氣冷式散熱鰭片技術

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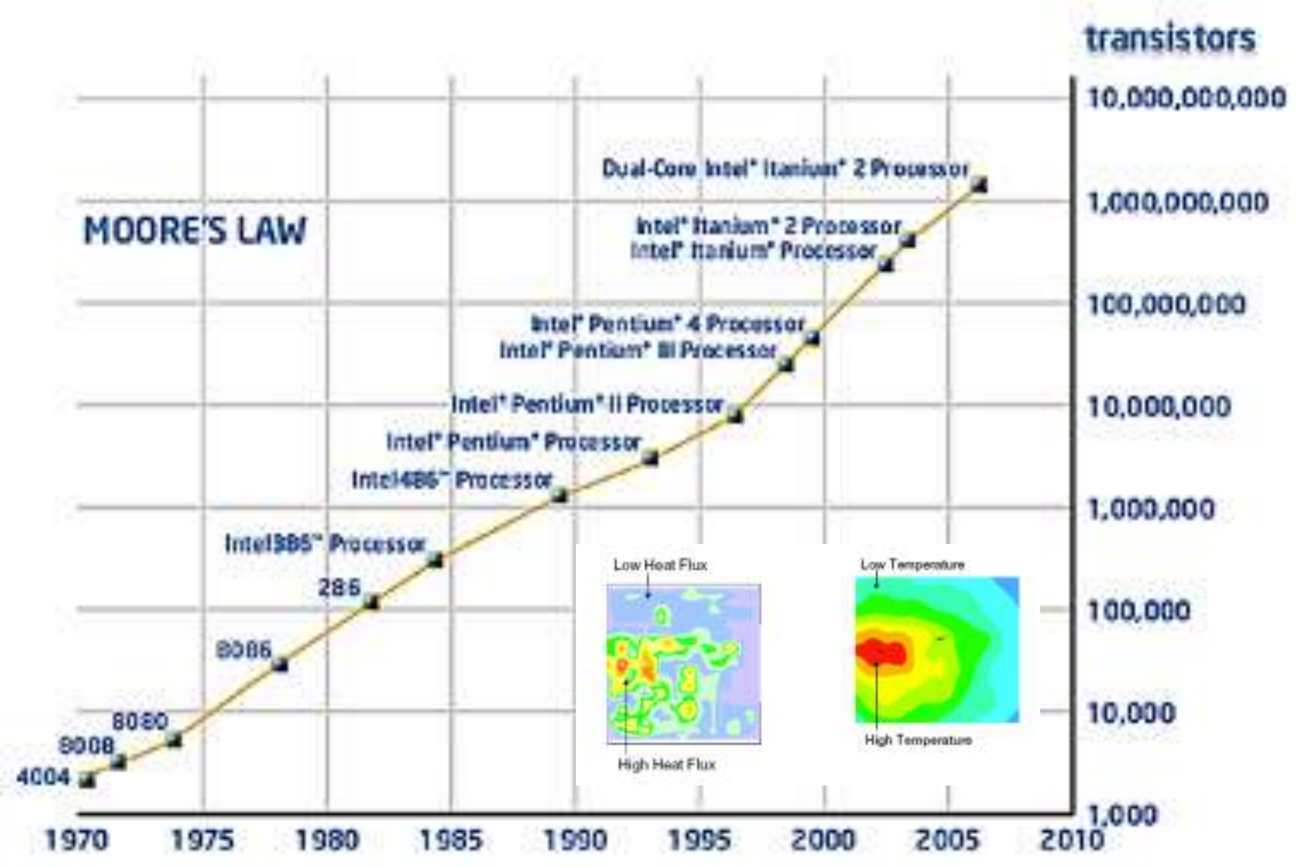
Acknowledgements

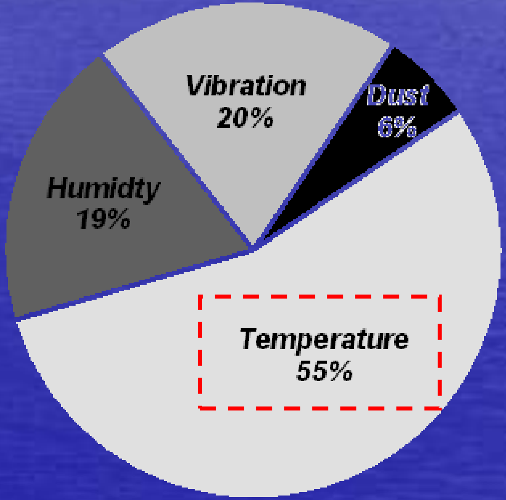
- Financial supports provided by the **Bureau of Energy** and **Department of Industrial Technology of Ministry of Economic Affairs,** R.O.C.
- Technical Supports from Miss CoCo HSu, Mr. M.C. Lu, Dr. K.S. Yang, Dr. K.H. Chien, Mr. R. T. Huang
- Advisory support from Dr. R.J. Shyu
- Major Collaboration Professors: Shyu W.T. (NTHU), Chen I.Y. (YunLin Tech U), Lin Y.T. (YZU)

Outline

- Fundamentals of fin design
- Passive enhancement technique
- Natural Convection vs. Forced Convection
- Influence of fin geometry
- Design by Non-uniformity
- Summary

