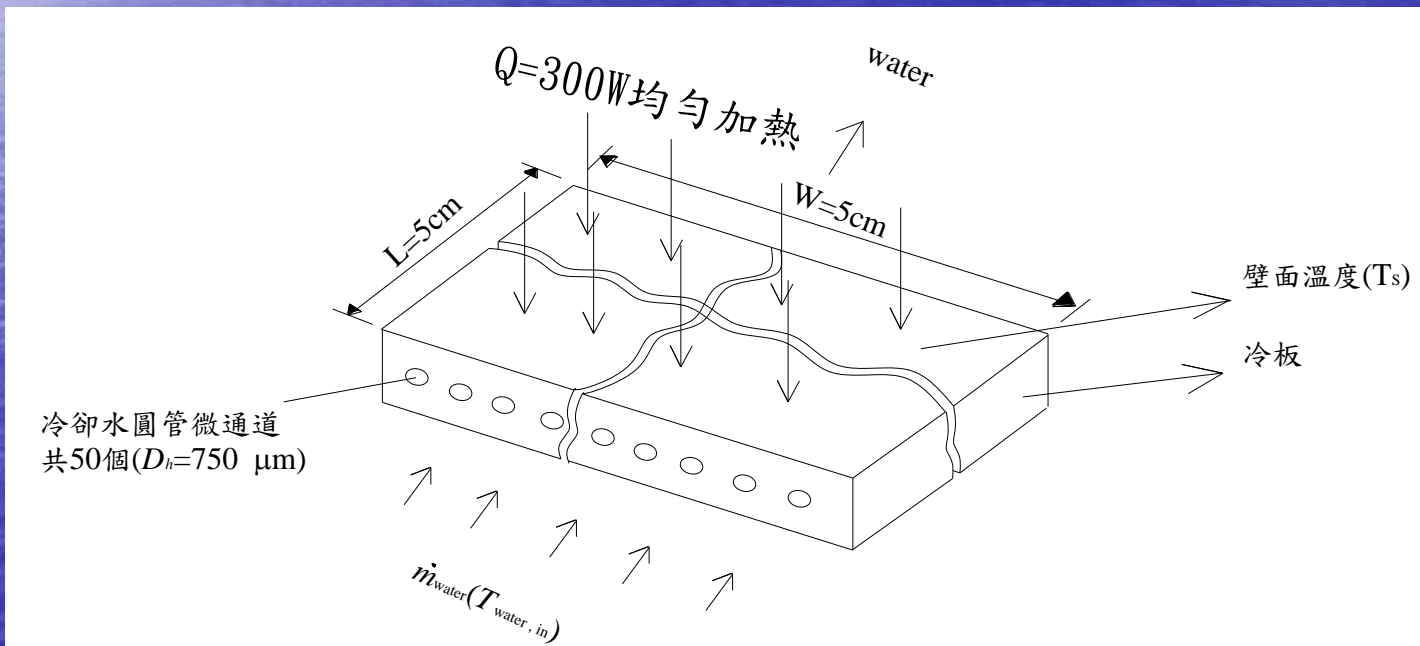
Then

$$L_{hy,lmminar} \approx 0.05(\text{或}0.06) \times 1000 \times D_h = 50 \sim 60 D_h$$

$$L_{hy,turbulent} \approx 4.4 \times 10000^{1/6} \times D_h = 20.5 D_h$$



例：考慮一圓形微通道熱交換器，微通道圓管水冷式冷板如圖1-11所示，圓管直徑為 $750\ \mu\text{m}$ ，共有50個通道，冷板貼於一均勻發熱之熱源，其發熱量為 $400\ \text{W}$ ，設計上使用入水溫度為 34.3°C ，此溫度下相關的物性為 $\rho_{\text{water}} = 994.27\ \text{kg}/\text{m}^3$ ， $\mu_{\text{water}} = 0.000743525\ \text{Pa}\cdot\text{s}$ ， $c_{p,\text{water}} = 4178\ \text{J}/\text{kg}\cdot\text{K}$ ， $k_{\text{water}} = 0.6235\ \text{W}/\text{m}\cdot\text{K}$ ， $\text{Pr}_{\text{water}} = 5.0$ ，需要將發熱表面的平均溫度控制在 50°C 以下，試問設計水流量應至少要多大？



圓管微通道熱交換冷板示意圖



例：考慮一平行通道微通道，平行通道間距為 $500\ \mu\text{m}$ ，長度為 $5\ \text{公分}$ ，入口的雷諾數為 1000 ，試估算通過平行通道的無因次壓降 ΔP^* 。

首先計算平行板通道的水力直徑 D_h ，由其定義 $D_h = 4A_c/P_h$ 可知 D_h 能等於兩倍的平行板通道間距 $= 2 \times 500\ \mu\text{m} = 1\ \text{mm} = 0.001\ \text{m}$ 。由式1-63，

$$\frac{L_{hy}}{D_h} = \frac{0.315}{0.0175\text{Re}+1} + 0.011\text{Re} = \frac{0.315}{0.0175 \times 1000 + 1} + 0.011 \times 1000 = 11.02$$

$$\therefore L_{hy} = 11.02 \times D_h = 11.02 \times 0.001 = 0.01102\ \text{m}$$

也就是說入口長度約佔總長度的22%

$$\text{由式1-67， } K(\infty) = 0.64 + \frac{38}{\text{Re}} = 0.64 + \frac{38}{1000} = 0.6438$$

$$\text{所以 } \Delta P^* = f_{FD} \frac{4x}{D_h} + K(\infty) = \frac{24}{\text{Re}} \frac{4x}{D_h} + K(\infty) = \frac{24}{1000} \times \frac{4 \times 0.05}{0.001} + 0.6438 = 5.4438$$

$$\text{另外由式1-58， } x^+ = \frac{x}{D_h \text{Re}} = \frac{0.05}{0.001 \times 1000} = 0.05$$

$$\text{故 } f_{app} \text{Re} = \frac{3.44}{\sqrt{x^+}} + \frac{24 + \frac{0.1685}{x^+} - \frac{3.44}{\sqrt{x^+}}}{1 + 0.00029(x^+)^{-2}} = \frac{3.44}{\sqrt{0.05}} + \frac{24 + \frac{0.1685}{0.05} - \frac{3.44}{\sqrt{0.05}}}{1 + 0.00029(0.05)^{-2}} = 26.1242$$

$$\therefore f_{app} = \frac{26.12423}{\text{Re}} = 0.02612423$$

$$\Delta P^* = f_{app} \frac{4x}{D_h} = 0.026124 \times \frac{4 \times 0.05}{0.001} = 5.2248$$

此計算結果顯示式1-54與式1-55的結果差不多，其中的差異來自於相關經驗式的誤差。另外提醒讀者，在本例中用完全發展部份的摩擦係數來估算全部的壓降（即 $f_{FD} \frac{4x}{D_h}$ ），可得值4.8，會比式1-55的計算值低上近12%。



在計算前，必須先要瞭解通道中的流動狀態，才能去估算其中的熱傳性能，由於通道的尺寸為 $750 \mu\text{m}$ ，即 $D_h = 0.00075 \text{ m}$ ，由於不知道水的流量，因此首先假設一個流量，例如水流量為每分鐘 1 kg ，即 $\dot{m}_{\text{water}} = 1/60 = 0.01667 \text{ kg/s}$ ，因為發熱量為 300 W

($Q = \dot{m}_{\text{water}} c_{p,\text{water}} (T_{\text{water,out}} - T_{\text{water,in}})$)，所以冷卻水的出口溫度為：

$$T_{\text{water,out}} = \frac{Q}{\dot{m}_{\text{water}} c_{p,\text{water}}} + T_{\text{water,in}} = \frac{300}{0.01667 \times 4178} + 34.3 = 38.6^\circ\text{C}$$

而每一通道的截面積：

$$A_c = \frac{\pi D_h^2}{4} = \frac{\pi \times 0.00075^2}{4} = 4.41786 \times 10^{-7} \text{ m}^2$$

因此每一通道中的質量速度：

$$G_{\text{water}} = \frac{\dot{m}_{\text{water}}}{A_c} = \frac{0.01667}{4.41786 \times 10^{-7}} = 754.51 \text{ kg/m}^2 \cdot \text{s}$$

其相對的雷諾數：

$$\text{Re}_{\text{water}} = \frac{G_{\text{water}} D_h}{\mu_{\text{water}}} = \frac{754.51 \times 0.00075}{0.00074352} = 761.08$$

檢查溫度長發展長度 x^* ，由式1-80，

$$x^* = \frac{L}{D_h \text{ Re Pr}} = \frac{0.05}{0.00075 \times 761.08 \times 5.0} = 0.0175$$



本例其實屬於流場與溫度場同時發展，又由於水冷式設計的應用中(以本例而言)，進出口溫度其實不大，因此本例的熱邊界條件反而比較接近等溫，由式103可知完全發展長度為0.037，故此一冷板微通到熱交換器均落在發展區，由表1-9，當 $x^* = 0.0175$ ， $Pr = 5$ 時， $Nu_{m,T} \approx 5.697$ ，故平均熱傳係數

$$h_{m,T} = Nu_{m,T} k_{water} / D_h = 5.697 \times 0.6235 / 0.00075 = 4736.5 \text{ W/m}^2 \cdot \text{K}$$

同樣的由能量平衡

$$Q = hA\Delta T_m$$

這裡的面積為圓管有效傳熱面積 $= A =$ 通道數目 \times 每一圓管的表面積
 $= 50 \times \pi \times D_h \times L = 50 \times \pi \times 0.00075 \times 0.05 = 0.00589 \text{ m}^2$

ΔT_m 此為壁面與流體間的有效溫差，此一溫度差可由對數平均溫差(log mean temperature difference, LMTD, 見第二章介紹)來估算(也可以用流體進出口溫度的平均值與壁面溫度的差值當作平均溫差，差異應該不大)，故：

$$\Delta T_m = Q / (h_{m,T} A) = 300 / 4736.5 / 0.00589 = 10.75^\circ\text{C}$$

假設壁面為等溫，再由LMTD的定義：

$$LMTD = \frac{(T_s - T_{water,in}) - (T_s - T_{water,out})}{\ln\left(\frac{T_s - T_{water,in}}{T_s - T_{water,out}}\right)} = \frac{(T_s - 34.3) - (T_s - 38.61)}{\ln\left(\frac{T_s - 34.3}{T_s - 38.61}\right)} = 10.75$$

上式為一非線性方程式，讀者可藉由試誤法算出 $T_s \approx 47.35^\circ\text{C}$ ，此一溫度低於 50°C 的需求，因此可考慮持續降入水量，重覆此計算過程，最後可得到最小的水量約為 0.693 kg/min 。

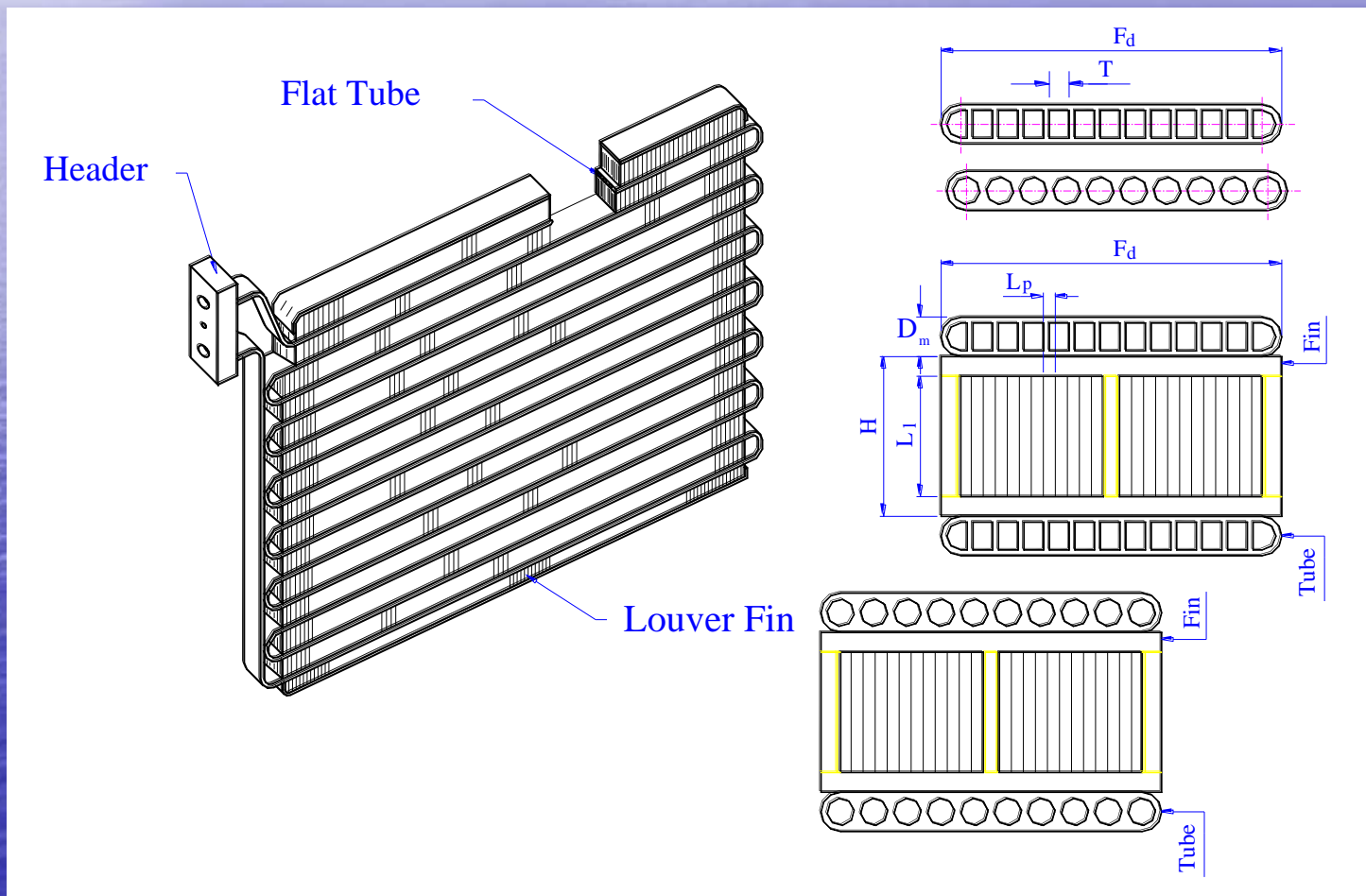


Two-phase Flow in Microchannels

- Fact: For single phase flow, thermalfluids characteristics virtually no change either in Microchannel or conventional macrochannel.
- How about two-phase flow?

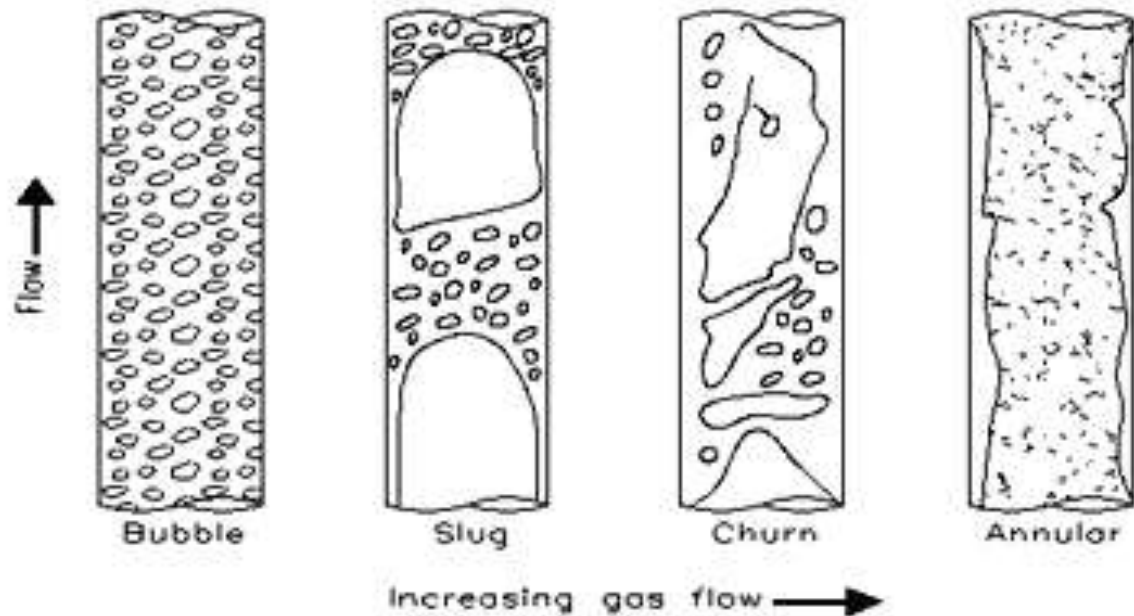
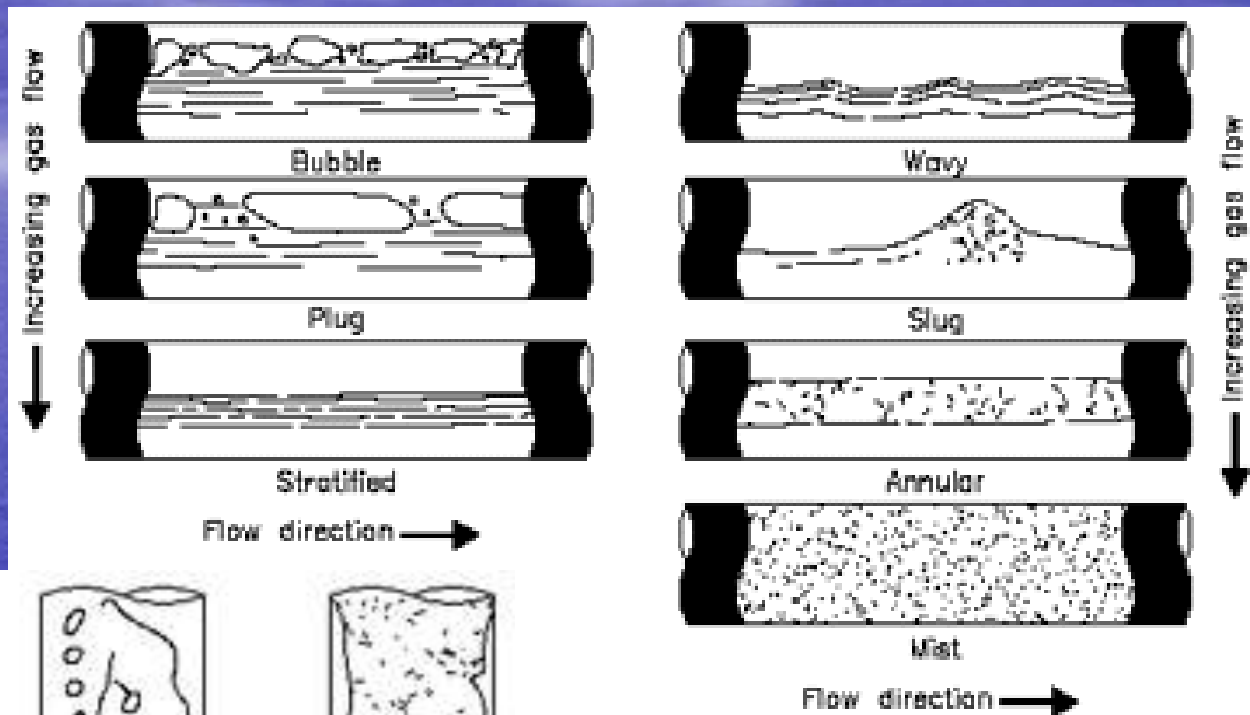


A Typical application of two-phase microchannel heat exchanger



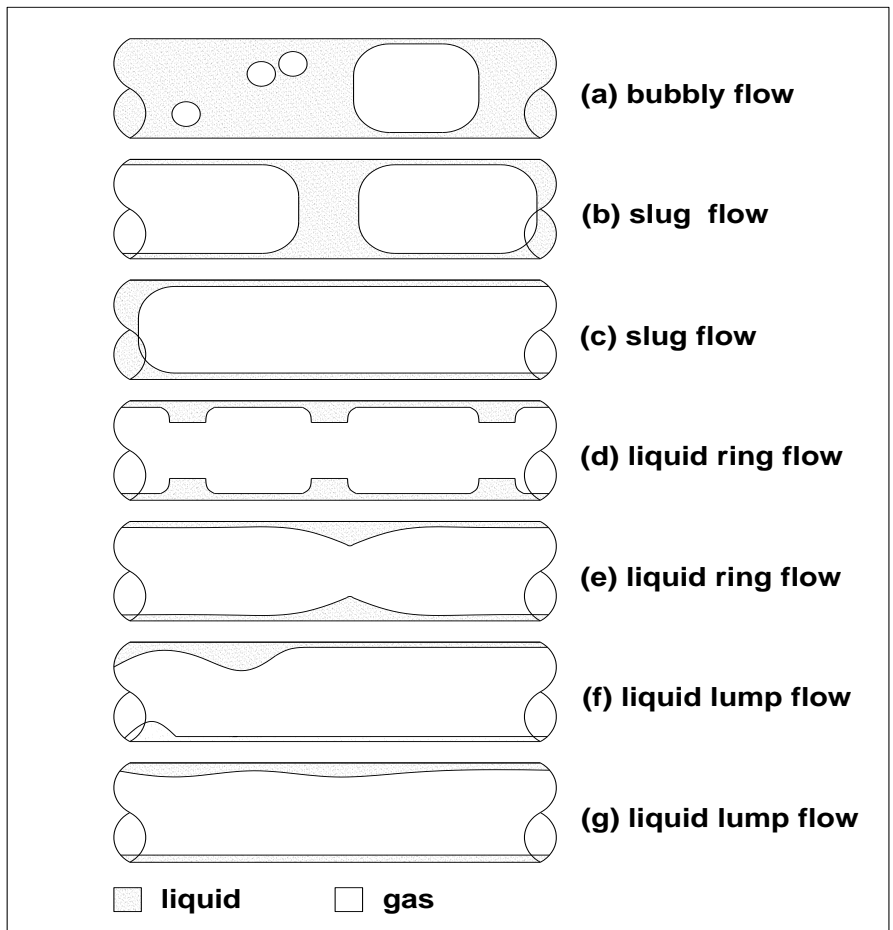
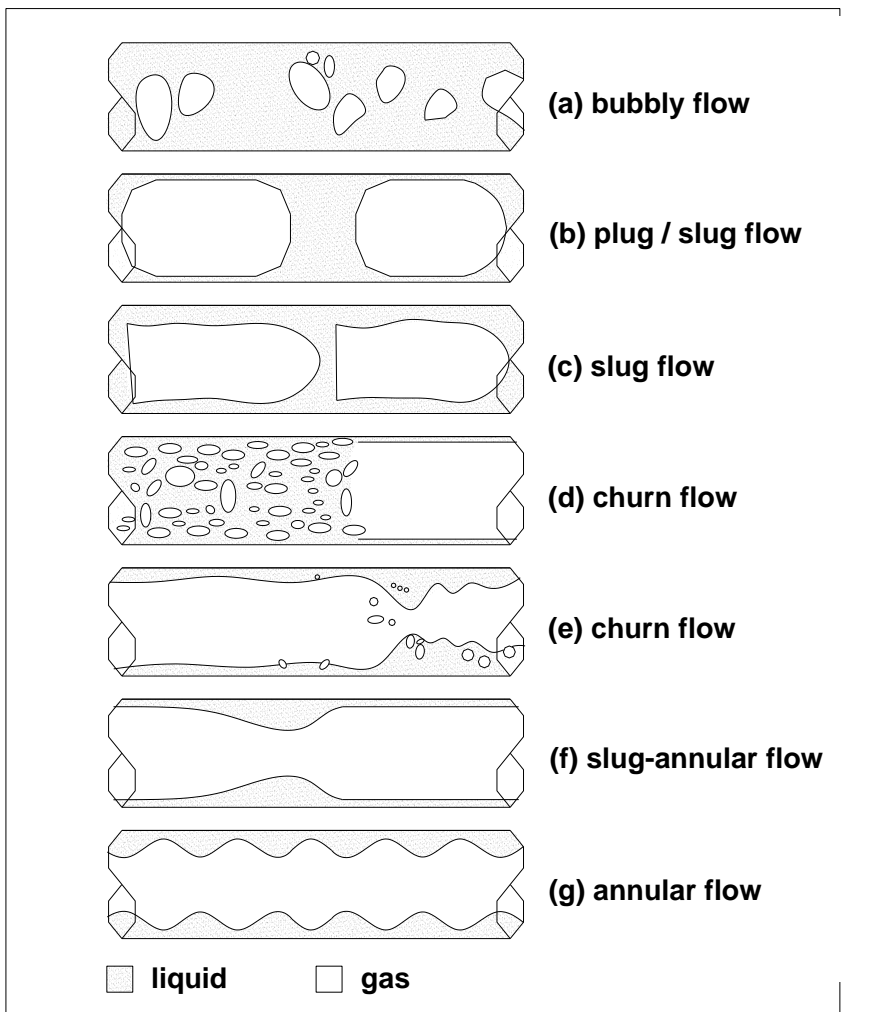


Typical flow pattern for conventional channel





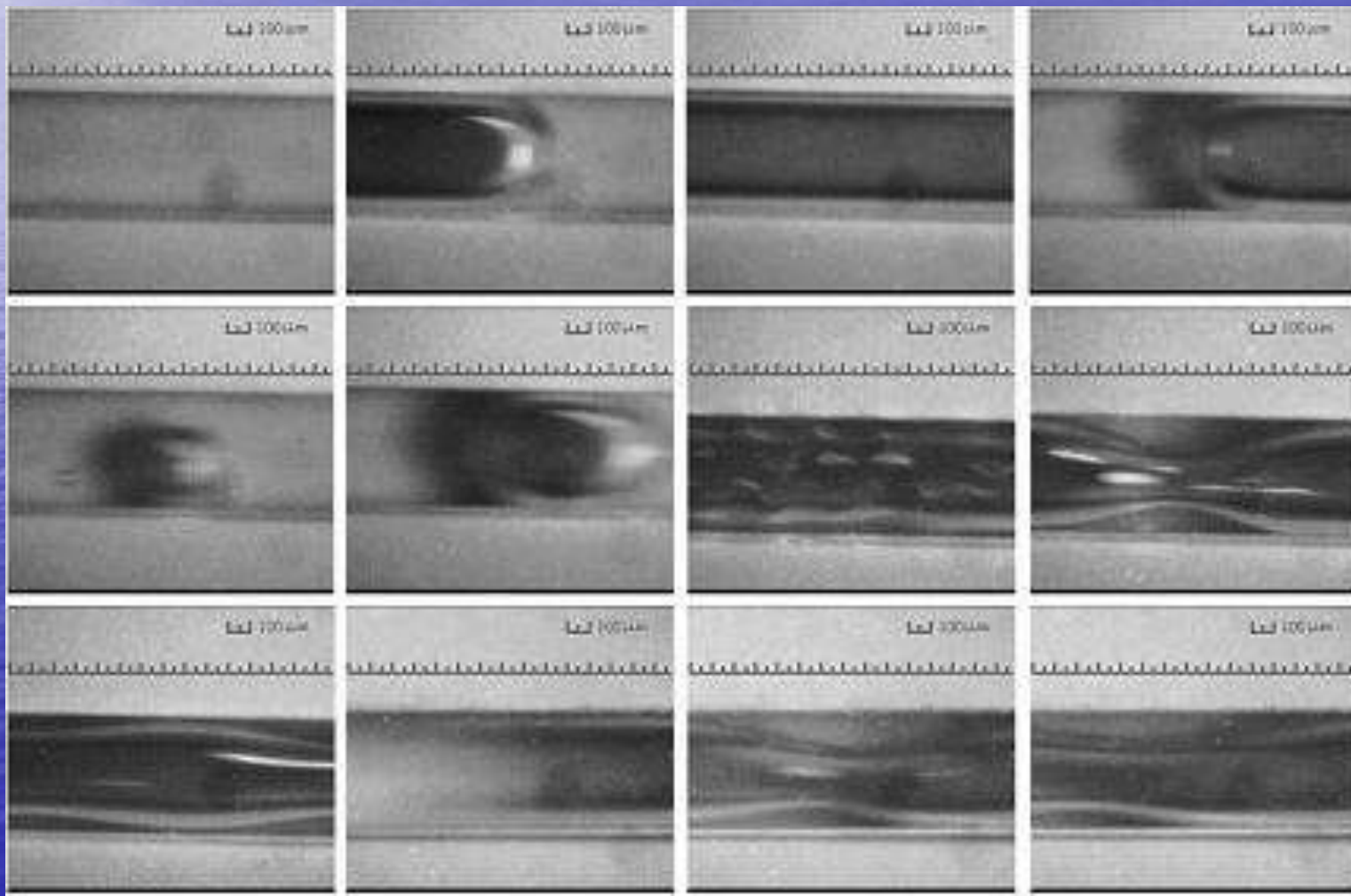
What's difference in two-phase flow pattern





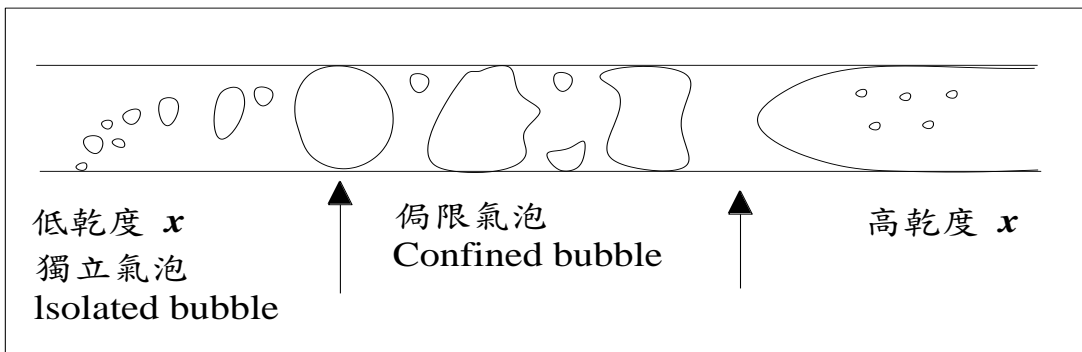
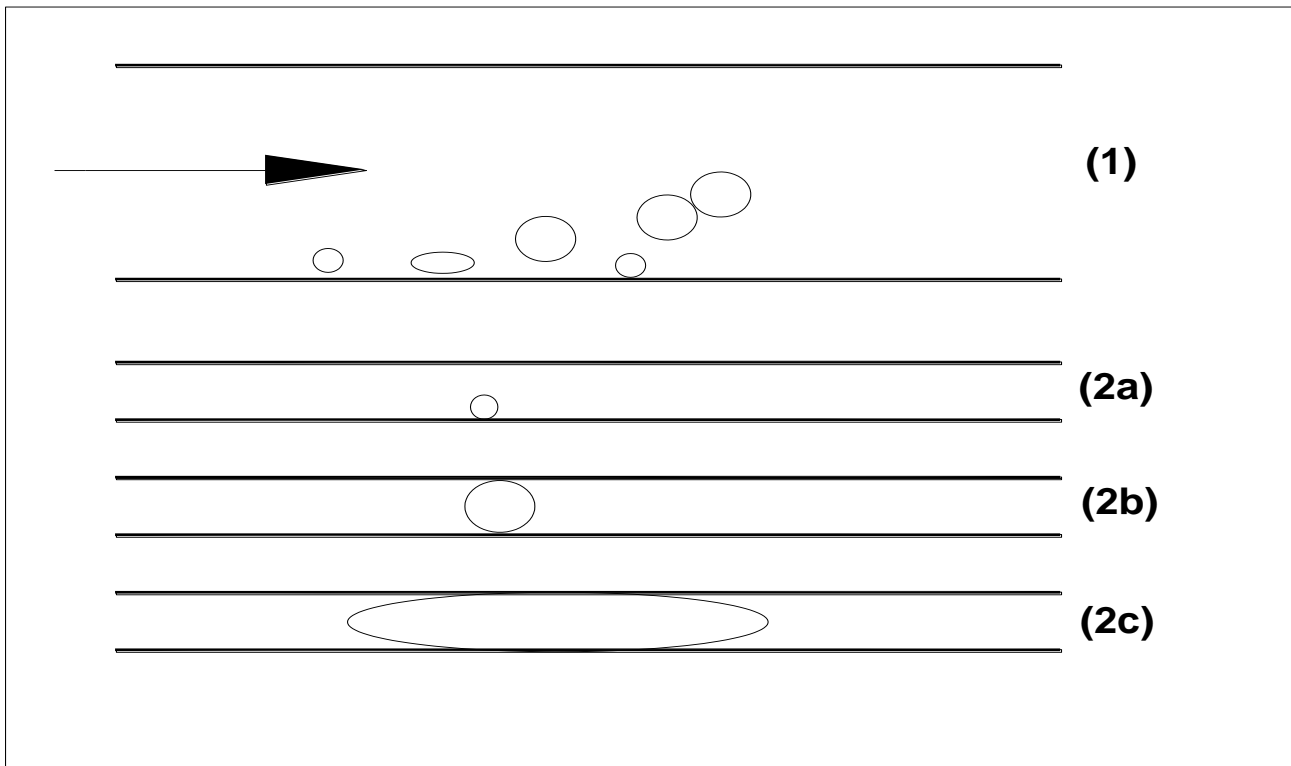
Adiabatic gas-liquid two-phase flow patterns of micro-channels

plaza.nthu.edu.tw/~rxiong/research/STflow.htm





氣泡尺寸相對於通道大小所展現的特性





What is “microchannel” in two-phase system?

- 用Fritz (1935)估算持沸騰條件下氣泡脫離大小的方程式來估算何時可出現微通道

$$d_{Fritz} = 0.0208\beta \left(\frac{\sigma}{g(\rho_L - \rho_G)} \right)^{1/2}$$

上式中的 β 為接觸角(以 $^\circ$ 為單位)，如果考慮流體為R-134a，飽和溫度為 10°C ，假設接觸角為 30° ，則

$$d_{Fritz,R-134a} = 0.0208\beta \left(\frac{\sigma}{g(\rho_L - \rho_G)} \right)^{1/2} = 0.0208 \times 30 \left(\frac{0.01014}{9.81(1261.0 - 20.22)} \right)^{1/2} \approx 0.57 \text{ mm}$$

考慮工作流體為水於相同的飽和溫度，如果假設其接觸角為 80° ，則：

$$d_{Fritz,H_2O} = 0.0208\beta \left(\frac{\sigma}{g(\rho_L - \rho_G)} \right)^{1/2} = 0.0208 \times 80 \left(\frac{0.07424}{9.81(999.6 - 0.00939)} \right)^{1/2} \approx 4.58 \text{ mm}$$

Chinnov and Kabov (2006)則提出與式2類似的觀念，利用毛細常數將通道分成四個區塊

$$l_{\sigma} = \left(\frac{\sigma}{g(\rho_L - \rho_G) \cos \xi} \right)^{1/2}$$

- (1) 大尺度通道 (large-scale channel), $D_h > 5l_{\sigma}$, 此區域中毛細力的影響可以完全忽略(當然, 毛細力在氣液界面與氣泡生成脫離仍扮演一定的角色)。
- (2) 重力/毛細力通道 (gravity-capillary channel), $0.5l_{\sigma} < D_h < 5l_{\sigma}$, 此區重力與毛細力都聚一定程度的影響, 但重力的影響大於毛細力。
- (3) 毛細力/重力通道 (capillary-gravity channel), $0.1l_{\sigma} < D_h < 0.5l_{\sigma}$, 此區毛細力的影響大於重力。
- (4) 毛細力通道 (capillary-gravity channel), $D_h < 0.1l_{\sigma}$, 此區重力的影響可以完全忽略。



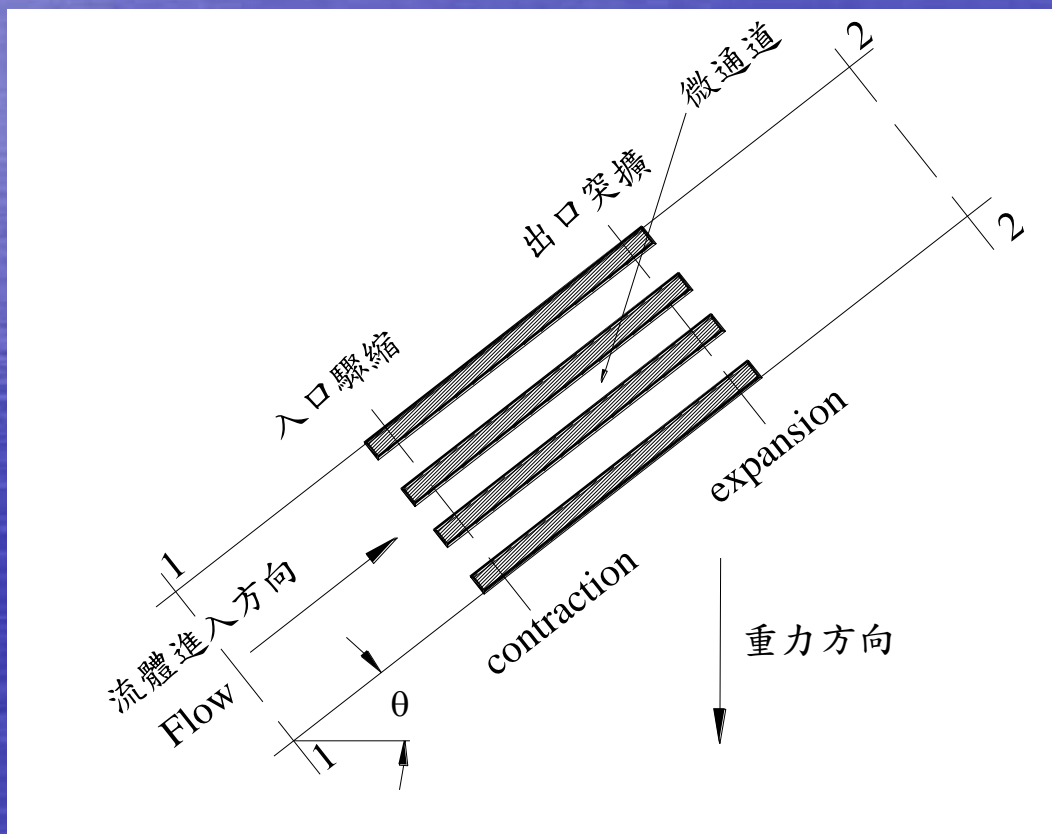
Selection criteria for working fluids

- 在工作溫度下，壓力略高於大氣壓(以避免真空洩漏與管件搭配的相關問題)。
- 潛熱高。
- 好的熱傳與壓降特性。
- 如果要直接運用到電子零件的散熱，流體最好具備高介電常數。
- 如果要直接運用到電子矽晶片的散熱，流體必需與矽材料相容。
- 好的管路流動的密封性。
- 在操作條件下，化學穩定高。