Background

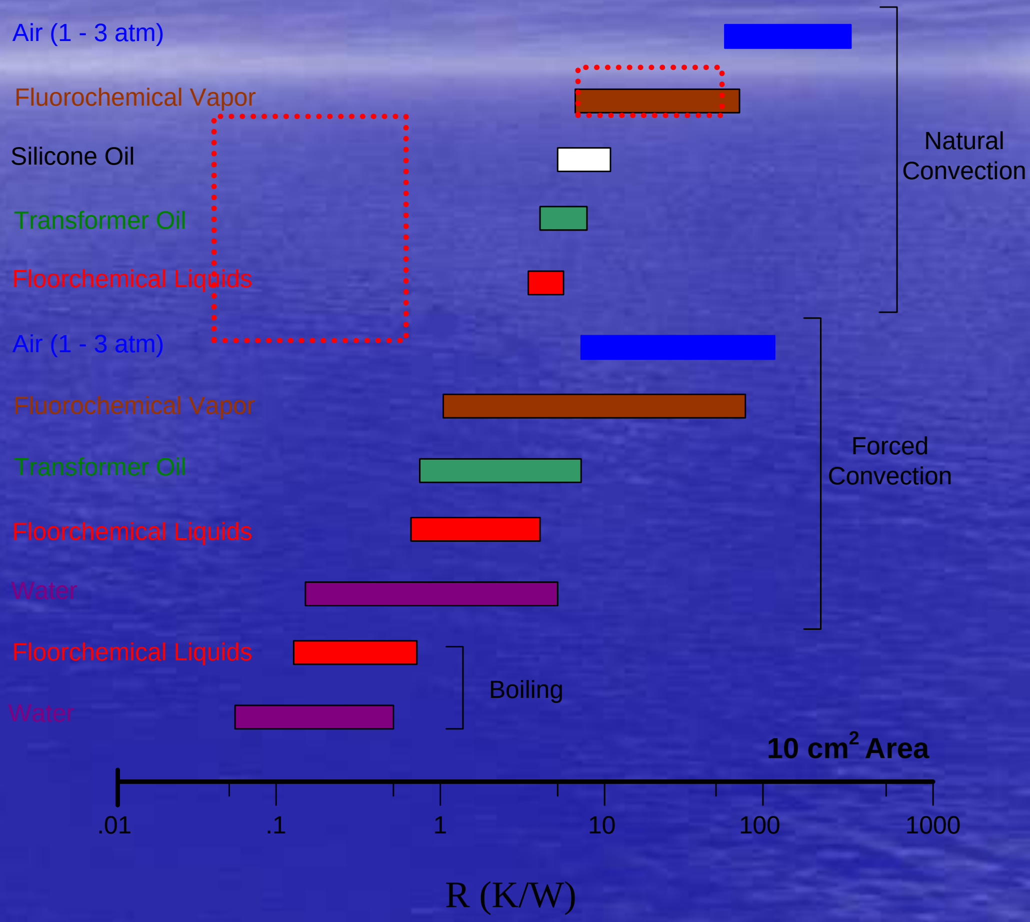




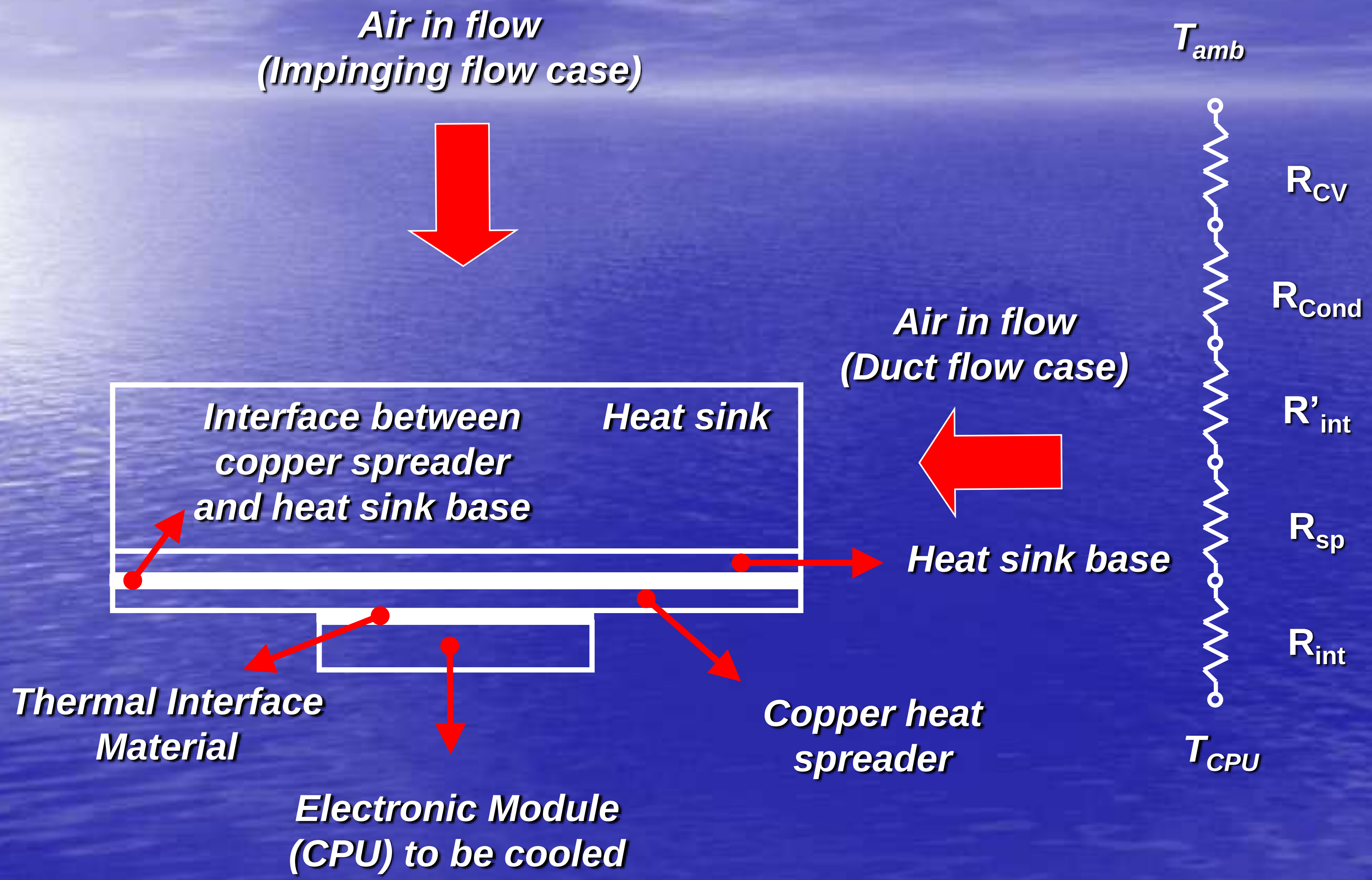
Source: *L. T. Yeh, ASME J. Electronic Packaging, vol.117, pp.333 –339, (1995).*

Background

- Electronic cooling
  - Air cooling
  - Liquid cooling
    - Single phase
    - Two-phase
  - Refrigeration
  - Thermoelectric
  - …
- Direct air-cooling is still the most popular way for its simplicity, reliability, and low cost.
- Major Problems for Air-cooling
  - Poor heat transfer characteristics
    - Increase A (fins) to increase heat transfer (higher pressure drop penalty)
  - Noisy
    - Reduce air flow rate



Thermal resistance network from CPU surface to ambient



Air in flow
(Impinging flow case)

T_{amb}

R_{CV}

R_{Cond}

Air in flow
(Duct flow case)