微通道內兩相壓降的估算

- $\Delta P = \Delta P_c + \Delta P_f + \Delta P_g + \Delta P_a + \Delta P_e$





- ΔP_c 為流入微通道時因流道變小所造成的壓降，Coleman (2003)建議採用Hewitt (2000)的估算方法：

$$\Delta P_c = \frac{G^2}{2\rho_L} \left(\left(\frac{1}{C_o} - 1 \right)^2 + 1 - \frac{1}{\sigma_c^2} \right) \psi_h \quad (5)$$

$$C_o = \frac{1}{0.639 \left(1 - \frac{1}{\sigma_c} \right)^{0.5} + 1} \quad (6)$$

$$\psi_h = \left(1 + x \left(\frac{\rho_L}{\rho_G} - 1 \right) \right) \quad (7)$$



- ΔP_a 為流體因加熱造成密度變化所造成的壓降，如果考慮均質流動，則可近似如下

$$\Delta P_a = \int_{x=x_{in}}^{x=x_{out}} G^2 v_{LG} \frac{dx}{dz} dz = G^2 \left(\frac{1}{\rho_G} - \frac{1}{\rho_L} \right) \Delta x \quad (8)$$

如果考慮均勻加熱，則

$$dx = \frac{q P_h dz}{i_{LG}} \quad (9)$$

其中 q 為熱通量， P_h 為周長， i_{LG} 為潛熱。

$$\Delta P_a = \int_{z=0}^{z=z_{out}} G^2 v_{LG} \frac{dx}{dz} dz = \int_{z=0}^{z=z_{out}} G^2 v_{LG} dx = G^2 \left(\frac{1}{\rho_G} - \frac{1}{\rho_L} \right) (x_{out} - x_{in}) \quad (10)$$



ΔP_g 為流體因為高度變化由重力所造成的壓降

$$\Delta P_g = \int_{x=x_{in}}^{x=x_{out}} \frac{g \sin \theta}{v_L (1 + xv_{fg}/v_L)} dz \approx \frac{g \sin \theta}{\rho_L \left(1 + \rho_L \left(\frac{x_{in} + x_{out}}{2} \right) \left(\frac{1}{\rho_G} - \frac{1}{\rho_L} \right) \right)}$$



ΔP_e 為流出微流道時通道變大所造成的壓降，
Coleman (2003) et al. (2003) 建議採用 Hewitt et
al. (1994) 均質流動的估算模式如下：

$$\Delta P_e = \frac{G^2}{\rho_L} \sigma_e (1 - \sigma_e) \psi_s$$

$$\psi_s = 1 + \left(\frac{\rho_L}{\rho_G} - 1 \right) (0.25x(1-x) + x^2)$$



The best correlation for evaluation of Two-phase ΔP_f in microchannel

- Müller-Steinhagen and Heck (1986) correlation

$$\left(\frac{dP}{dz}\right)_f = \left(\left(\frac{dP}{dz}\right)_L + 2x \left(\left(\frac{dP}{dz}\right)_G - \left(\frac{dP}{dz}\right)_L \right) \right) (1+x)^{1/3} + \left(\frac{dP}{dz}\right)_G x^3$$

上式中的 $\left(\frac{dP}{dz}\right)_L$ 與 $\left(\frac{dP}{dz}\right)_G$ 分別代表將全部兩相流體視為全部為單相液

體或氣體的條件下所計算出的壓降梯度，即：

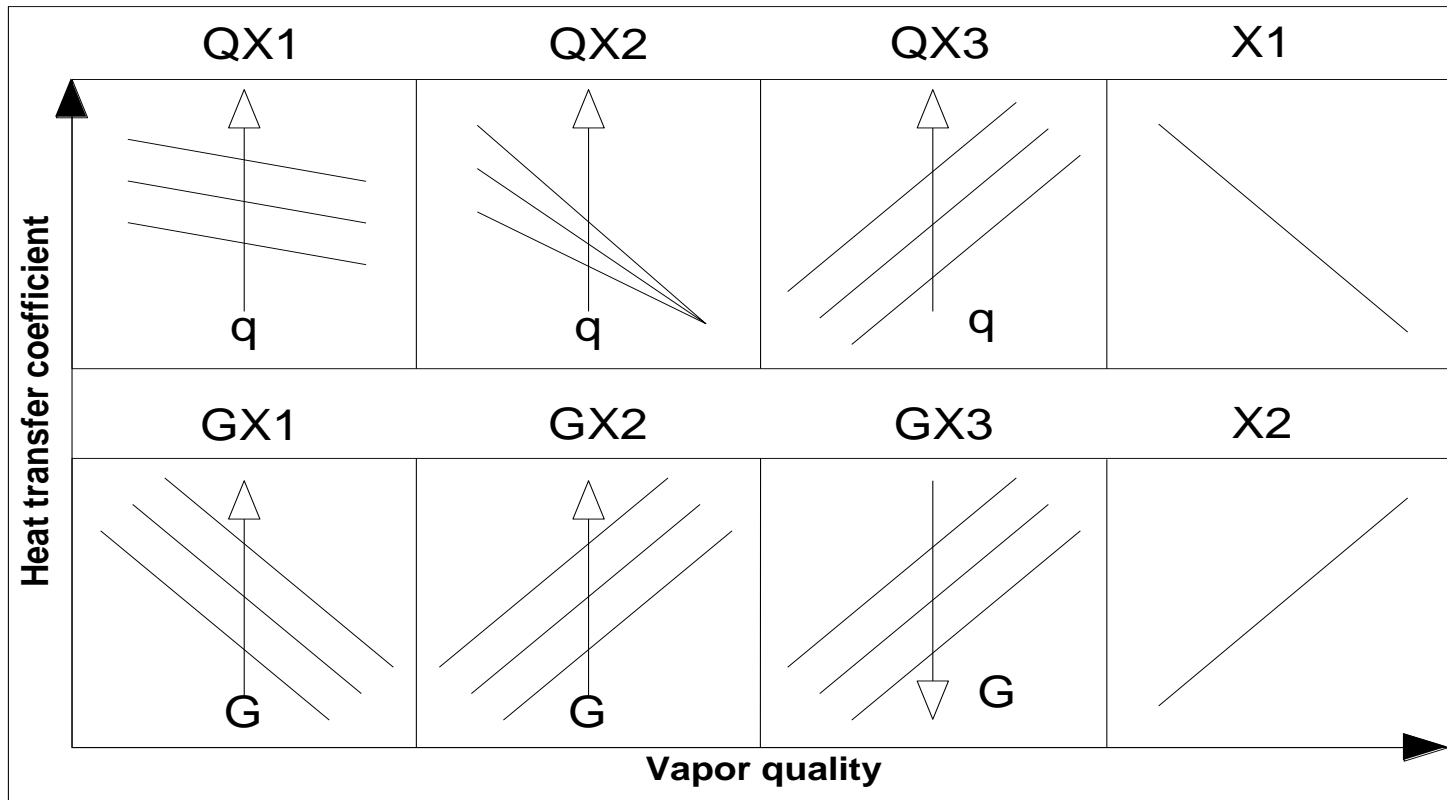
$$\left(\frac{dP}{dz}\right)_L = f_L \frac{2G^2}{D_h \rho_L} \quad (15)$$

$$\left(\frac{dP}{dz}\right)_G = f_G \frac{2G^2}{D_h \rho_G} \quad (16)$$

其中摩擦係數可依流動型態為層流或紊流來計算：

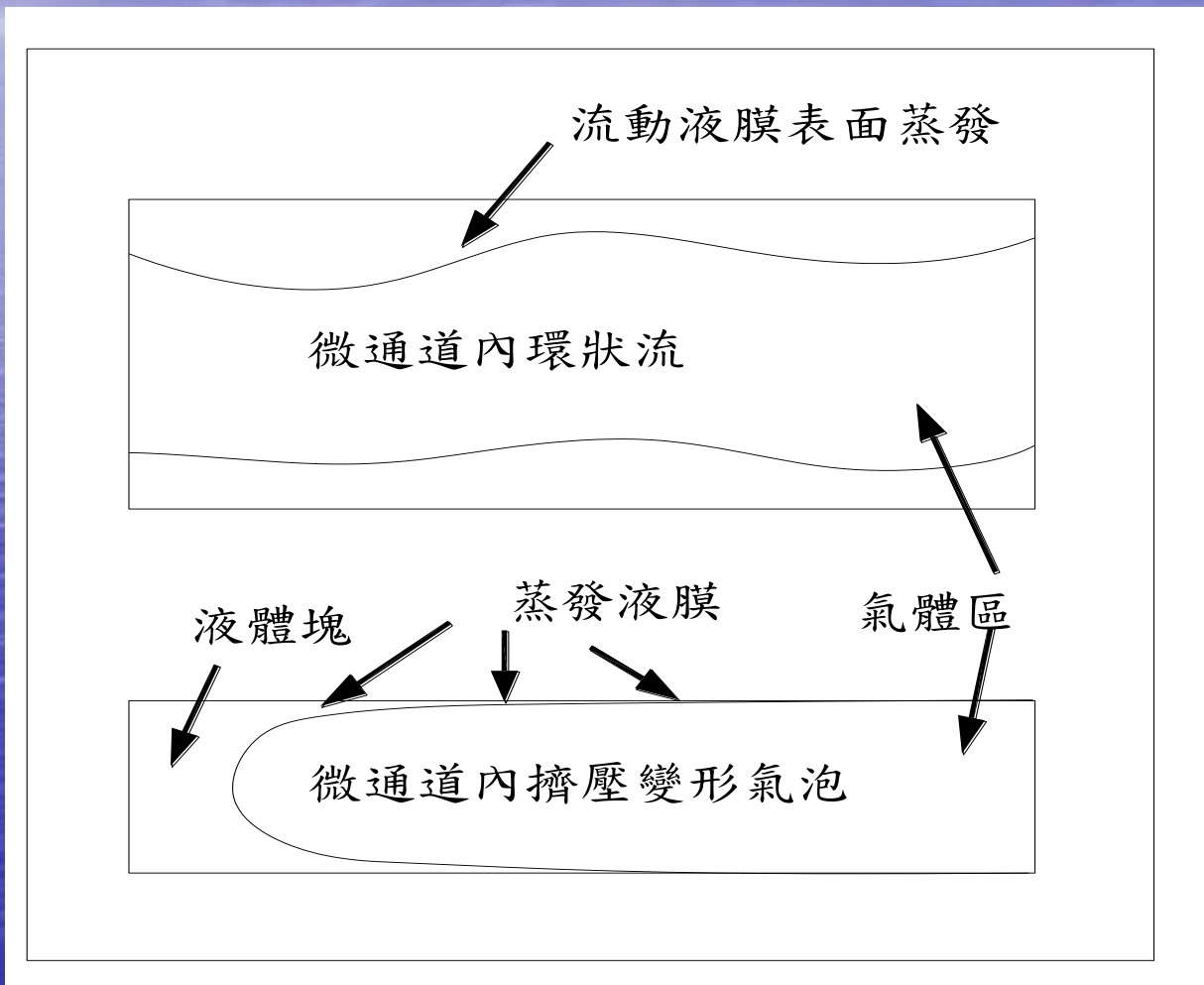
$$f_{L或G} = \begin{cases} \frac{16}{\text{Re}_{L或G}} & \text{如果 } \text{Re}_{L或G} < 2300 \\ 0.0791 \text{Re}_{L或G}^{-0.25} & \text{如果 } \text{Re}_{L或G} \geq 2300 \end{cases} \quad (17)$$

微通道內兩相沸騰熱傳係數的估算



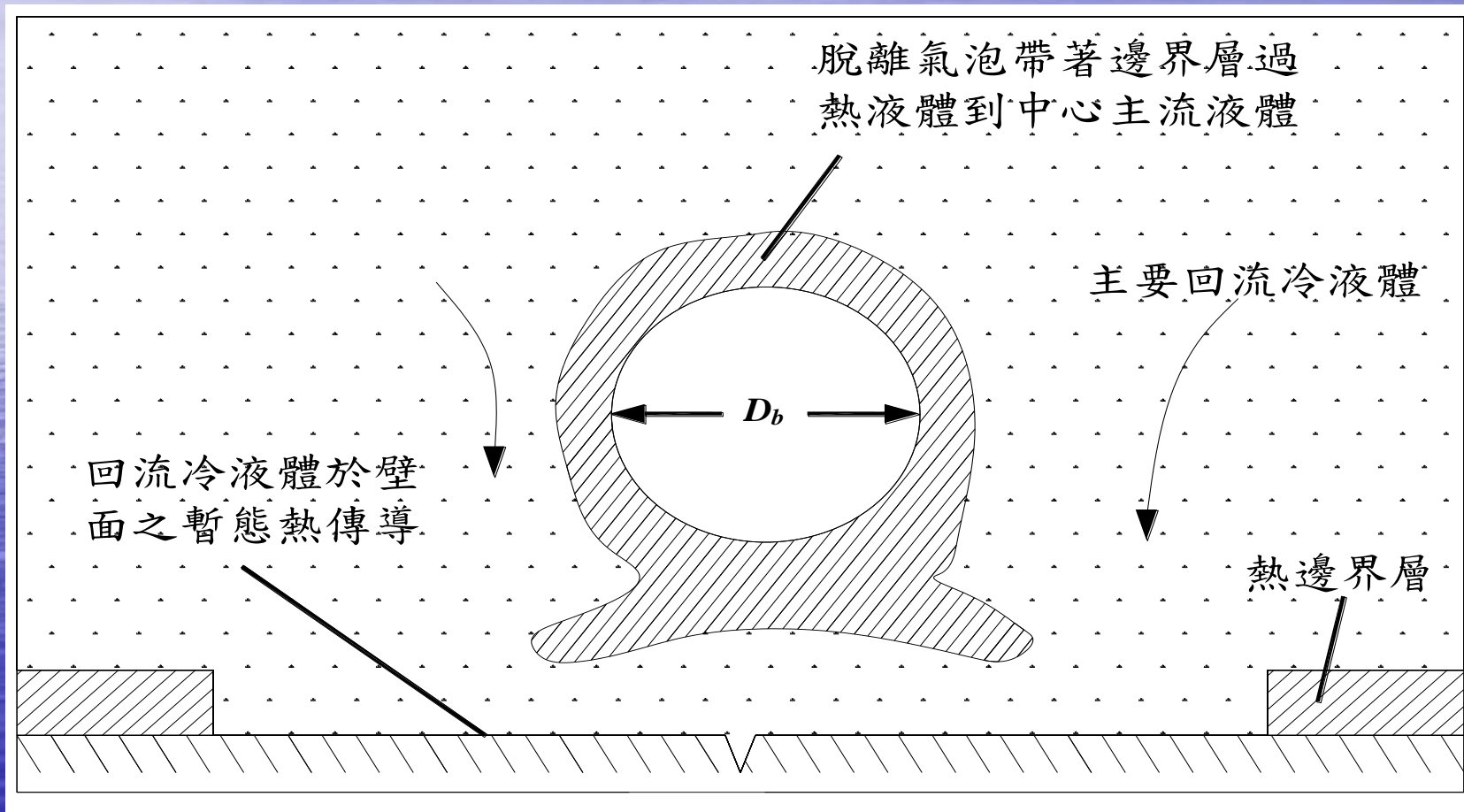


Basic Heat Transfer mechanism in Microchannel vs. evaporation in annular flow



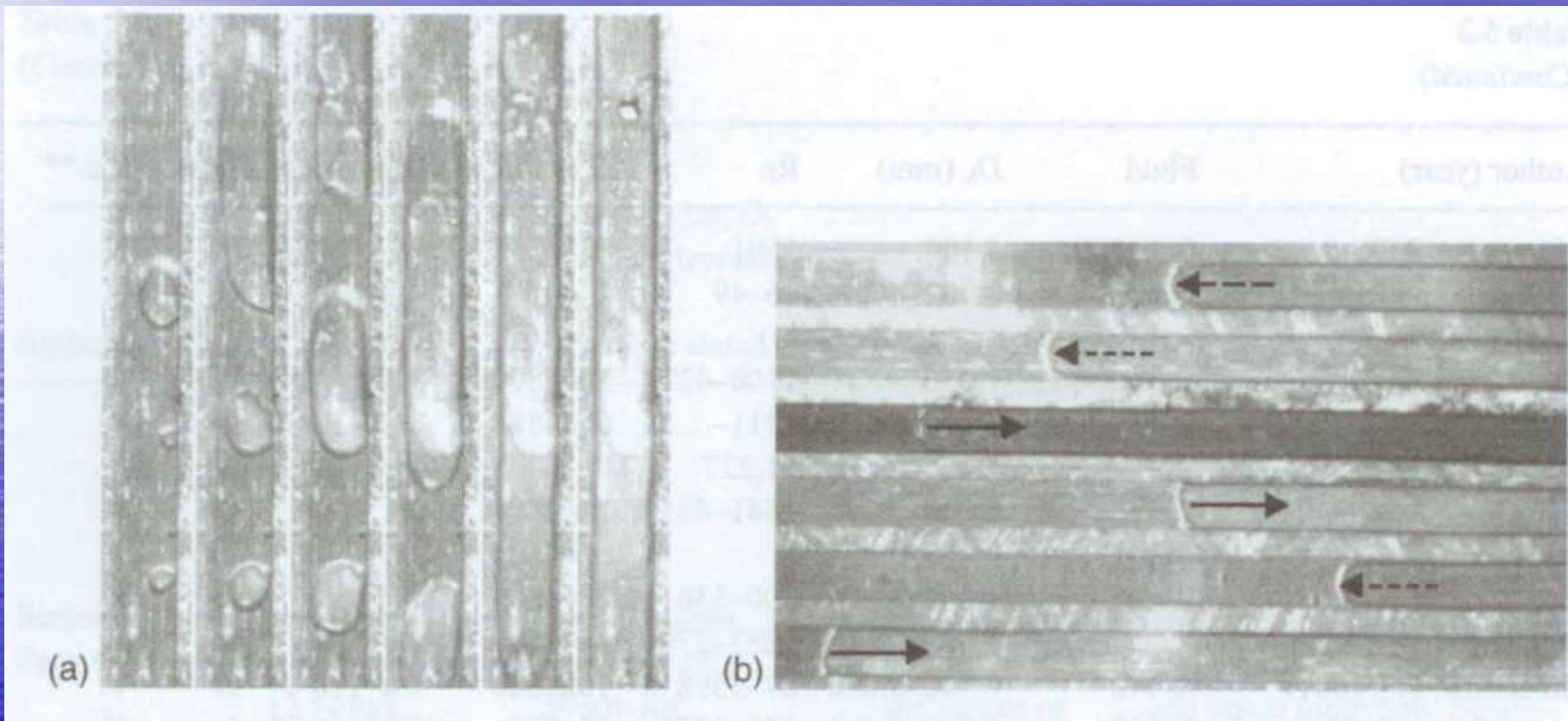


Conventional Macroscale Boiling mechanism





多通道微通道的流動震盪現象





The correlation for two-phase flow in microchannels Kandlikar and Balasubramanian (2004)

$$h_{D,c} = \begin{cases} h_{D,sbo} & \text{中兩者比較大者} \\ h_{D,cbo} \end{cases} \quad (18)$$

$$\frac{h_{D,sbo}}{h_{L,c}} = 0.6683Co^{-0.2}(1-x)^{0.8} + 1058Bo^{0.7}(1-x)^{0.8}F_{FI} \quad (19)$$

$$\frac{h_{D,cbo}}{h_{L,c}} = 1.136Co^{-0.9}(1-x)^{0.8} + 667.2Bo^{0.7}(1-x)^{0.8}F_{FI} \quad (20)$$

$$Bo = \frac{q}{Gj_{f,b}} \quad (\text{boiling number}) \quad (21)$$

$$Co = \left(\frac{1-x}{x}\right)^{0.8} \left(\frac{\rho_G}{\rho_L}\right)^{0.5} \quad (\text{Convection number}) \quad (22)$$

$$h_{LO} = \frac{D_h}{k_L} = \begin{cases} \left[\frac{\left(\frac{f_L}{2}\right)(Re-1000)Pr}{1.07 + 12.7\sqrt{\frac{f_L}{2}}(Pr_L^{2/3}-1)} \right] & \text{當 } 100 \leq Re_{LO} \leq 1600 \\ \text{以 } Re_{LO} = 1600 \text{ 與 } Re_{LO} = 3000 \\ \text{的計算值再進行線性內插} & \text{當 } 1600 \leq Re_{LO} \leq 3000 \\ \left[\frac{\left(\frac{f_L}{2}\right)Re_{LO}Pr_L}{1.07 + 12.7\sqrt{\frac{f_L}{2}}(Pr_L^{2/3}-1)} \right] & \text{當 } 3000 \leq Re_{LO} \leq 10^4 \\ \left[\frac{\left(\frac{f_L}{2}\right)Re_{LO}Pr_L}{1.07 + 12.7\sqrt{\frac{f_L}{2}}(Pr_L^{2/3}-1)} \right] & \text{當 } 10^4 \leq Re_{LO} \leq 5 \times 10^6 \end{cases} \quad (23)$$

冷媒	F_{FI} 常數 Kandlikar 方程式計算用
水 (water)	1.0
R-11	1.3
R-12	1.5
R-13B1	1.31
R-113	1.3
R-114	1.24
R-22	2.2
R-134a	1.63
Kerosene	0.488
R-141b	1.8
R-124	1.0
R-410A	3.3
R-152a	1.1

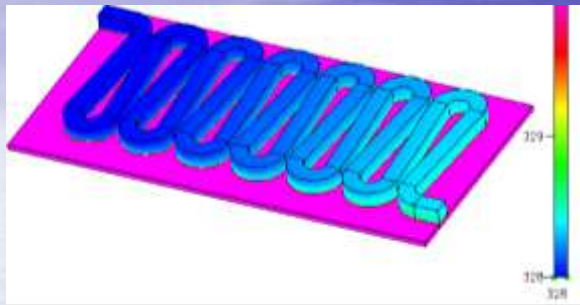
Summary of Microchannel HX

- Microchannel HX is considered as an alternative for high-flux electronic cooling applications, and it is applicable in both single and two-phase applications.
- The basic heat transfer and frictional characteristics for microchannel in single phase flow is the same as that of macrochannel. However, the two phase flow shows a considerable difference in it subject to the bubble/flow phenomena.
- Heat Transfer augmentation is an effective way for laminar flow cold-plate.
- Influence of entrance should be considered for microchannel HXs.
- Mal-distribution could be a concern for multiple port channel cold-plate.
- In two-phase flow application, contraction loss may be quite important when compared to frictional loss.
- The two-phase boiling heat transfer coefficient is usually increased with the rise of heat flux but shows no (or slight decrease) with vapor quality.
- The two-phase flow pattern for microchannel is quite different from that of macrochannel with virtually no stratified flow pattern.

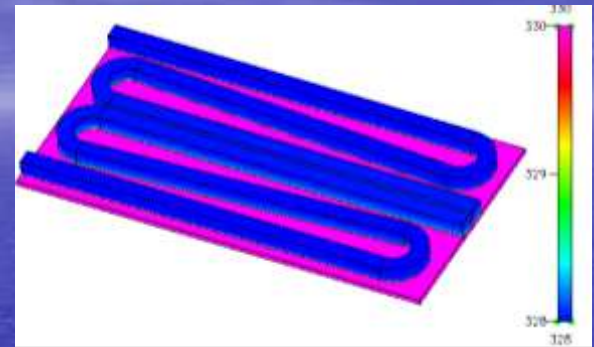


Effect of Mal-distributions..

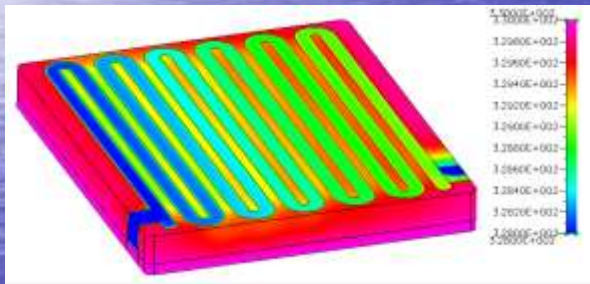
Serpentine vs. multi-port design



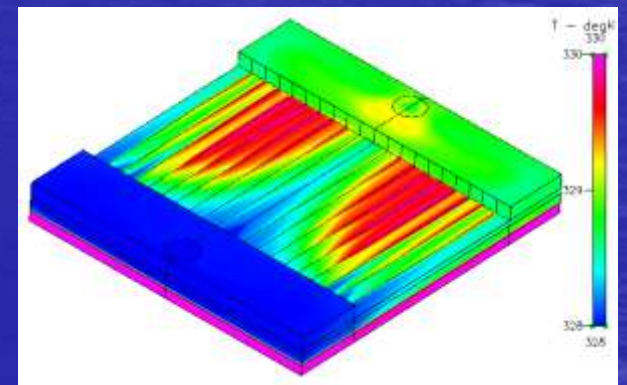
$\Delta P = 4.9 \text{ kPa}$
 $Q = 47.94 \text{ W}$



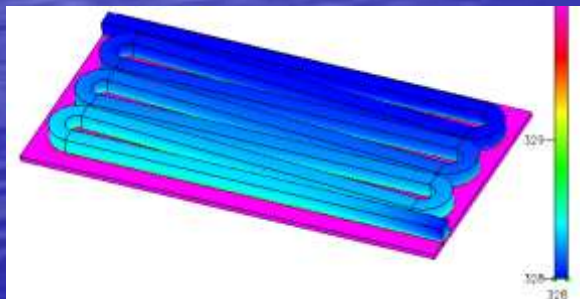
$\Delta P = 45.2 \text{ kPa}$
 $Q = 199.93 \text{ W}$



$\Delta P = 3.4 \text{ kPa}$
 $Q = 48.72 \text{ W}$



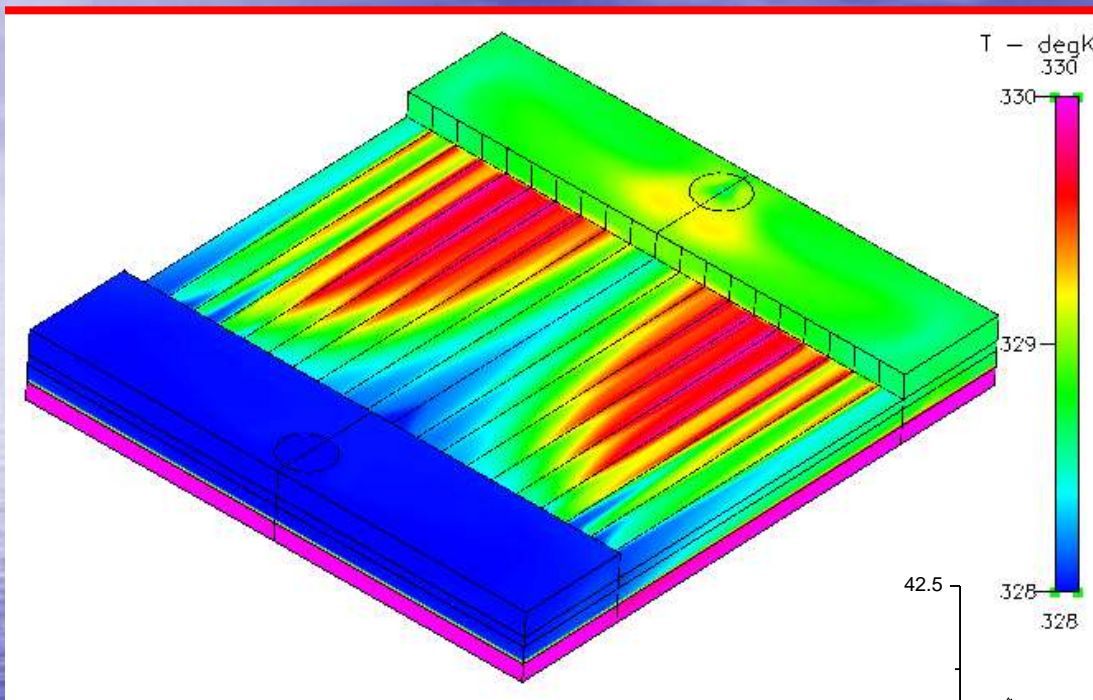
$\Delta P = 2.17 \text{ kPa}$
 $Q = 126.91 \text{ W}$



$\Delta P = 1.2 \text{ kPa}$
 $Q = 13.73 \text{ W}$

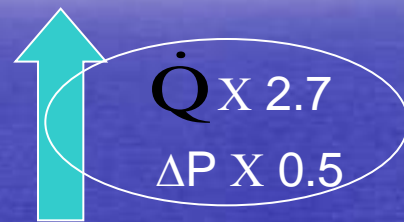


Multi-port HX

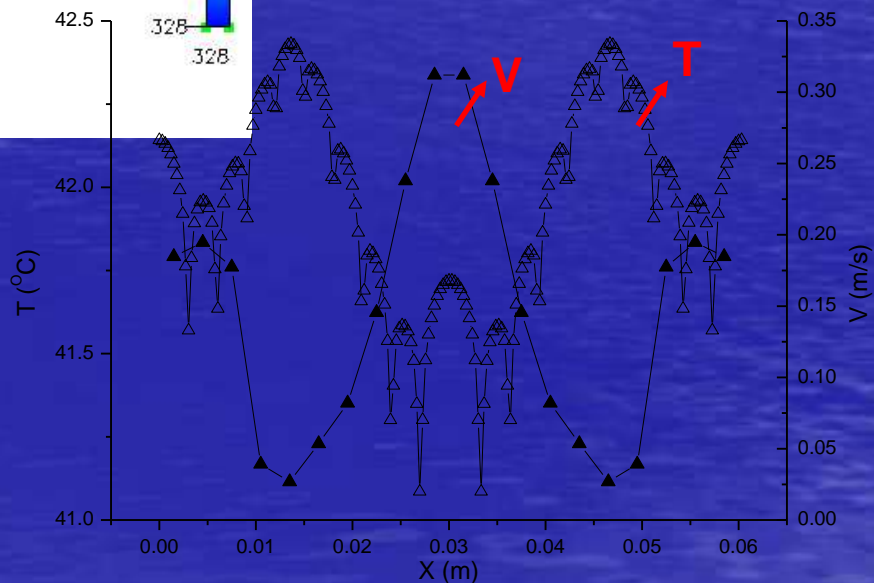


$$\Delta P = 2.17 \text{ kPa}$$

$$\dot{Q} = 126.91 \text{ W}$$



(1) $\Delta P = 4.9 \text{ kPa}$
 $\dot{Q} = 47.94 \text{ W}$



$$V_{in} = 1.0 \text{ (m/s):}$$

Flow mal-distribution:

$$\Delta V_{MAX} = 0.2852 \text{ (m/s)}$$

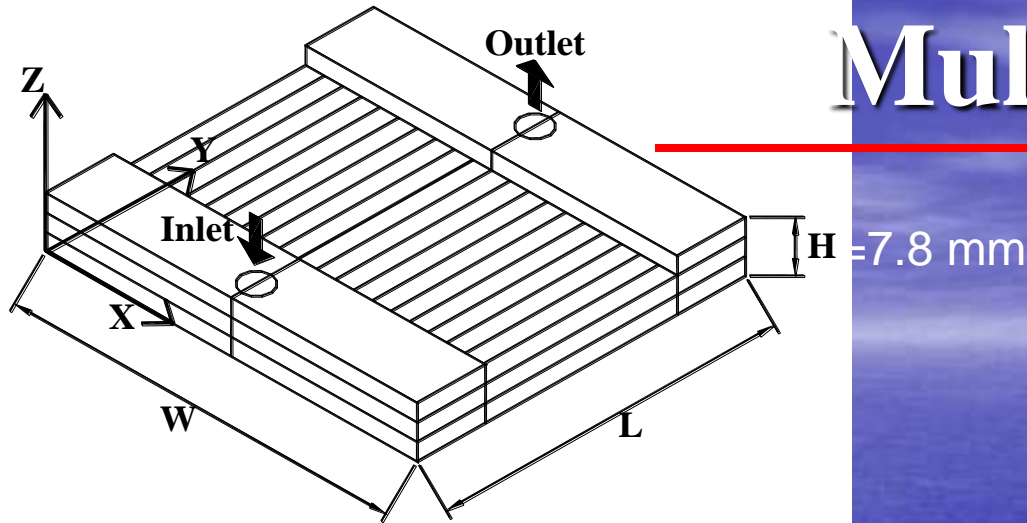
Non-uniformity of Temperature field:

$$\Delta T_{MAX} = 1.345 \text{ (}^\circ\text{C)}$$



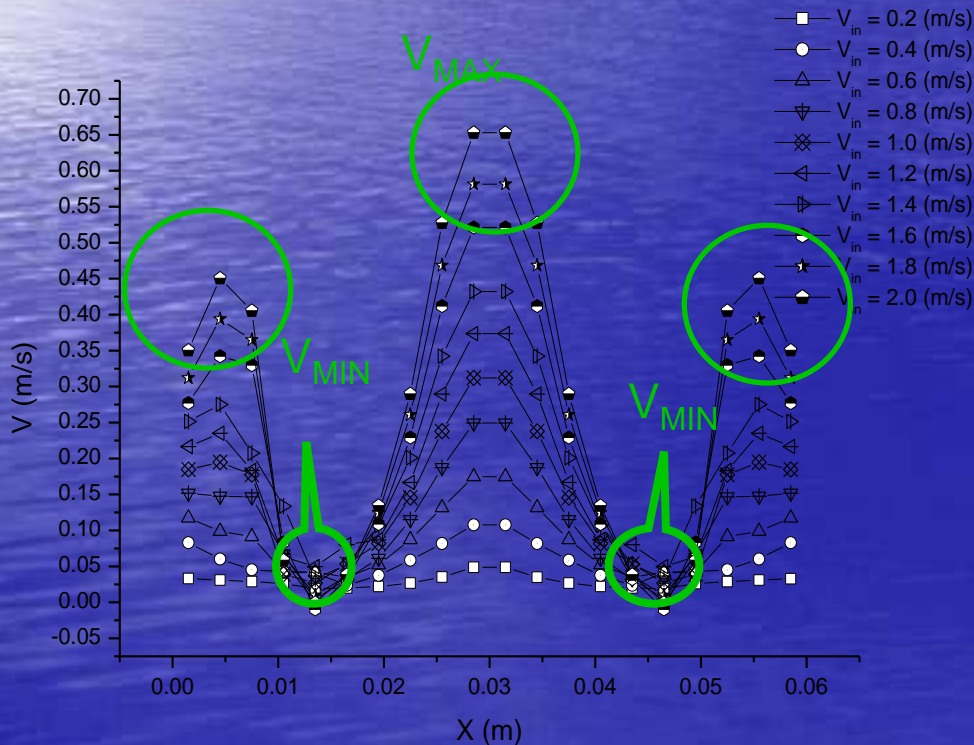
$$\Delta Q_{MAX} = 148 \text{ (W)}$$

Multi-port Cold Plate



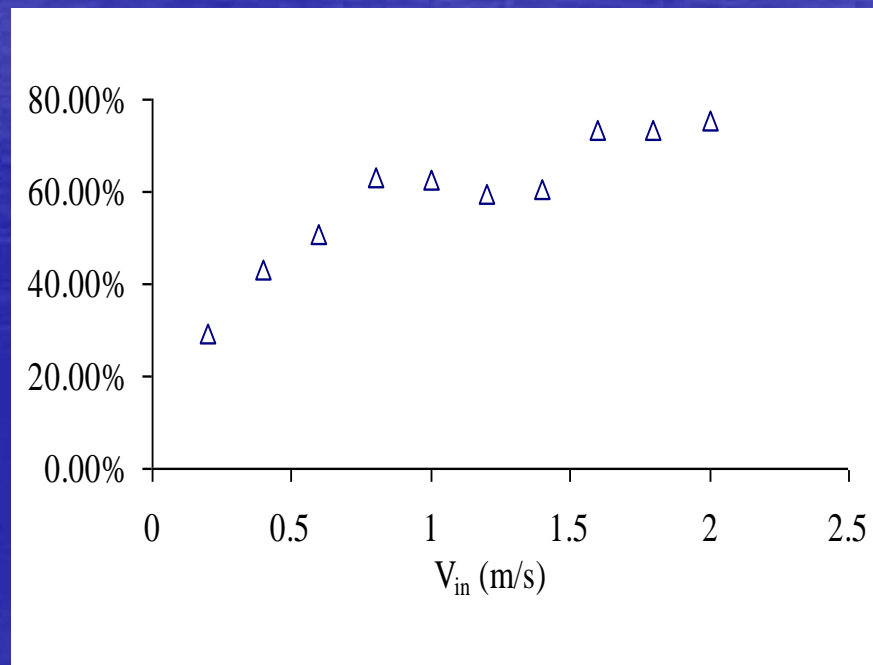
$$V_{STD} \equiv \sqrt{\frac{n \sum_i v_i^2 - (\sum_i v_i)^2}{n(n-1)}}$$

Transverse Velocity Distribution



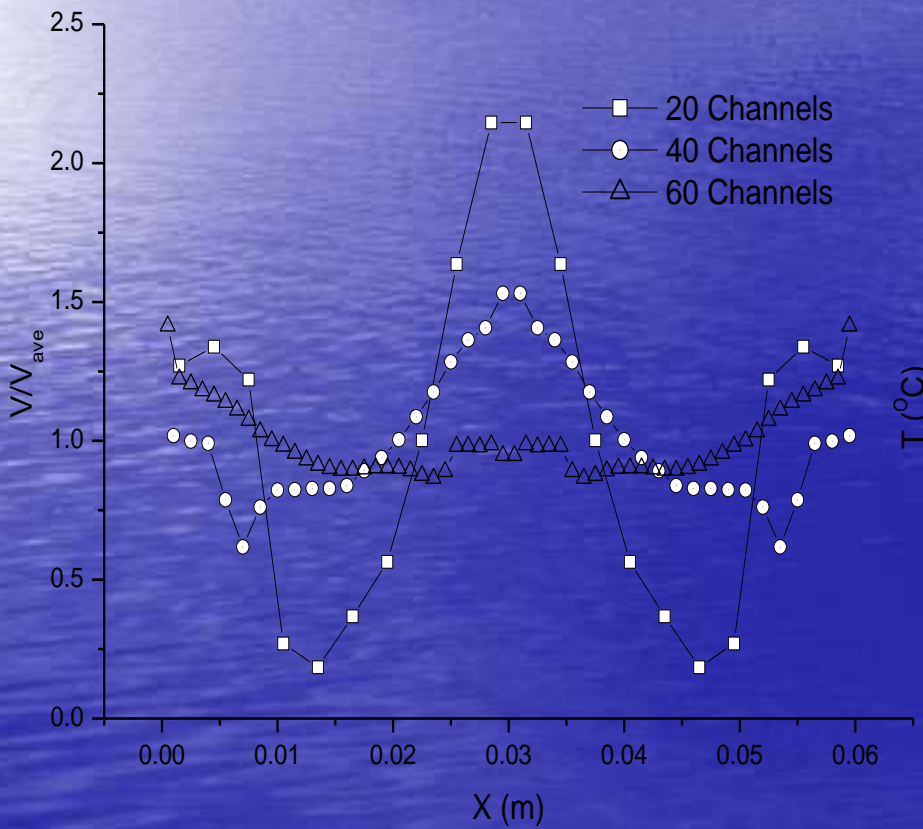
Flow Mal-distribution

$$V_{STD}/V_{ave}$$

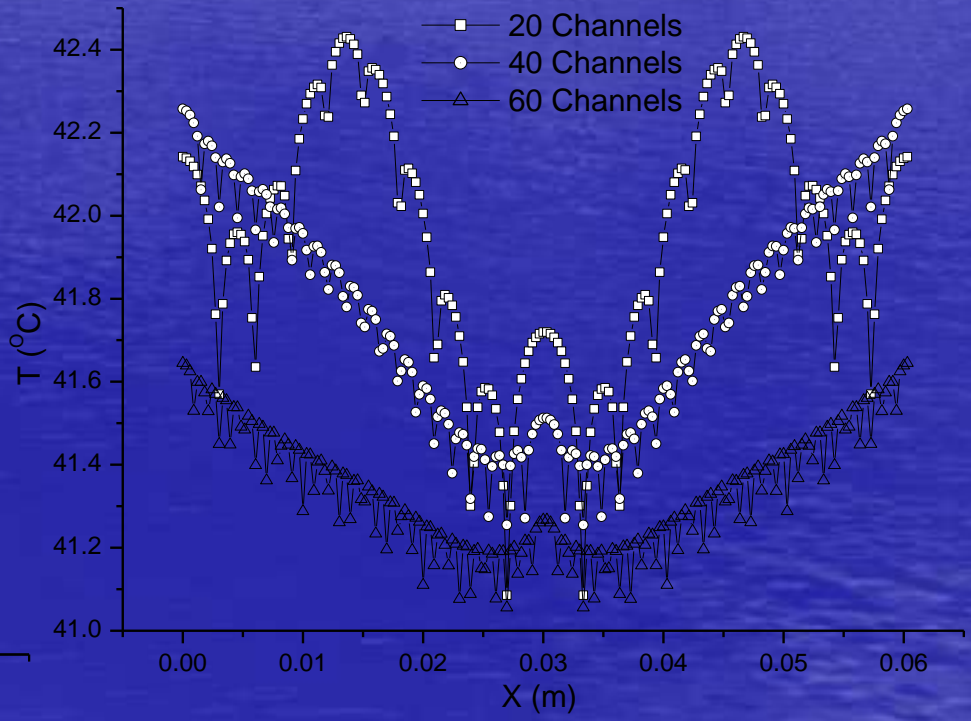


Effect of number of port (20, 40, 60)

Velocity profiles of 20(□), 40(O) and 60(Δ) channels for $V_{in} = 1.0$ m/s.