Interface between copper spreader and heat sink base

Heat sink

R'_{int}

Heat sink base

R_{sp}

R_{int}

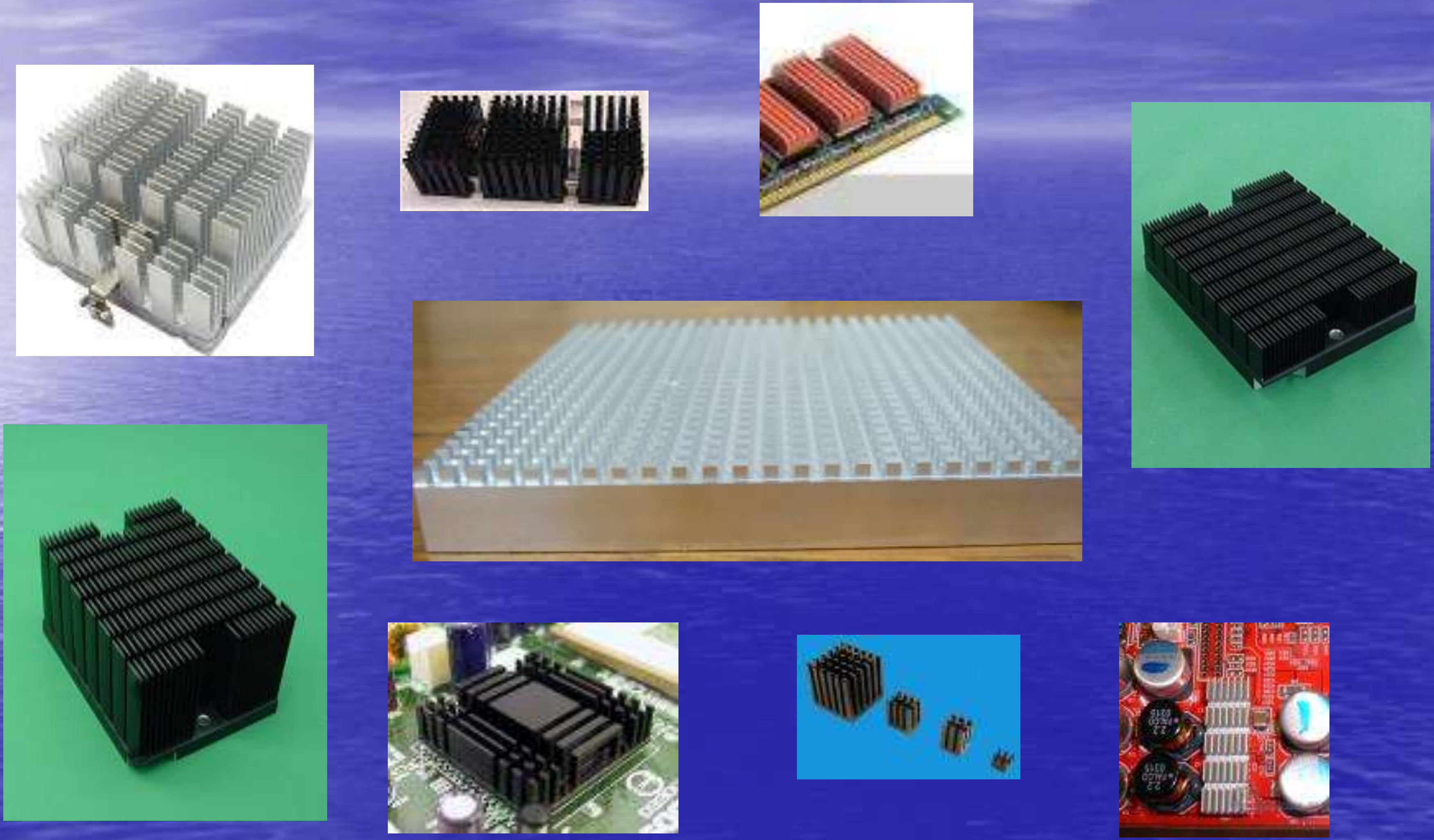
Thermal Interface Material

Copper heat spreader

T_{CPU}

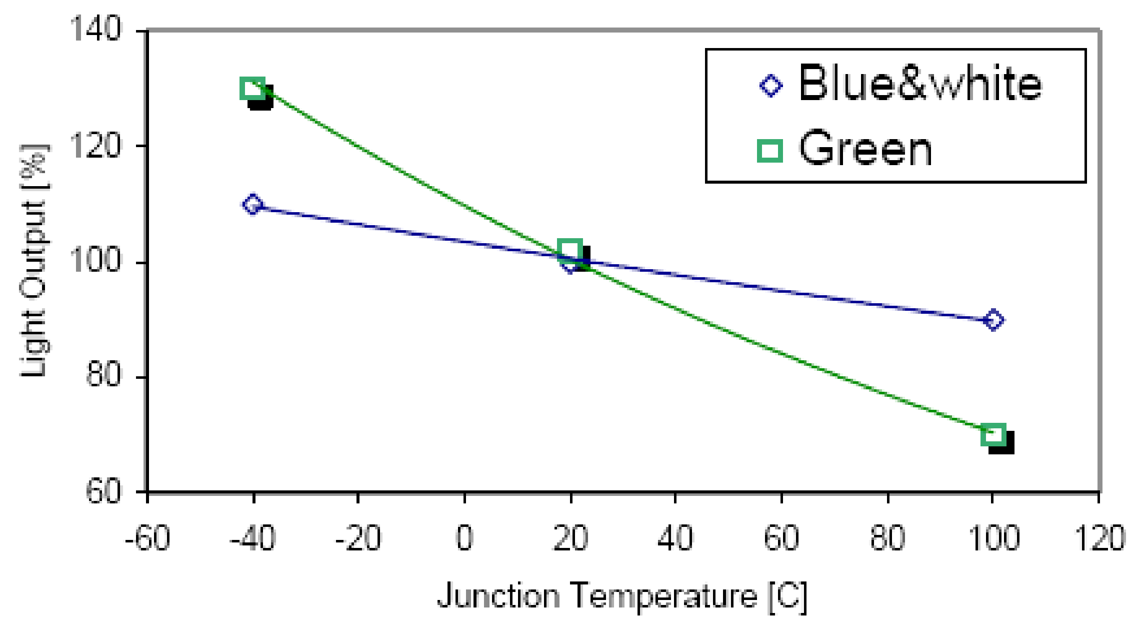
Electronic Module (CPU) to be cooled

Fundamentals – Passive Heat Sinks

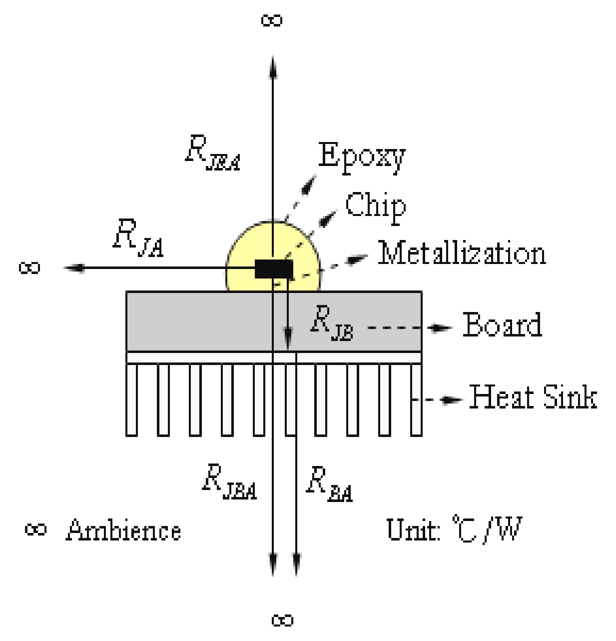


Why Natural Convection?

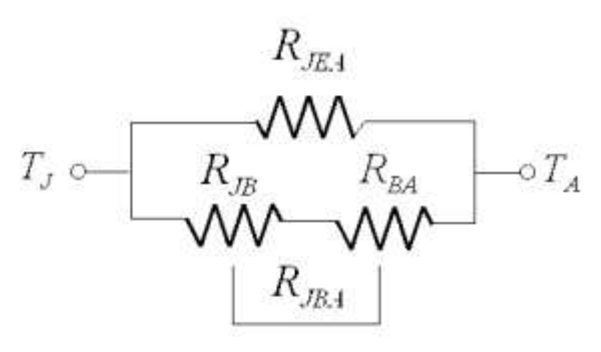
- Natural Convection is a noise-free and power-free thermal management method
- Under a junction temperature of 120 °C, white LEDs exhibit exceptional lifetime, exceeding 50000 hrs..



Variation of the light output with the junction temperature



A LED unit



Simplified resistance network.

$$\frac{1}{R_{JA}} = \cancel{\frac{1}{R_{JEA}}} + \frac{1}{R_{JBA}}$$

$$R_{JA} = R_{JBA} = R_{JB} + R_{BA}$$

J: junction
E: epoxy
B: board
A: ambience
cont: contact
hs: heat sink
conv: convection

R_{JB} and R_{BA} are of comparable magnitude

Our task : Minimize R_{BA}

$$R_{BA} = R_{cont} + R_{hs} + R_{conv}$$

Heat Transfer Engineering, 26(2):50–53, 2005

The Effect of Plate Size on the Natural Convective Heat Transfer Intensity of Horizontal Surfaces

EWA RADZIEMSKA and WITOLD M. LEWANDOWSKI

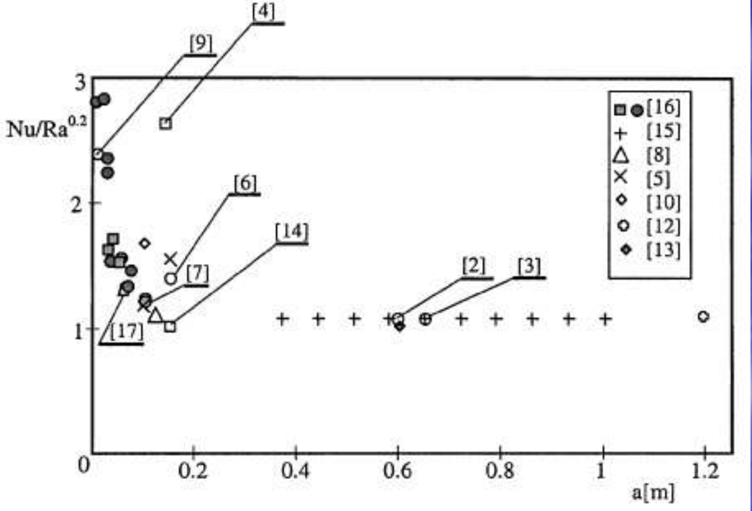
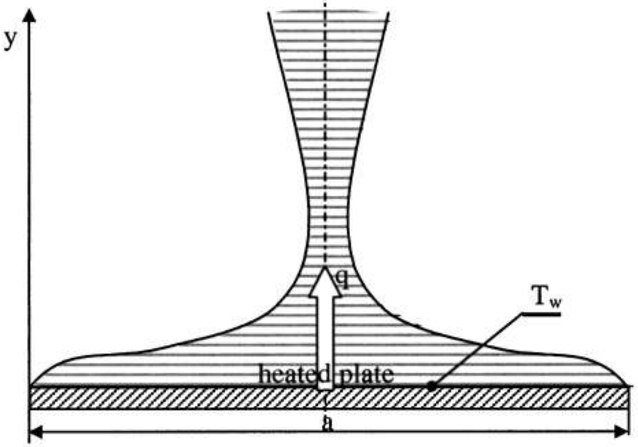
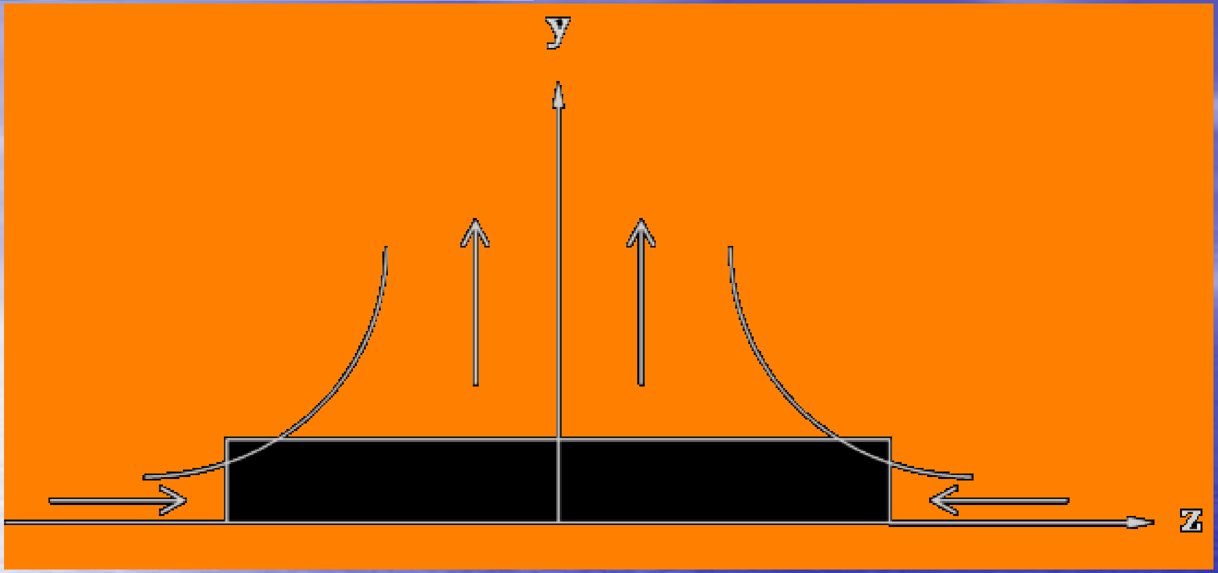