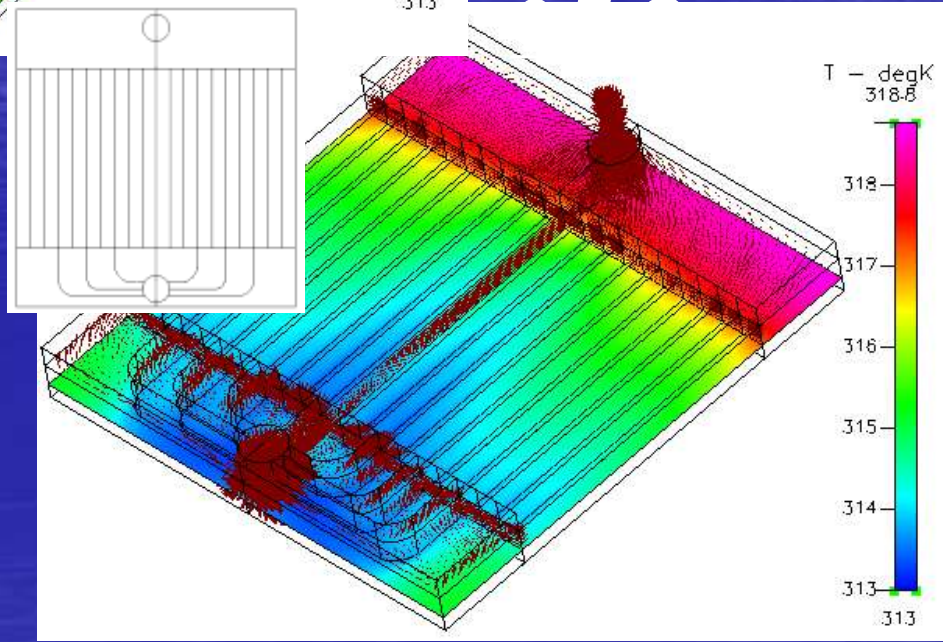
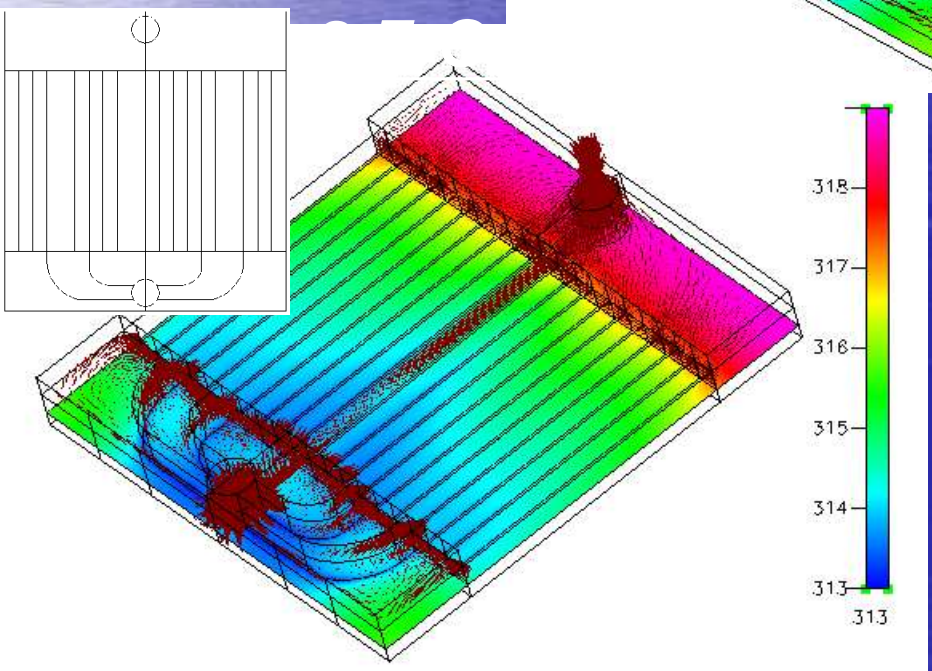
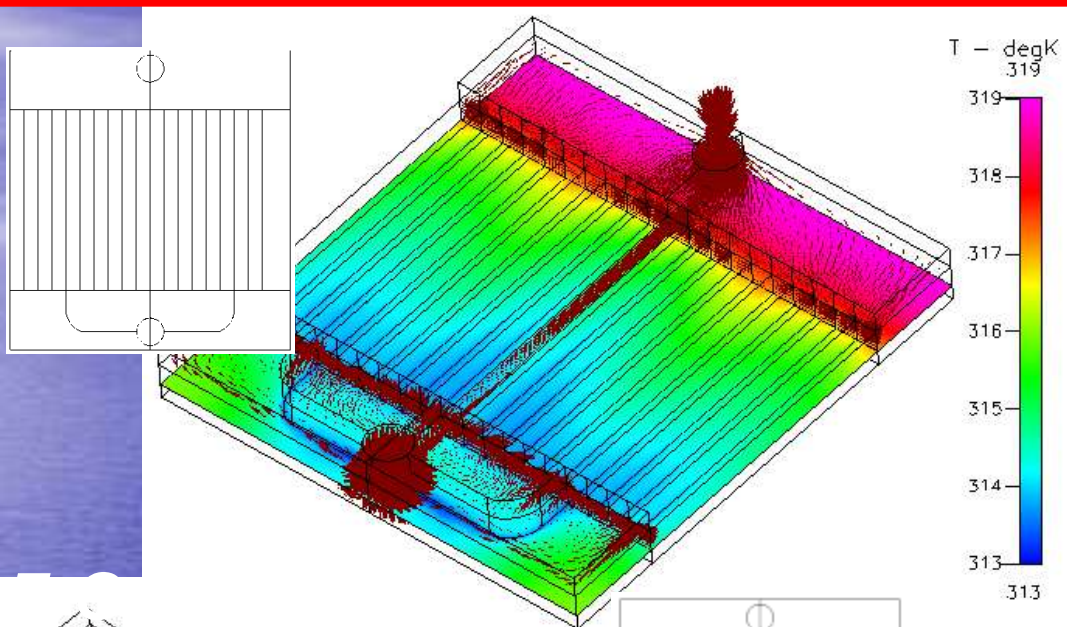
Temperature distribution of 20 (□), 40 (O) and 60 (Δ) channels cold-plates for $V_{in} = 1.0$ m/s.



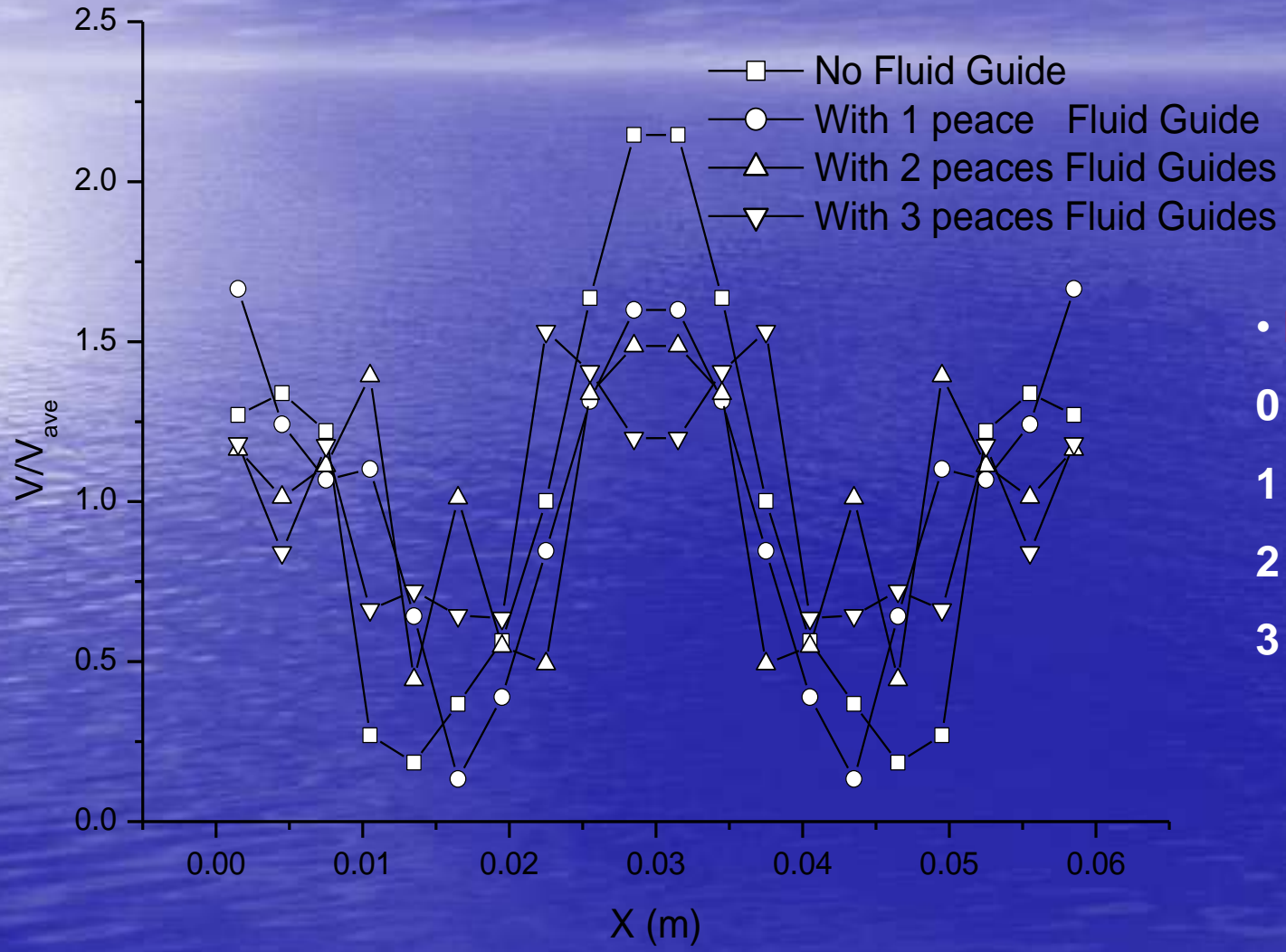


Influence of Guide-plate

1 F. G.



Influence of Guide Plate, Conti.



• **Flow Mal-dis.**

0 F.G. : 62.7 %

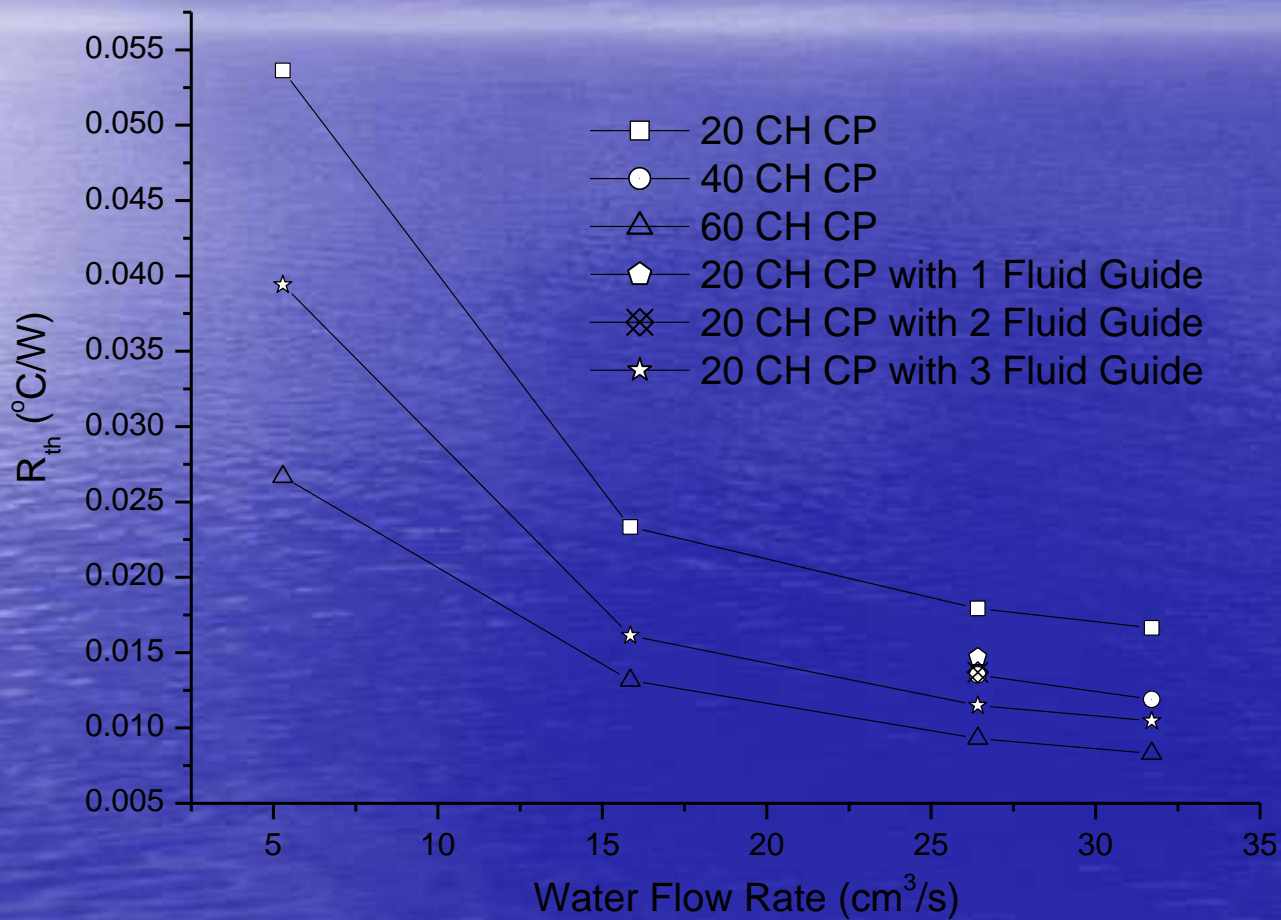
1 F.G. : 48.7 %

2 F.G. : 37.2 %

3 F.G. : 32.3 %

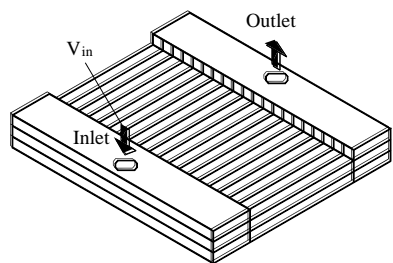


Effect of Guide Plate – R_{th}

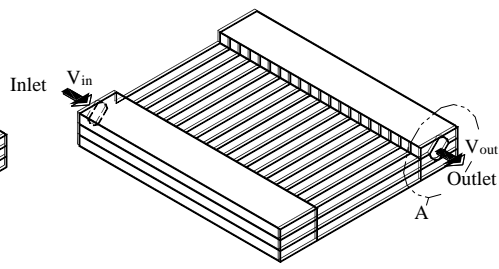




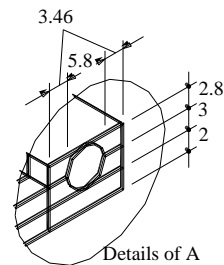
Effect of Inlet locations



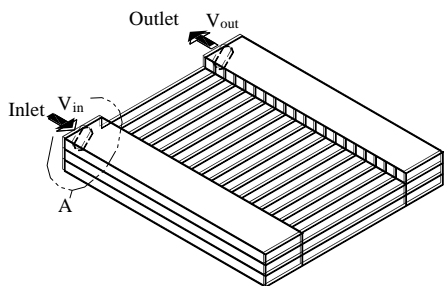
(a) I-arrangement



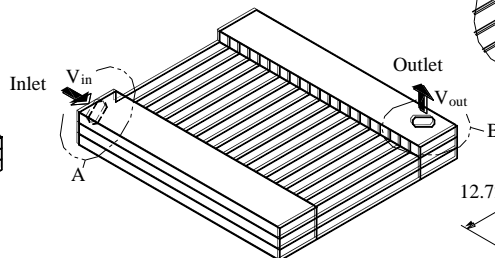
(b) Z-arrangement



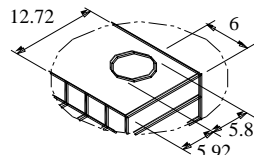
Details of A



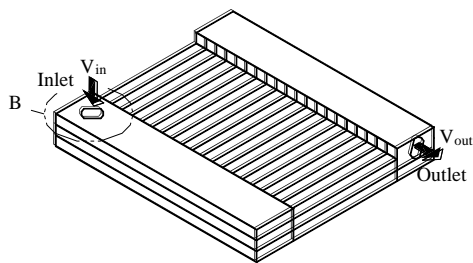
(c) J-arrangement



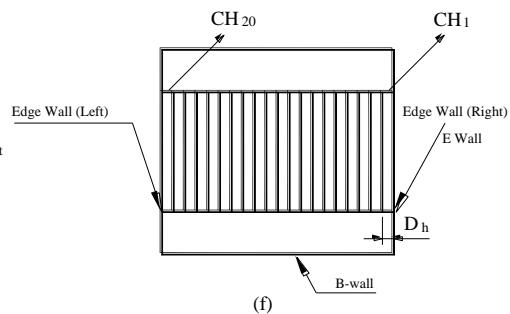
(d) L-arrangement



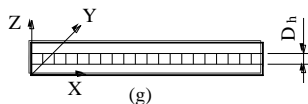
Details of B



(e) Γ -arrangement

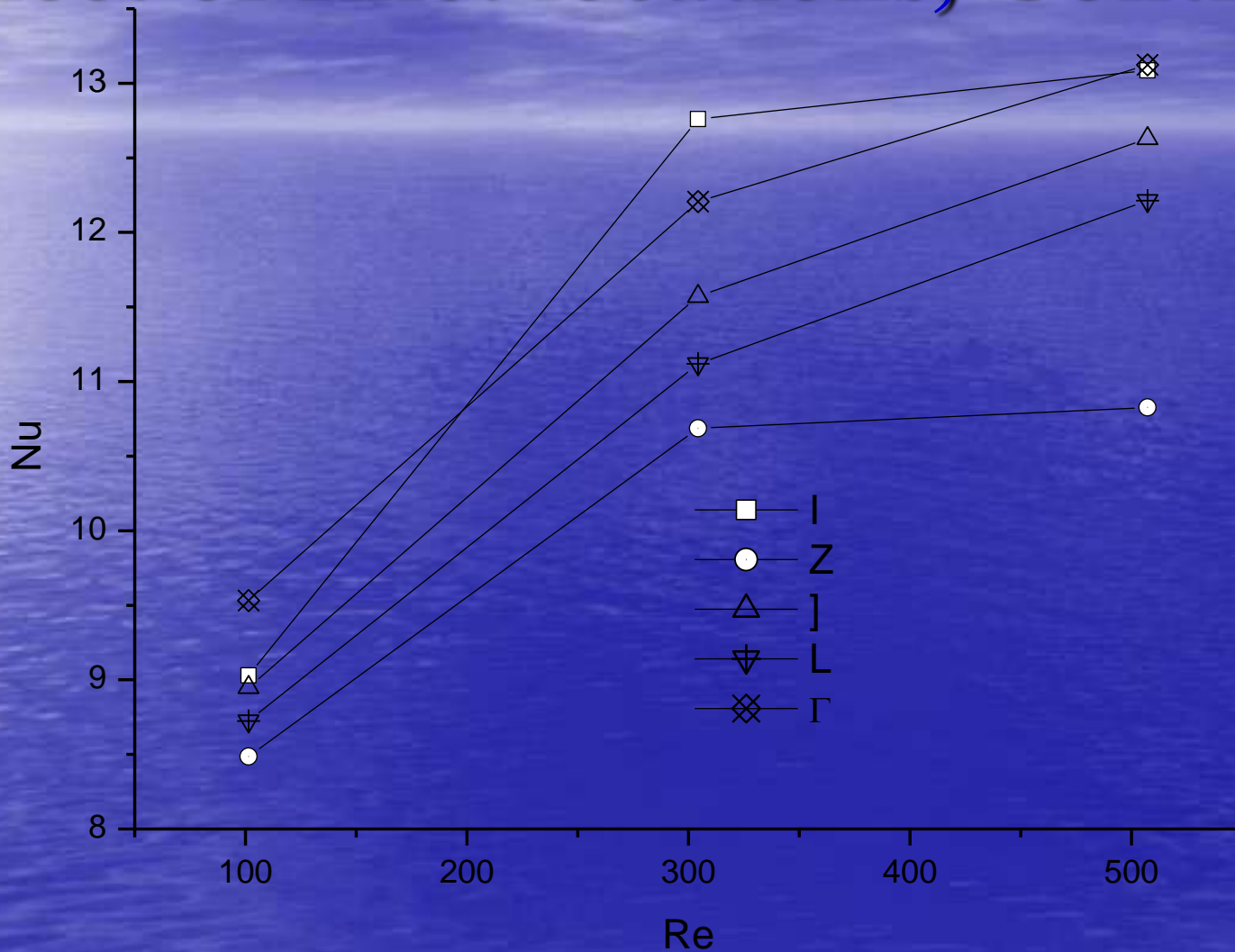


(f)

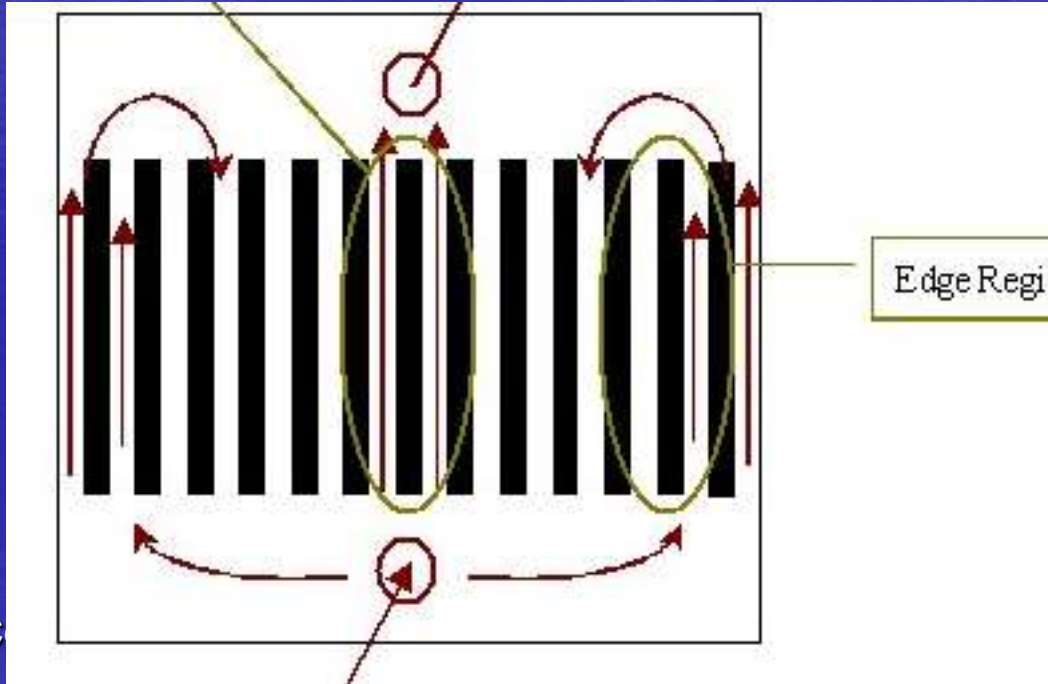
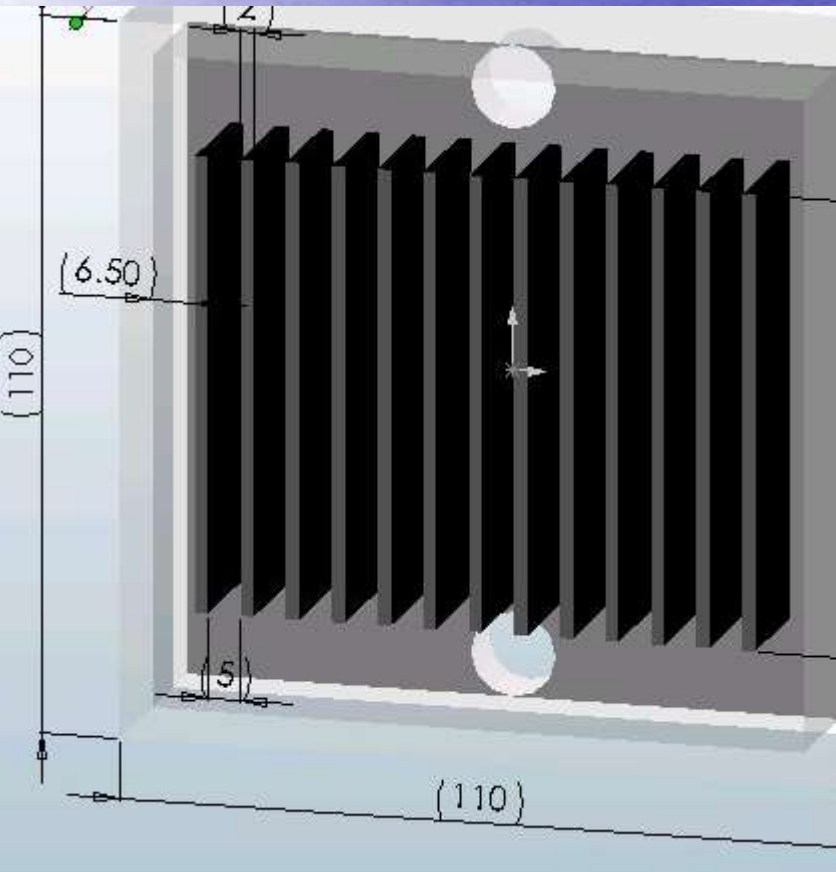
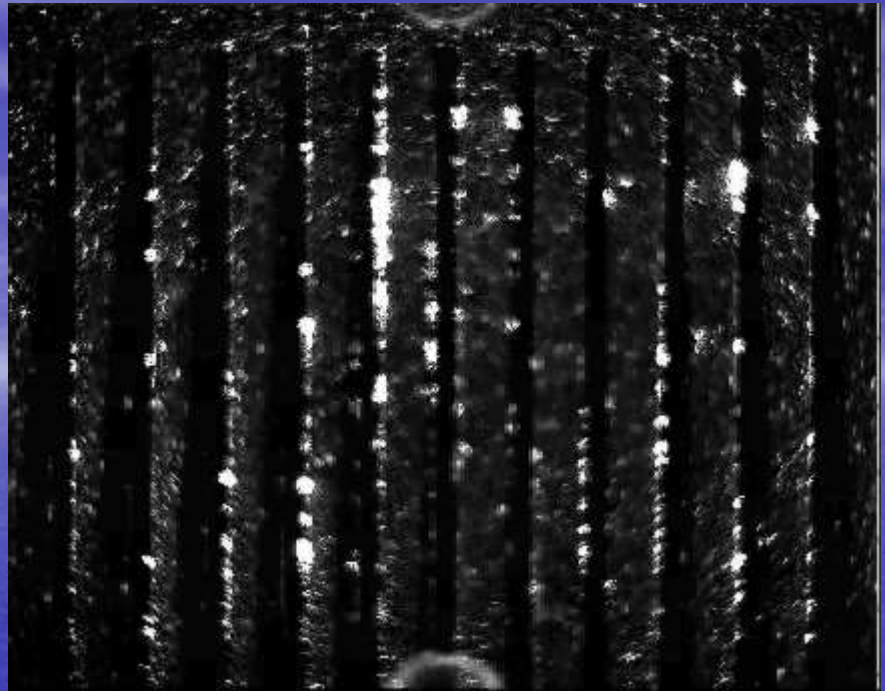


(g)

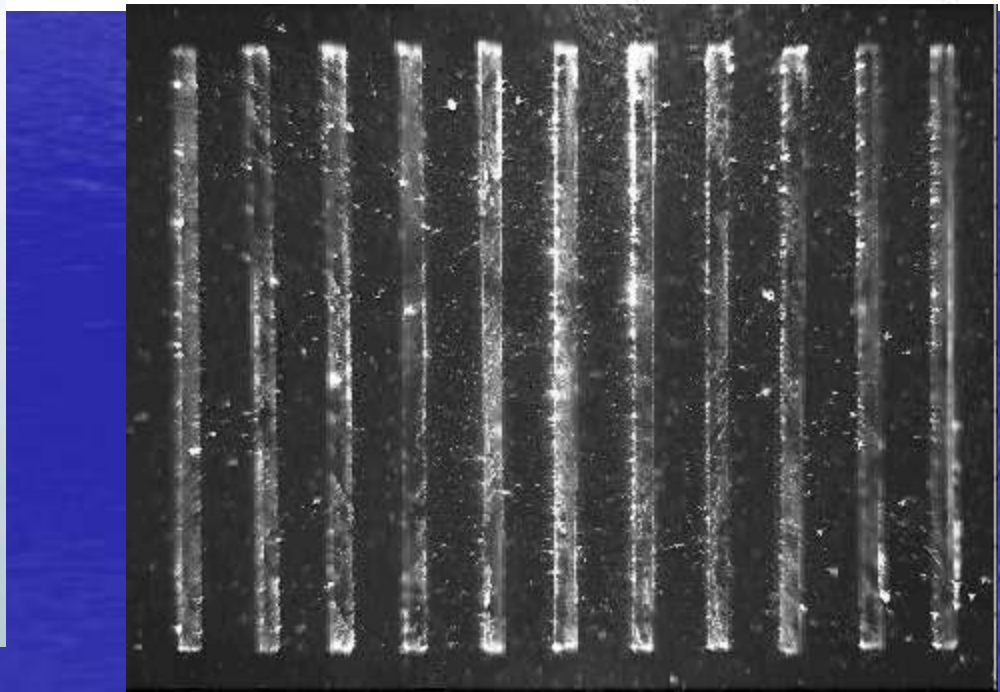
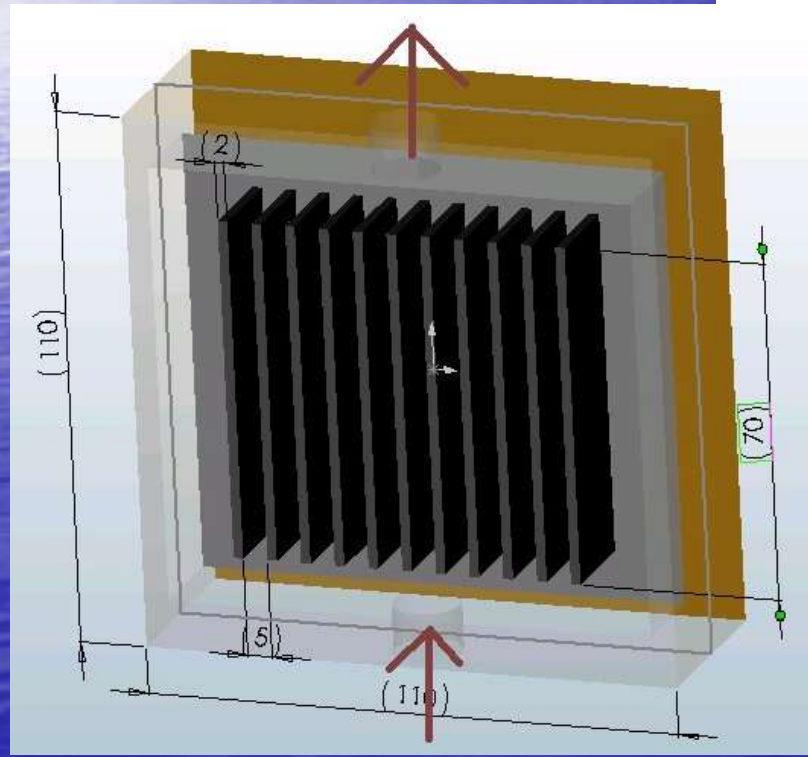
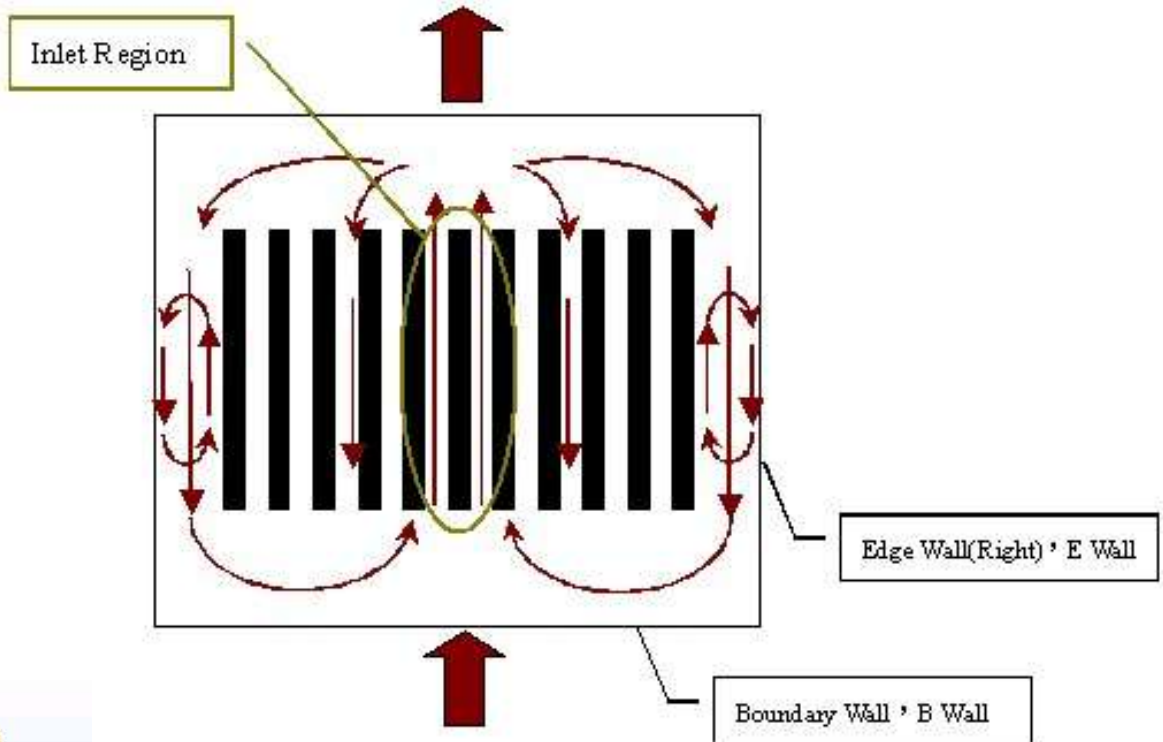
Effect of Inlet locations, Conti..