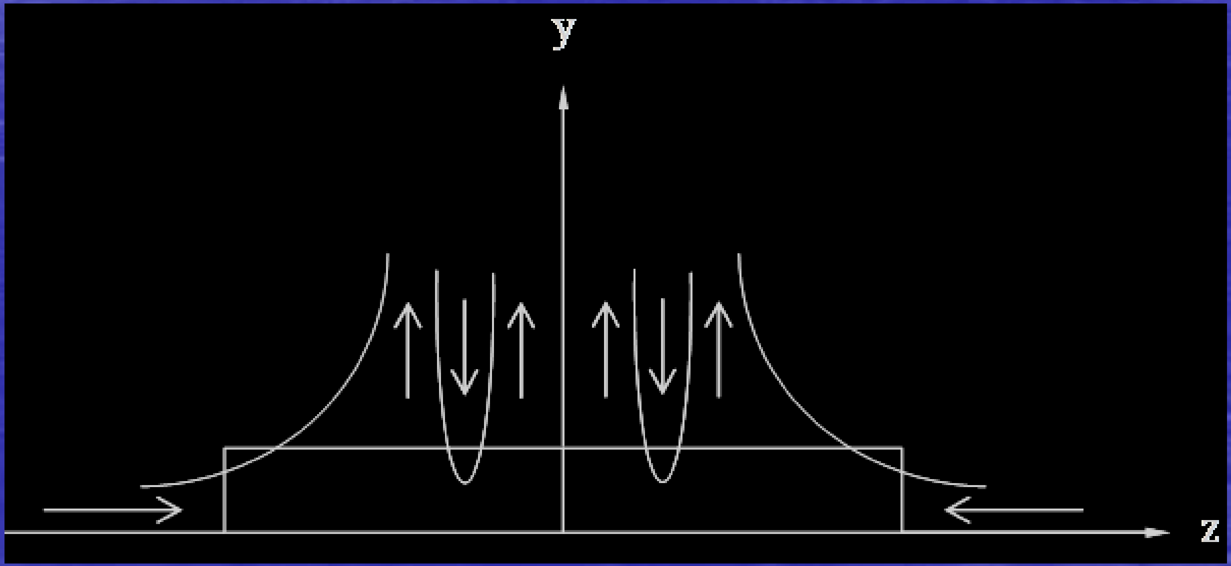
Table 1 Natural convection correlations for isothermal horizontal surfaces

Authors	Correlation	Converted correlations for $Ra = 10^6$ (for the purpose of comparison)	Width of the plate (a), m	Length of the plate (L), m	Fluid	Ra range
Al-Arabi and Sakr [3]	$Nu = 0.54(GrPr)^{0.25}$	$Nu = 1.077(GrPr)^{0.2}$	0.65	1.3	Air	$10^4 < Ra < 10^7$
Fishenden and Saunders [2]	$Nu = 0.54Ra^{0.25}$	$Nu = 1.077(GrPr)^{0.2}$	0.61	0.61	Air	$10^5 < Ra < 10^7$
Sharma and Adelman [4]	$Nu = 0.782Ra^{0.288}$	$Nu = 2.638Ra^{0.2}$	0.212	0.141	Water	$1.79 \times 10^5 < Ra < 10^9$
Reilly et al. [5]	$Nu = 0.591Ra^{0.25}$	$Nu = 1.179Ra^{0.2}$	0.2	0.098	—	—
Al-Arabi and El-Riedy [6]	$Nu = 0.17Ra^{0.36}$	$Nu = 1.55Ra^{0.2}$	0.302	0.149	—	—
Yousef et al. [7]	$Nu = 0.7Ra^{0.25}$	$Nu = 1.397Ra^{0.2}$	0.15	0.25–0.6	Air	$2 \times 10^5 < Ra < 10^9$
Lloyd and Moran [8]	$Nu = 0.622Ra^{0.25}$	$Nu = 1.2Ra^{0.2}$	0.1	0.1	Air	$3 \times 10^6 < Ra < 4 \times 10^7$
Yang et al. [9]	$Sh = 0.54Ra^{0.25}$	$Nu = 1.077Ra^{0.2}$	0.127	0.127	Electrolyte	$2.2 \times 10^4 < Ra < 8 \times 10^6$
Wilkes and Peterson [10]	$Nu = 0.125Ra^{0.25}$	$Nu = 2.245Ra^{0.2}$	0.025	0.025	Air	$10^4 < Ra < 10^7$
Griffith and Davis [12]	$h = 5.063(\Delta T)^{0.12}$	$Nu = 1.669Ra^{0.2}$	0.1	—	Air	—
Giesecke [13]	$h = 2.63(\Delta T)^{0.25}$	$Nu = 1.116Ra^{0.2}$	1.2	—	Air	—
Fujii and Imura [14]	$h = 3.158(\Delta T)^{0.25}$	$Nu = 1.015Ra^{0.2}$	0.6	—	Air	—
Michiejew [15]	$Nu = 0.16Ra^{1/3}$	$Nu = 1.01Ra^{0.2}$	0.15	0.3	Water	—
Radziemska	$Nu = 0.54(GrPr)^{0.25}$	$Nu = 1.077Ra^{0.2}$	0.25–1.0	—	—	$10^4 < Ra < 10^7$
and Lewandowski [16]	$Nu = 1.71Ra^{0.2}$	$Nu = 1.71Ra^{0.2}$	0.04	0.1	Water	$10^5 < Ra < 10^8$
	$Nu = 1.519Ra^{0.2}$	$Nu = 1.519Ra^{0.2}$	0.05	0.1	Water	$10^5 < Ra < 10^8$
	$Nu = 1.219Ra^{0.2}$	$Nu = 1.219Ra^{0.2}$	0.1	0.1	Water	$10^5 < Ra < 10^8$
	$Nu = 2.802Ra^{0.2}$	$Nu = 2.802Ra^{0.2}$	0.0048	0.1	Air	$10^4 < Ra < 10^7$
	$Nu = 2.393Ra^{0.2}$	$Nu = 2.393Ra^{0.2}$	0.0092	0.1	Air	$10^4 < Ra < 10^7$
	$Nu = 1.216Ra^{0.2}$	$Nu = 1.216Ra^{0.2}$	0.1	0.1	Air	$10^4 < Ra < 10^7$
	$Nu = 2.363Ra^{0.2}$	$Nu = 2.363Ra^{0.2}$	0.025	0.1	Air	$10^4 < Ra < 10^7$
	$Nu = 2.836Ra^{0.2}$	$Nu = 2.836Ra^{0.2}$	0.204	0.1	Air	$10^4 < Ra < 10^7$
	$Nu = 1.556Ra^{0.2}$	$Nu = 1.556Ra^{0.2}$	0.0531	0.1	Air	$10^4 < Ra < 10^7$
	$Nu = 1.463Ra^{0.2}$	$Nu = 1.463Ra^{0.2}$	0.0714	0.1	Air	$10^4 < Ra < 10^7$
	$Nu = 1.531Ra^{0.2}$	$Nu = 1.531Ra^{0.2}$	0.033	0.1	Air	$10^4 < Ra < 10^7$
	$Nu = 1.337Ra^{0.2}$	$Nu = 1.337Ra^{0.2}$	0.0659	0.1	Air	$10^4 < Ra < 10^7$
Lewandowski et al. [17]	$Nu = 1.228Ra^{0.2}$	$Nu = 1.228Ra^{0.2}$	0.07	0.1	Air	$10^5 < Ra < 10^8$