PIV Flow Visualization : I Arrangement – Uniform Gap

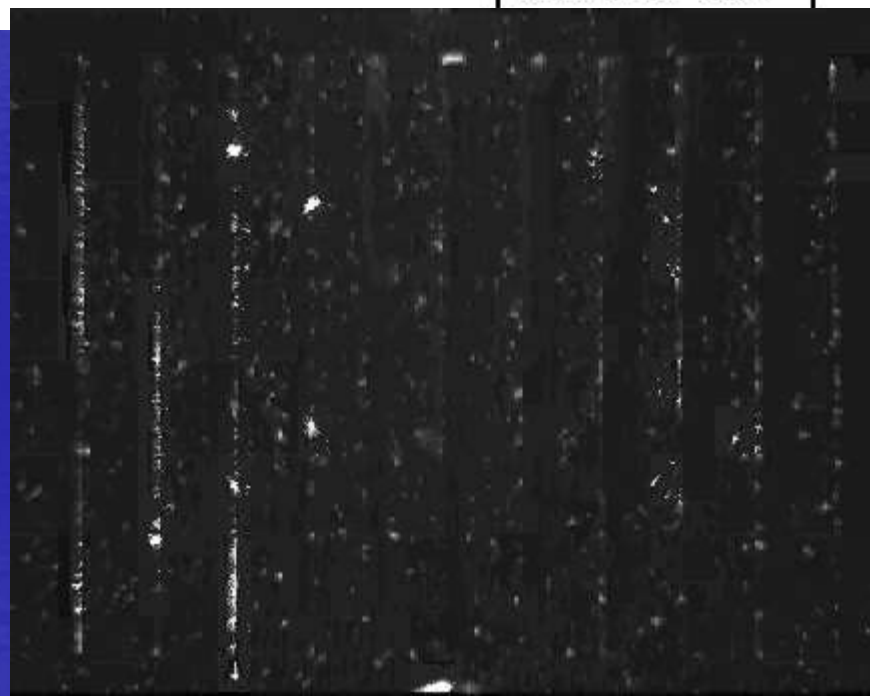
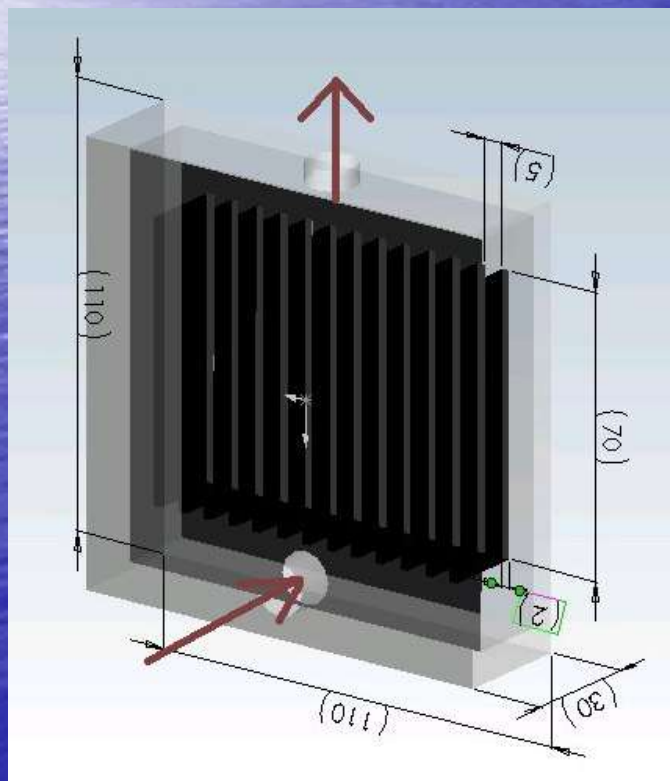
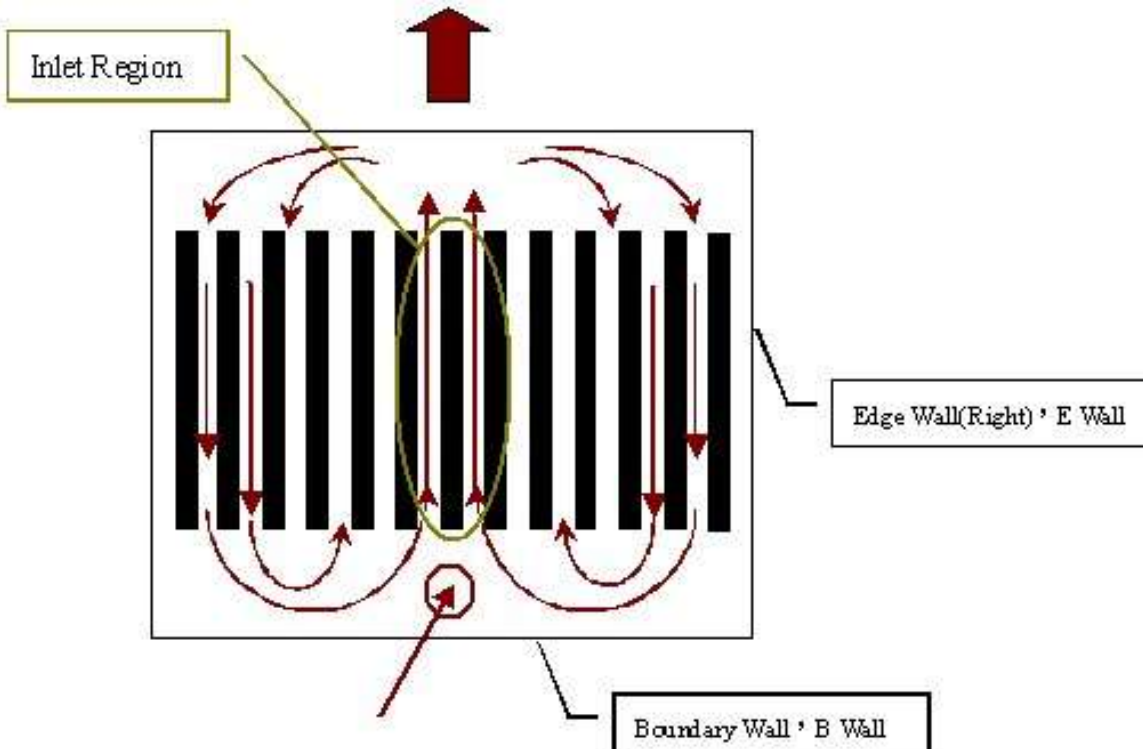


PIV Flow Visualization



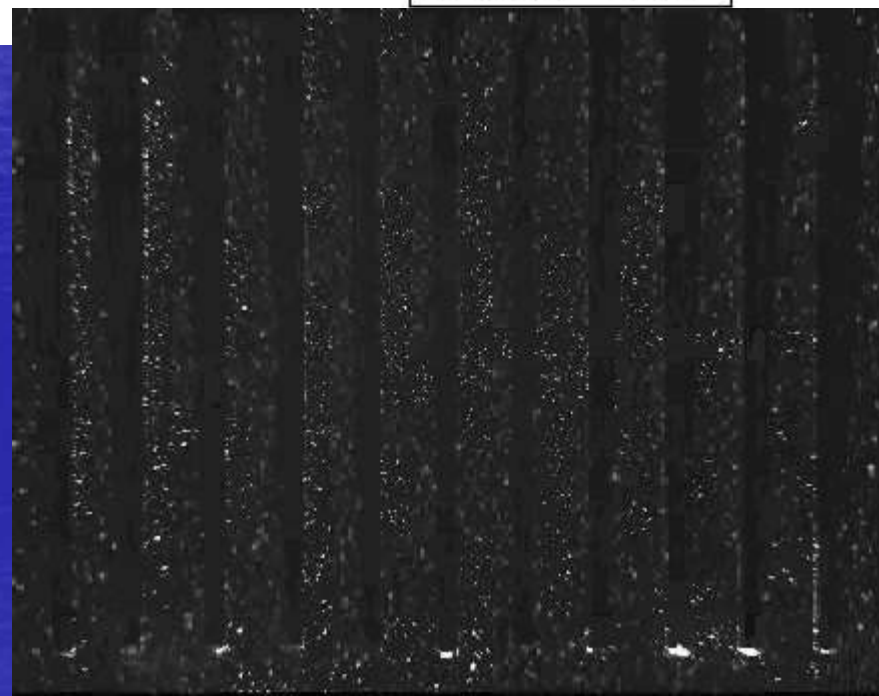
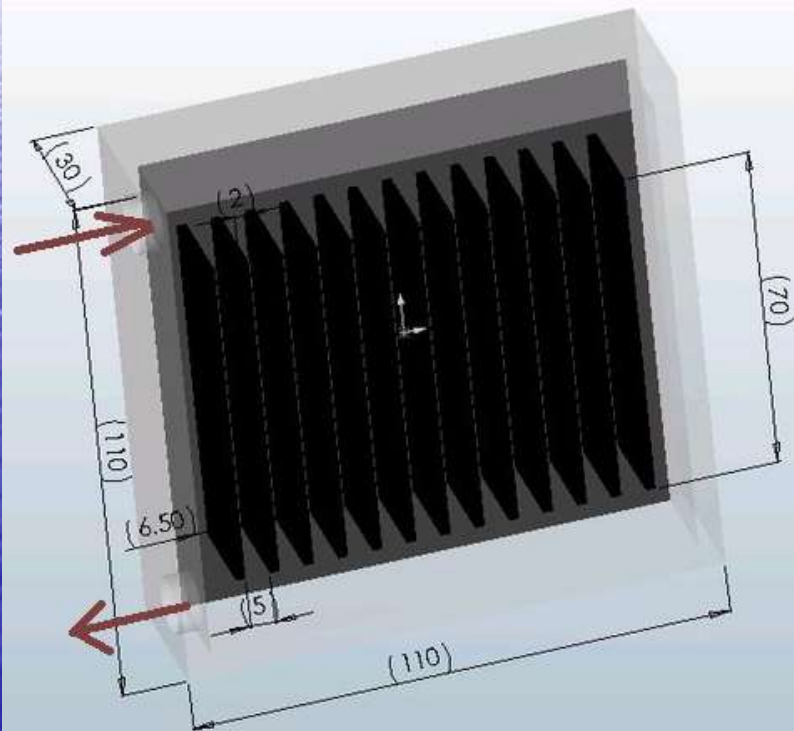
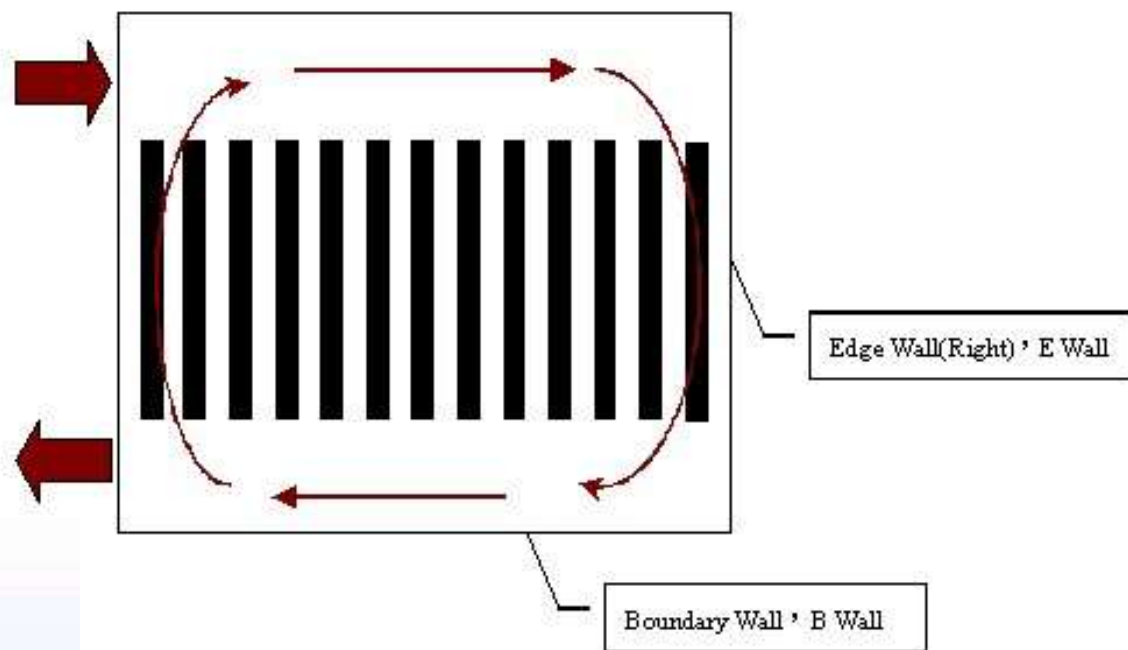


PIV Flow Visualization



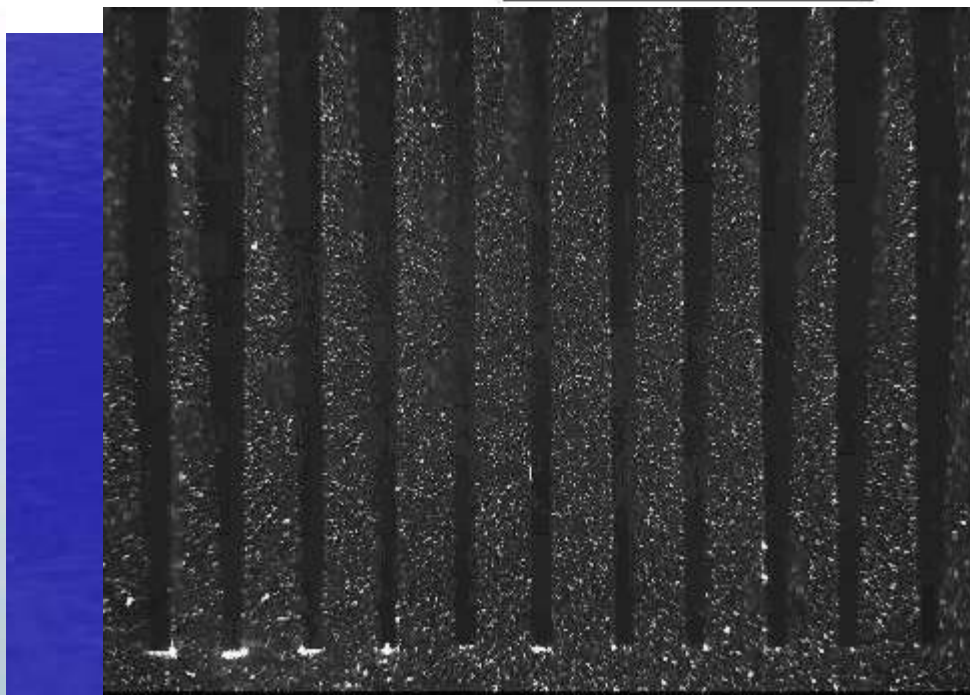
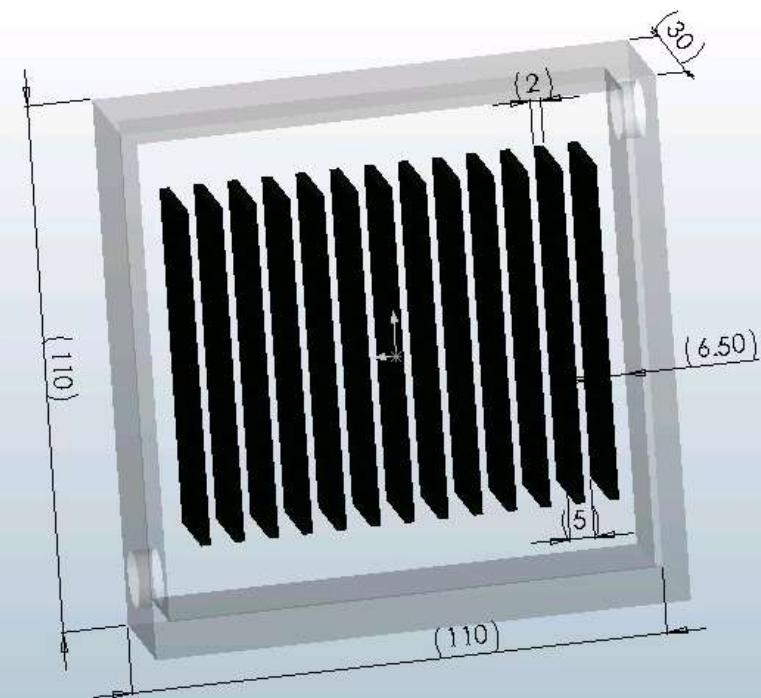
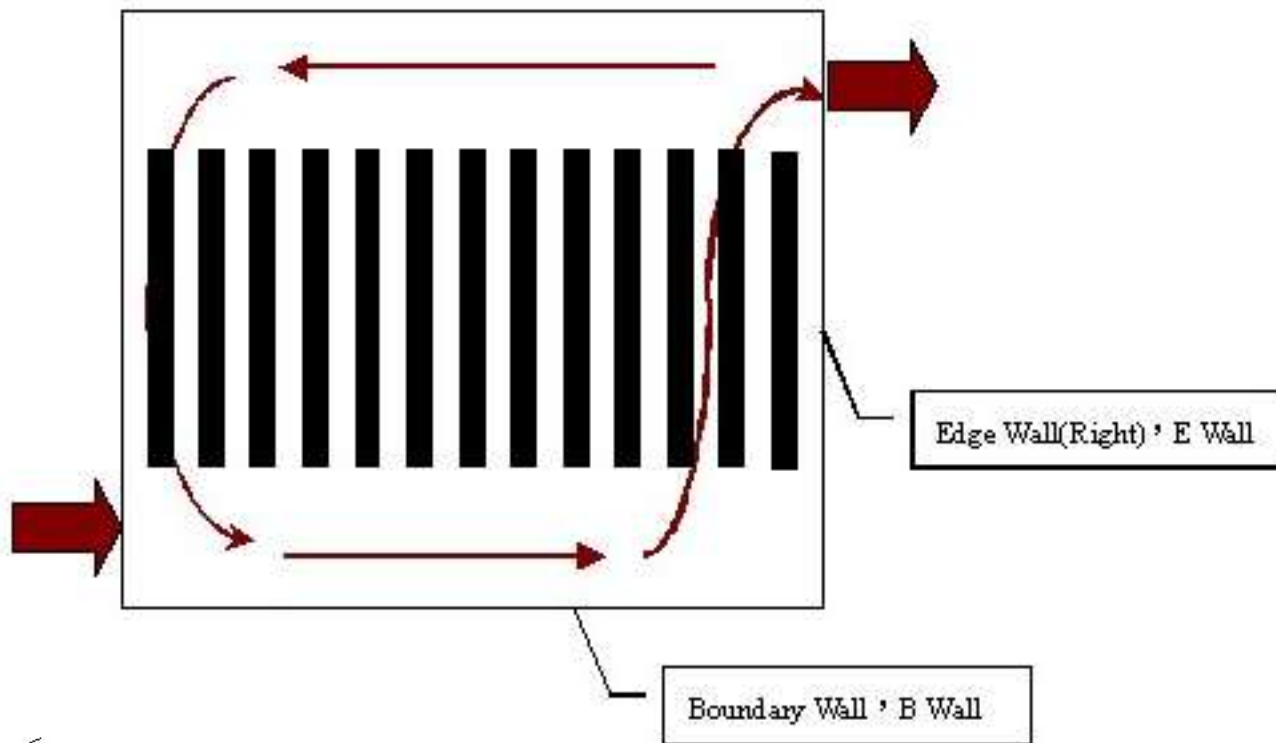


PIV Flow Visualization





PIV Flow Visualization





Coolermaster: AQUAGATE Mini:





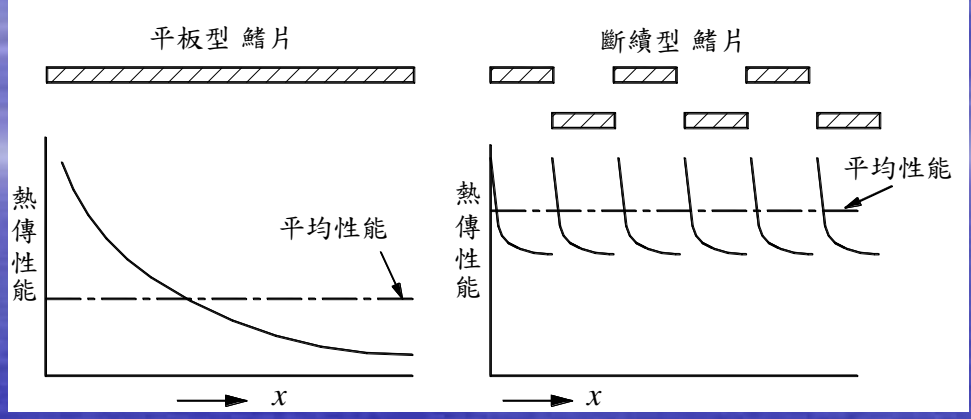
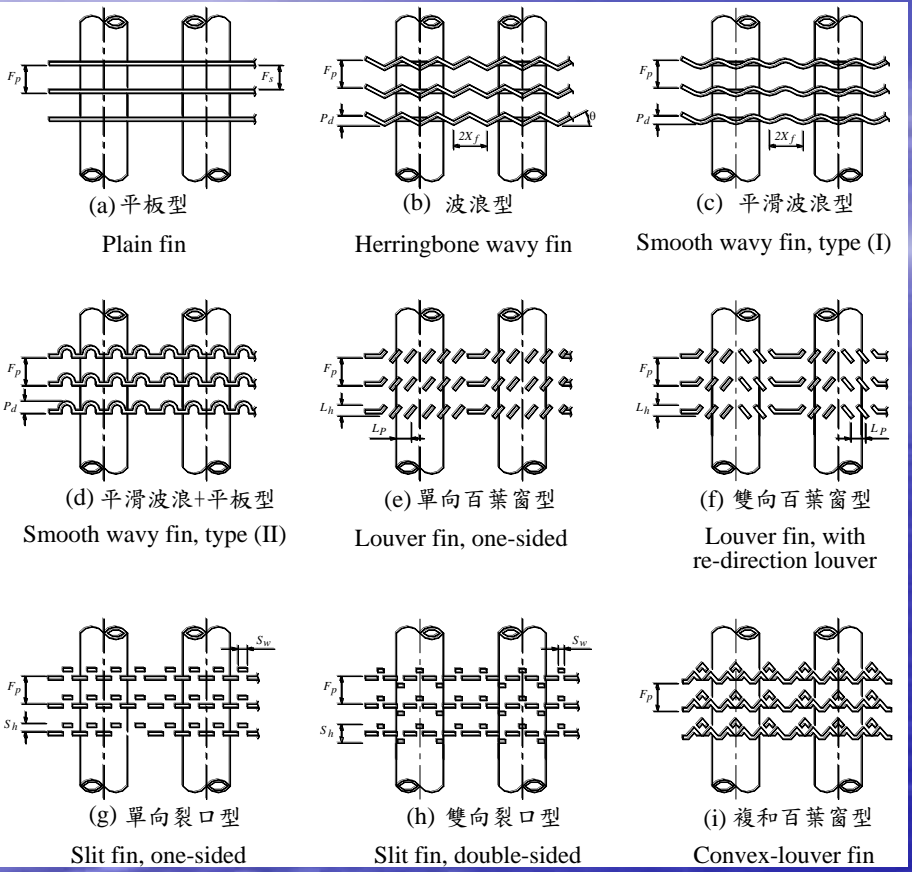
Radiator – air-cooled HX