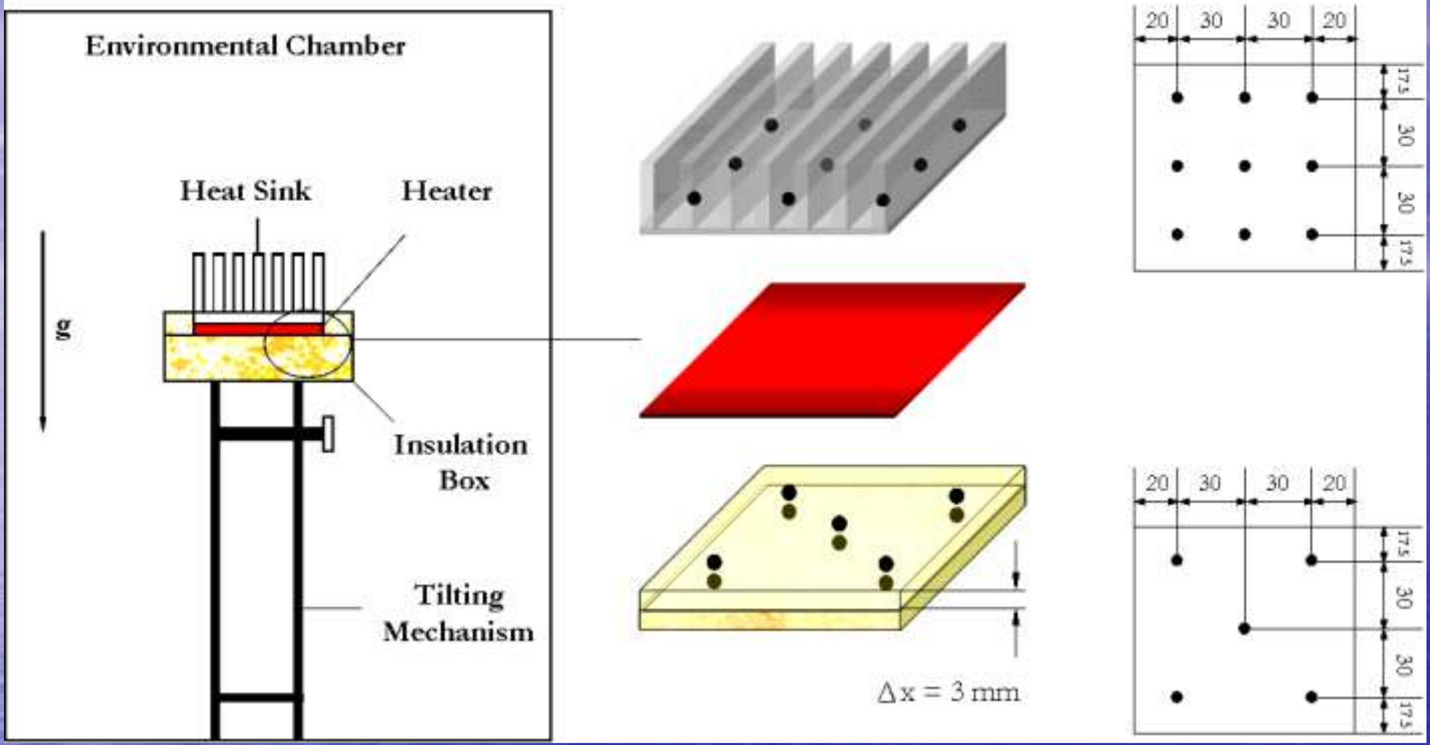


(a) Single chimney mode flow pattern.

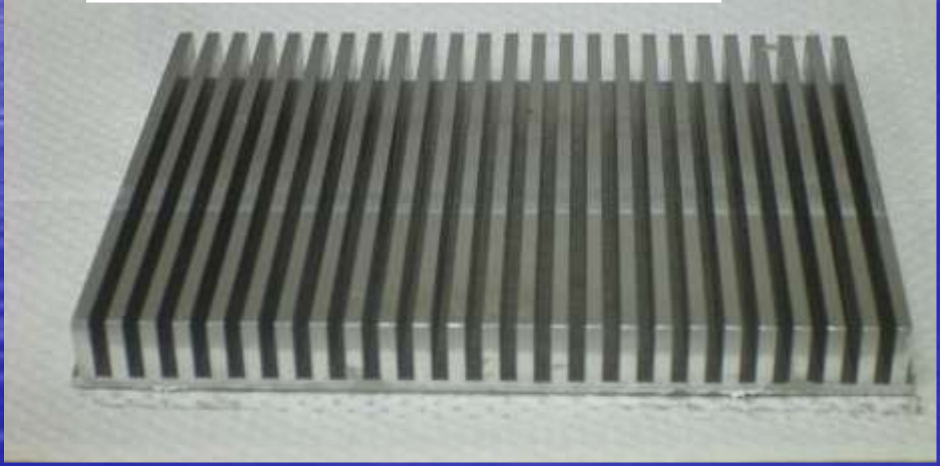


(b) Multiple chimney mode flow pattern.

Experimental Setup

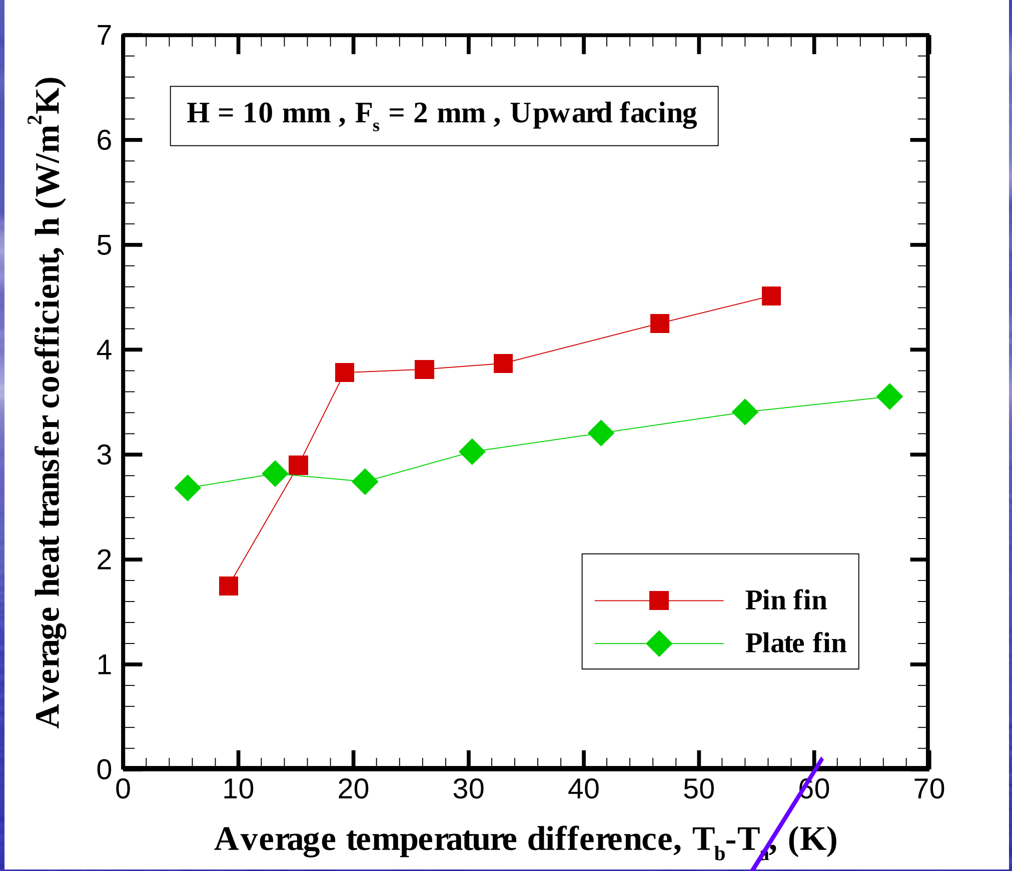


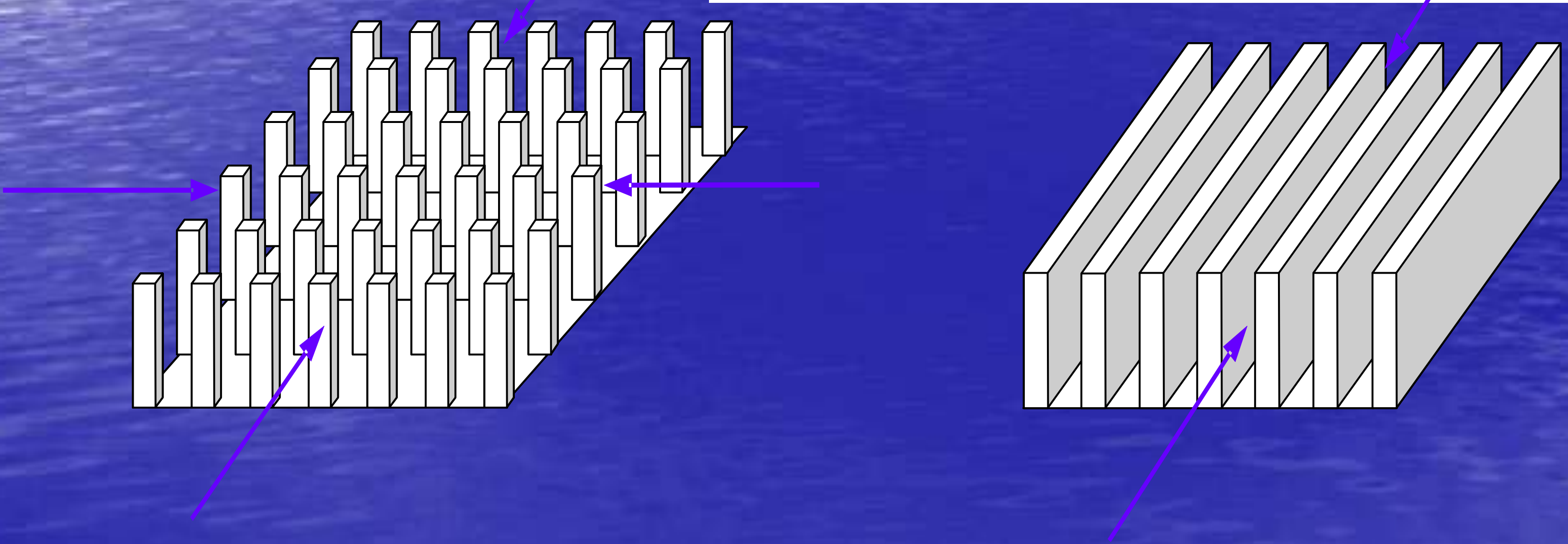
Test Samples: Plate Fin

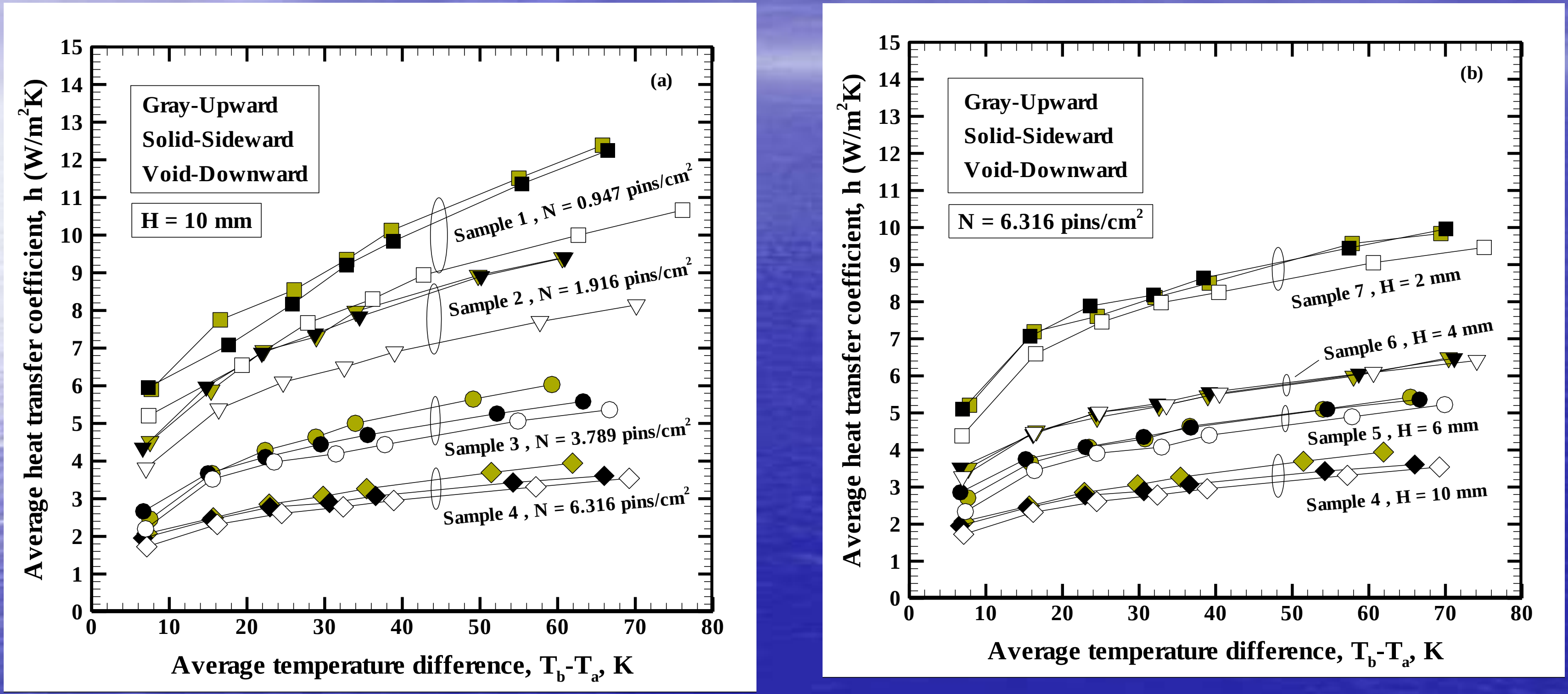


Test Samples: Pin Fin (Square)

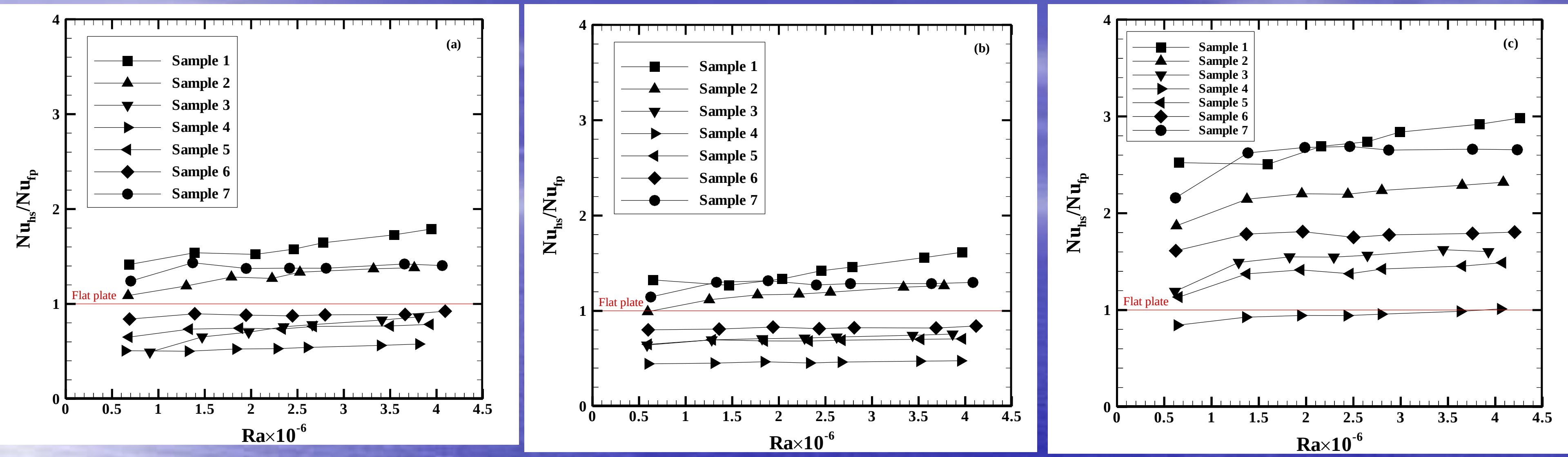
Performance comparison between pin fin and plate fin heat sinks.







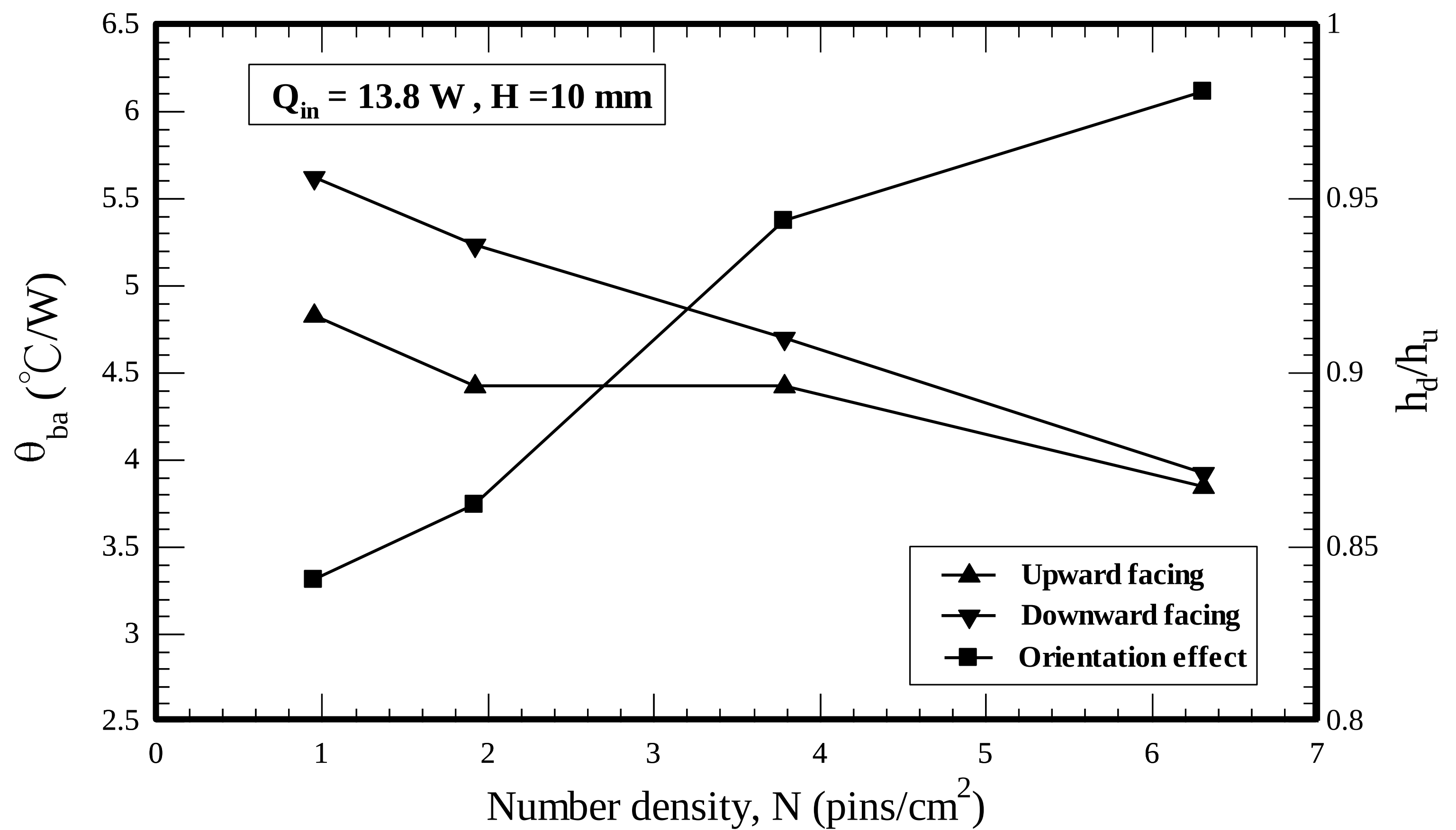
Performance comparison among the three orientations, Pin fin.