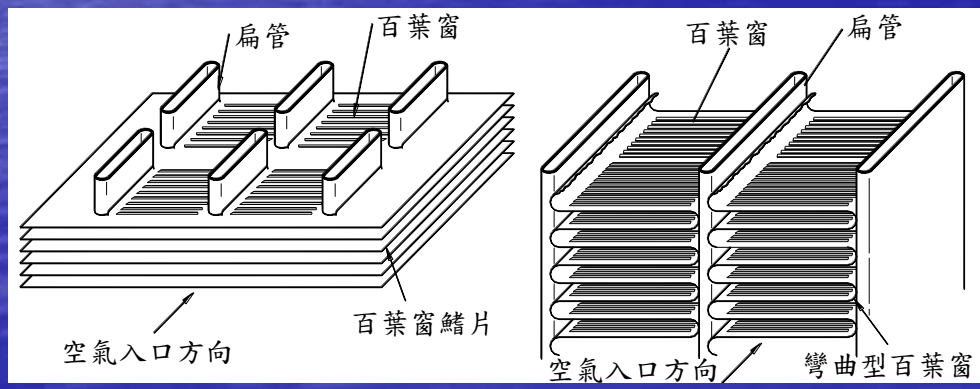
Copper Fin-and-tube HX



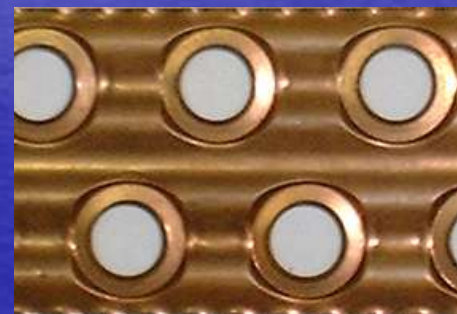
Aluminum Brazed Heat Exchanger



Various Fin Patterns



扁管式百葉窗型熱交換器



Various Fin Patterns

Note: ERL has the biggest database for all kinds fin patterns

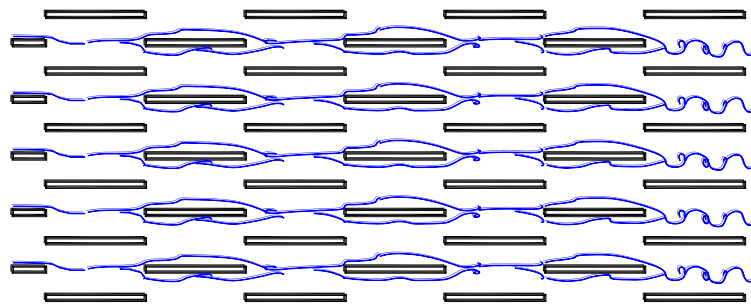
Design software are available (please call, Mr. J.S. Liaw at 03-5914220; jsliaw@itri.org.tw)



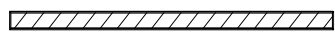
Passive method to improve airside performances - Technology Evolution

- Thermal Boundary Layer Restart
- Instability
- Thermal Wake Management
- Swirl

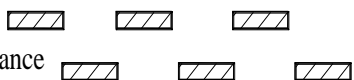
Air Flow



Plain fin - continuous fin

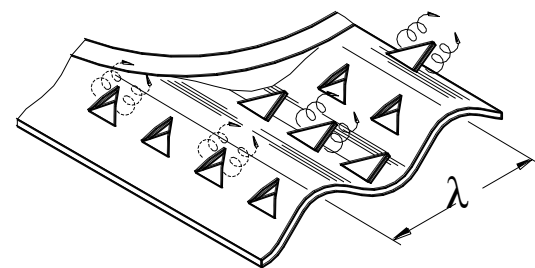
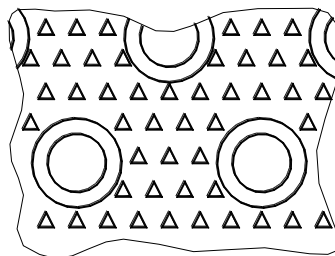
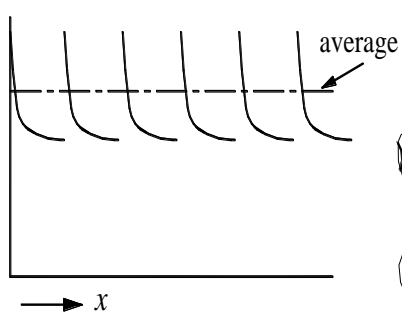
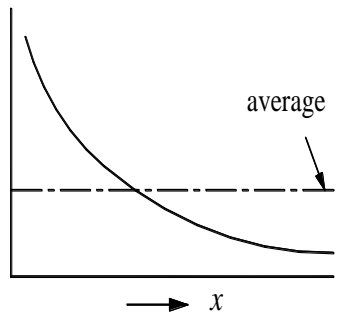


Interrupted surface



Performance

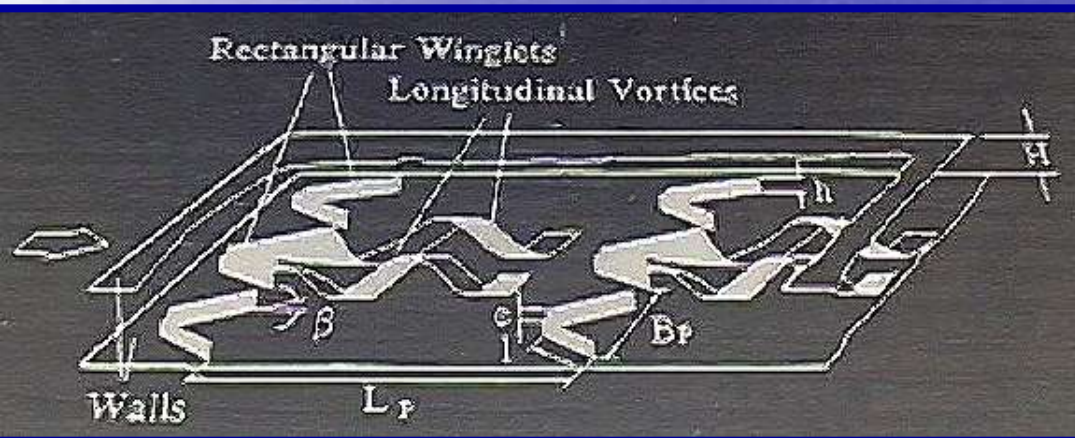
Performance



US patent 4817709

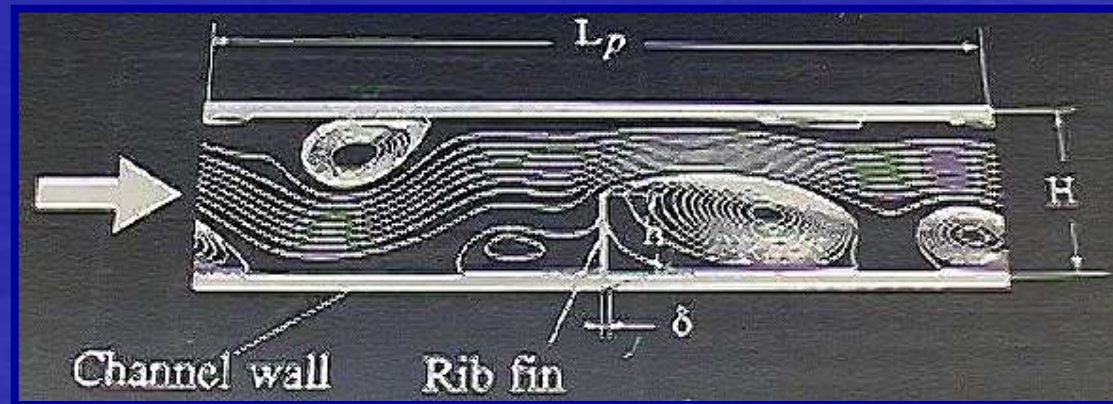
Type of vortex generators

Longitudinal vortex outperforms the transverse vortex



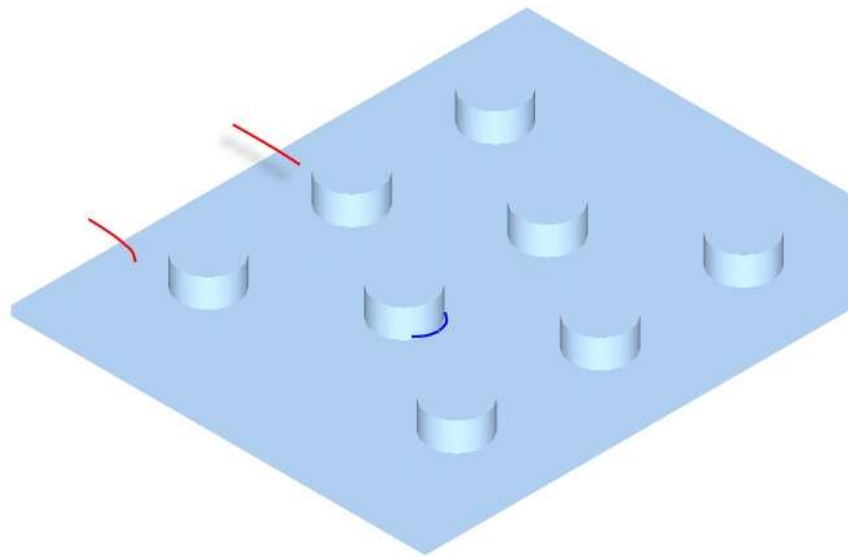
Longitudinal vortex

Transverse vortex

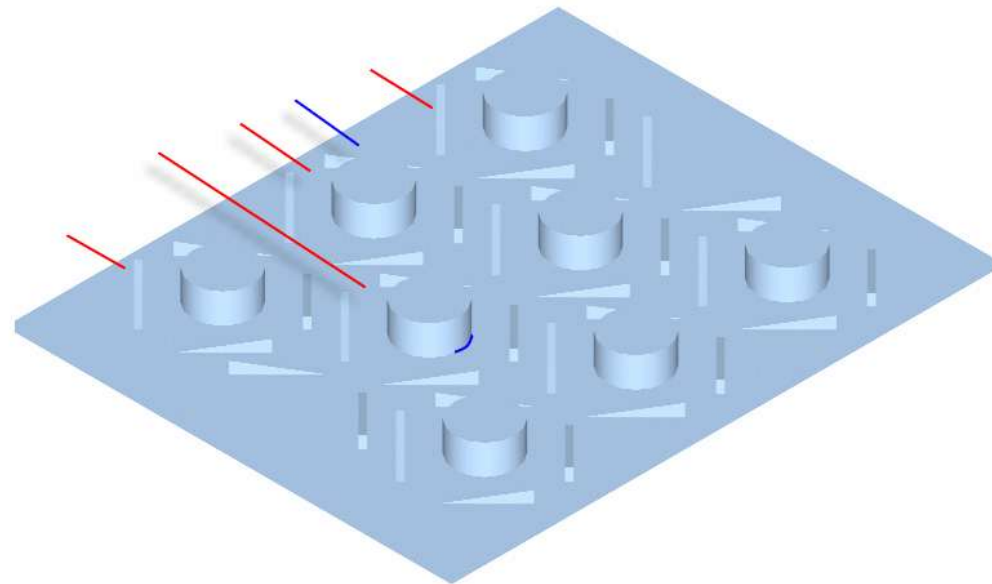


Benefits of vortex generator

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

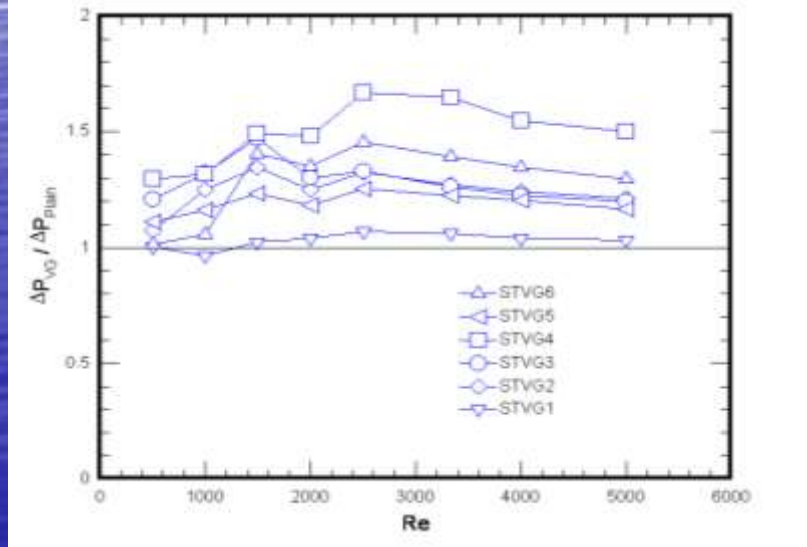
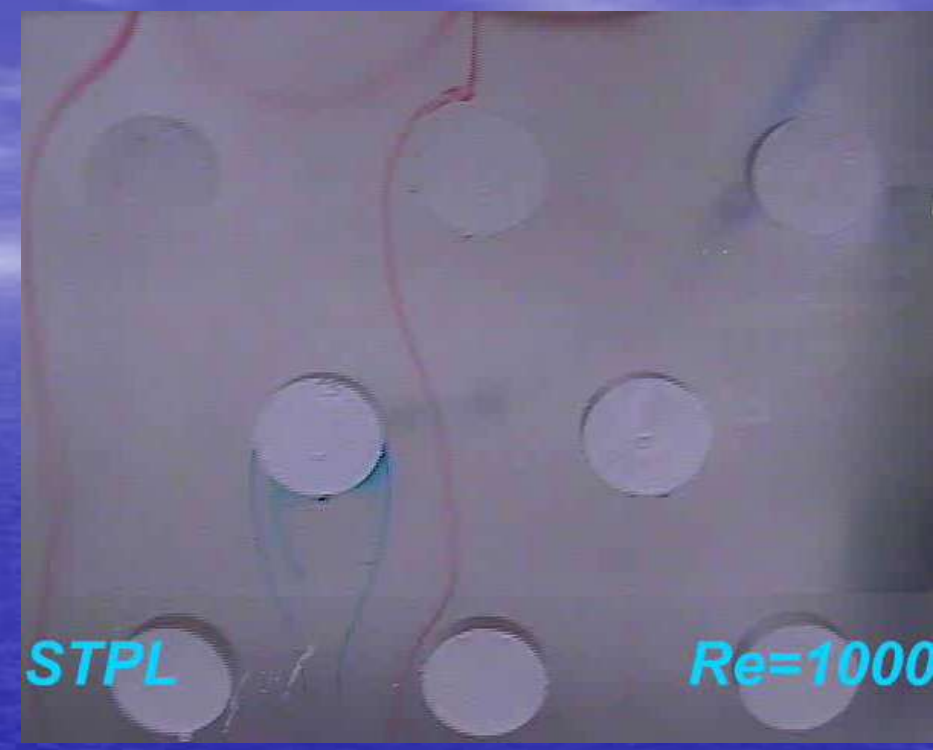
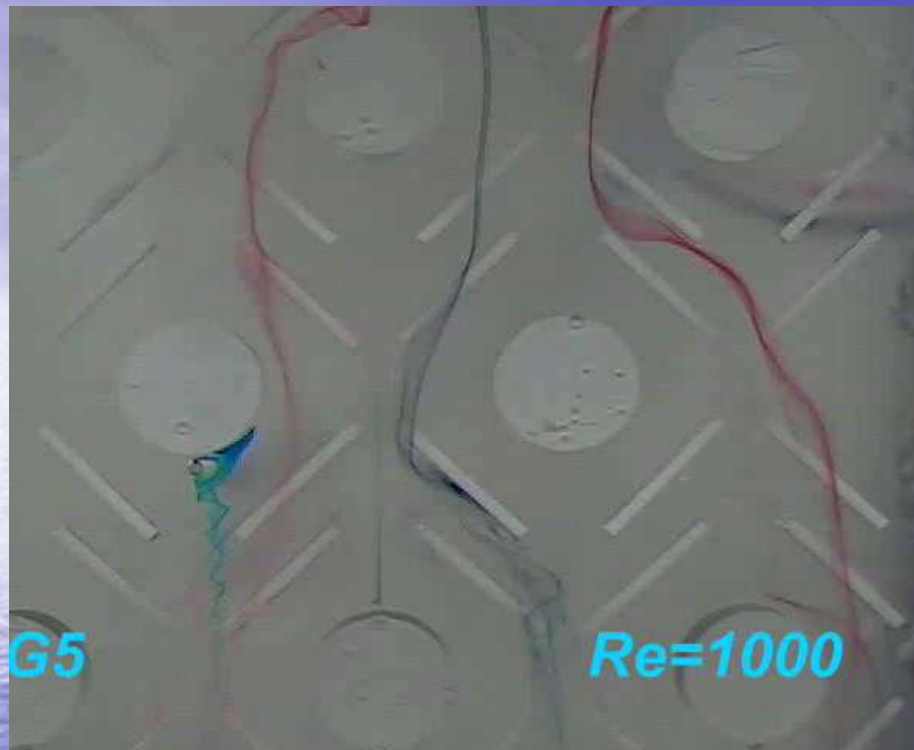


Re=1500,STPL



Re=1000,STVG5

Influence of vortex generator on flow field



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

Summary

- Liquid Cooling is considered as an alternative for high-flux electronic cooling applications.
- Micro-pumps is the key element of liquid cooling, reliability issue is the main concerns for those with valve designs while efficiency is for those valve-free design.
- Heat Transfer augmentation is an effective way for laminar flow cold-plate.
- Influence of entrance should be considered for microchannel HXs.
- Mal-distribution could be a concern for multiple port channel cold-plate.
- Radiator is the final place to dump heat – it is very crucial to choose suitable fin pattern .



Thanks for Your Attention
Questions?