Ratio of Nu vs. Ra number for all the test samples (a) upward; (b) sideward; and (c) downward.

	S_p (mm)	S (mm)	N (pins/cm^2)	H (mm)	A_t/A_b
Sample #1	2	8	0.947	10	1.758
Sample #2	2	5	1.916	10	2.533
Sample #3	2	3	3.789	10	4.032
Sample #4	2	2	6.316	10	6.053
Sample #5	2	2	6.316	6	4.032
Sample #6	2	2	6.316	4	3.021
Sample #7	2	2	6.316	2	2.011
Flat Plate	—	—	0	0	1

Geometric details of tested heat sinks.



Number density versus thermal resistance and orientation effect.

Augmented Natural convection

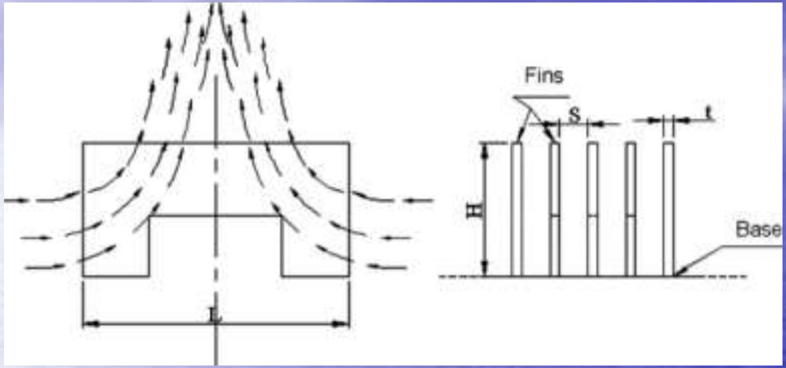
- Passive Augmentation
  - Metal Foam, Carbon Foam, Hollow surface
  - Coating
- Active enhancement

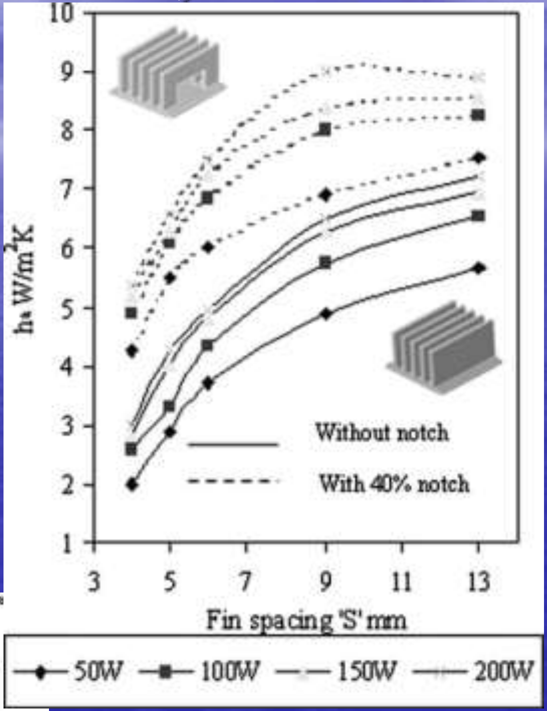
  \- EHD

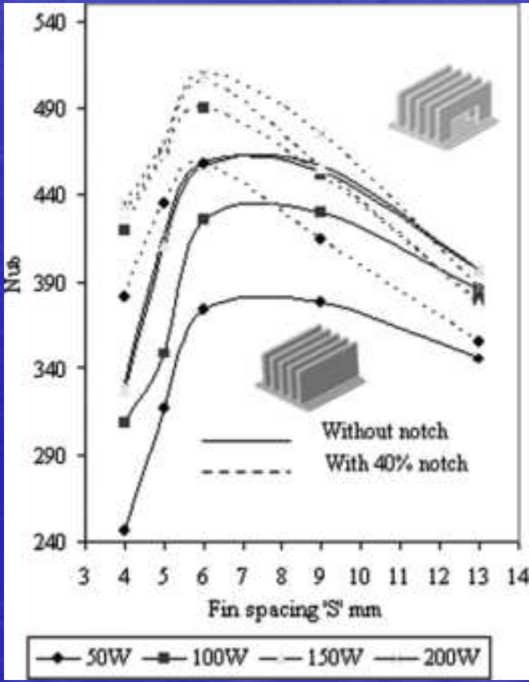
  \- Piezo Fans

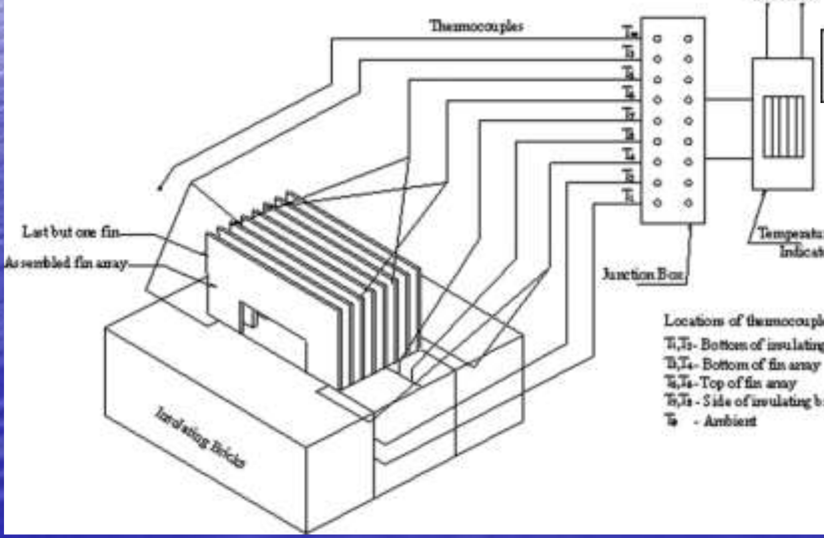
Notch, hollow surface

J. of Heat Transfer 2009, Vol. 131 / **082501-1**



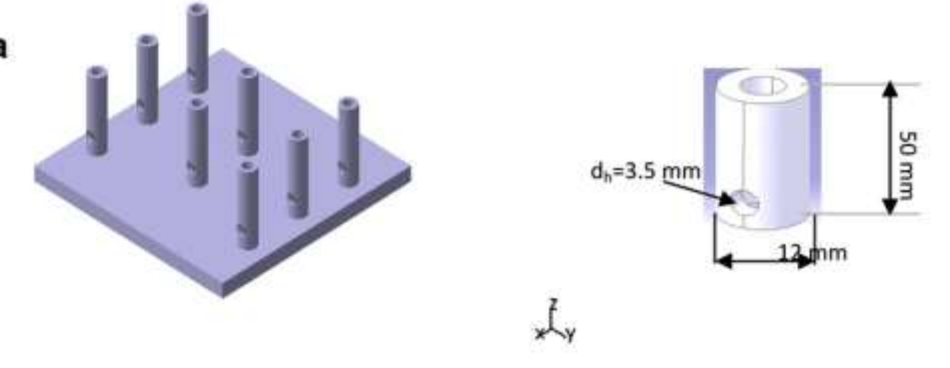




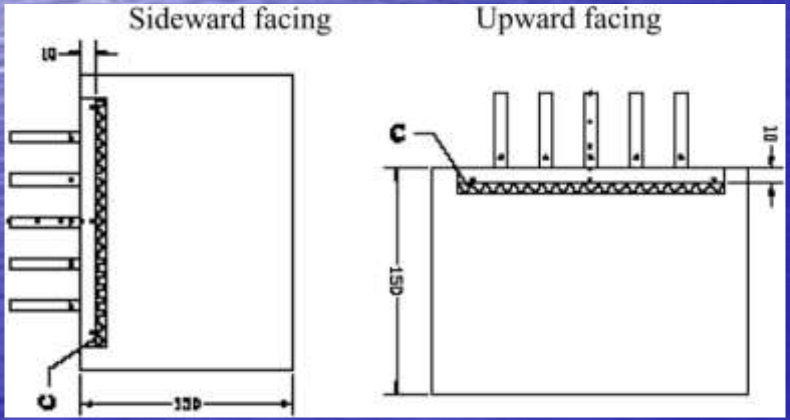


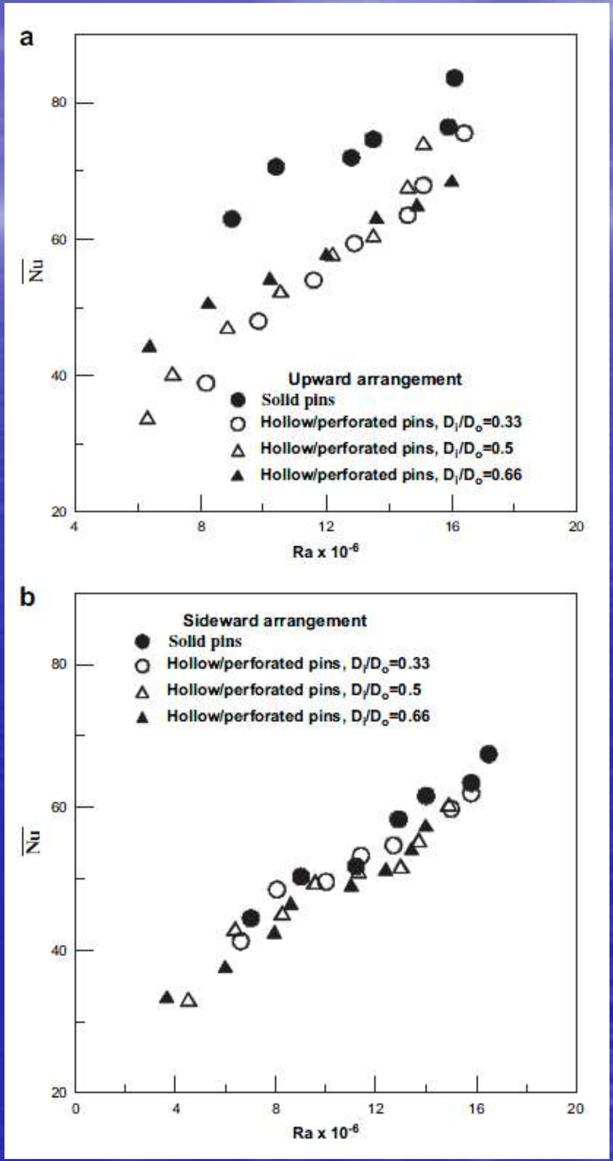
Locations of thermocouples
T_1,T_2- Bottom of insulating bricks
T_3,T_4- Bottom of fin array
T_5,T_6-Top of fin array
T_7,T_8 - Side of insulating bricks
T_9 - Ambient

Energy 35(2010) 2870-2877



Perspective view of Hollow/single perforated pin fin heat sink





Carbon Foam…

J. OF MATERIALS SCIENCE 39 (2004) 3659 – 3676

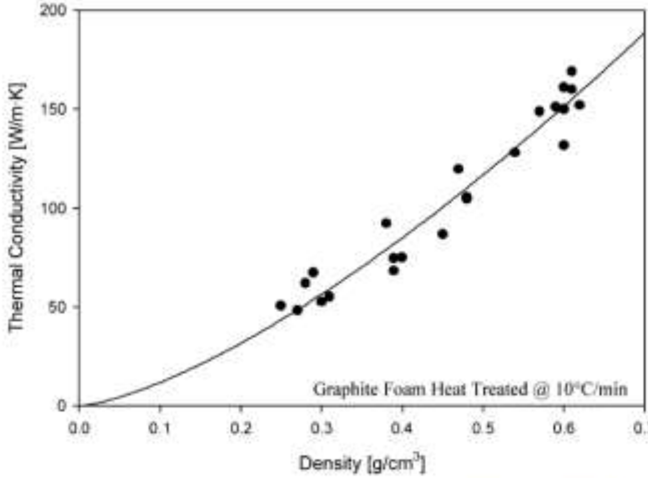


Figure 18 Plot of measured data and results of a new model which incorporates length.

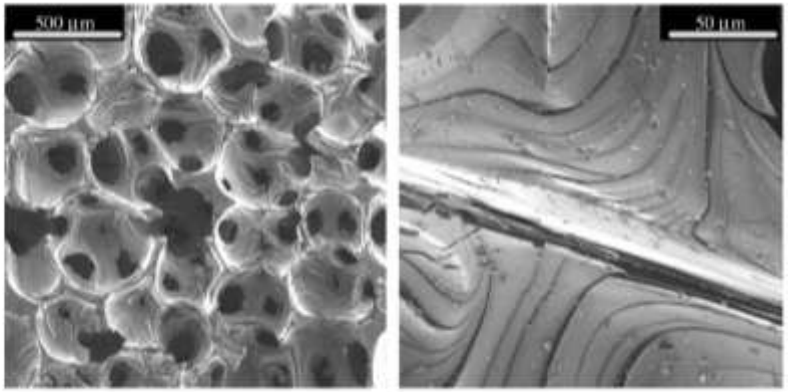


Figure 1 SEM images of mesophase pitch-derived foams.

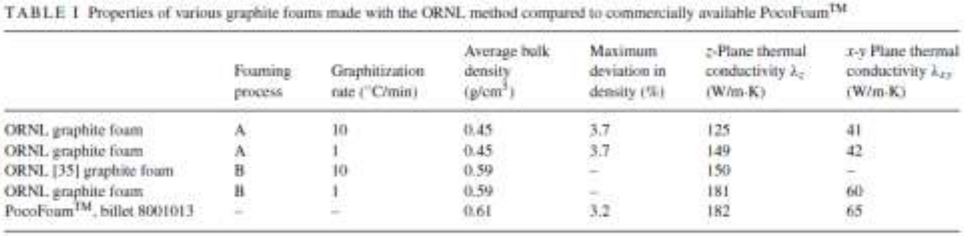


TABLE 1 Properties of various graphite foams made with the ORNL method compared to commercially available PocoFoam™

	Foaming process	Graphitization rate (°C/min)	Average bulk density (g/cm³)	Maximum deviation in density (%)	z-Plane thermal conductivity λ_z (W/m·K)	x-y Plane thermal conductivity λ_{xy} (W/m·K)
ORNL graphite foam	A	10	0.45	3.7	125	41
ORNL graphite foam	A	1	0.45	3.7	149	42
ORNL [35] graphite foam	B	10	0.59	–	150	–
ORNL graphite foam	B	1	0.59	–	181	60
PocoFoam™, billet 8001013	–	–	0.61	3.2	182	65

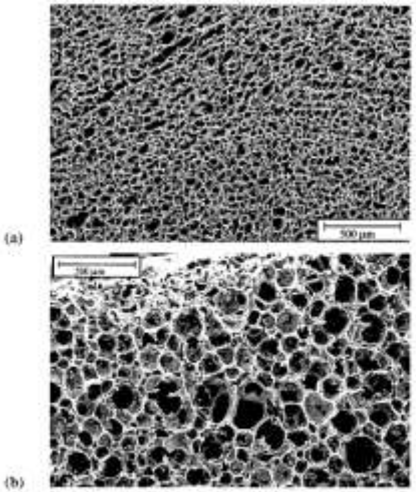


Figure 4. SEM Images of foam samples, (a) small, uniform

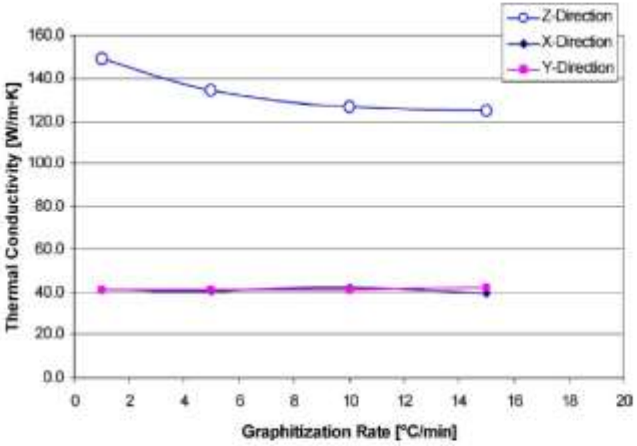


Figure 2 Effect of final heat treatment rate (graphitization) on the thermal properties of graphitic foams made with process A.

Metal Foam

Experimental Thermal and Fluid Science 32 (2008) 1740–1747

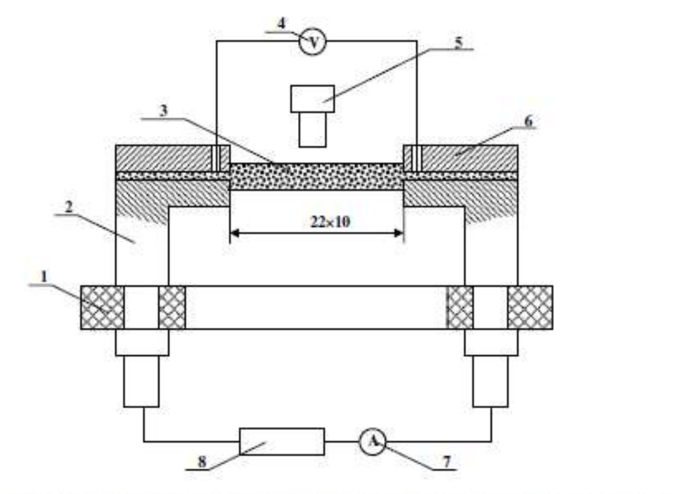
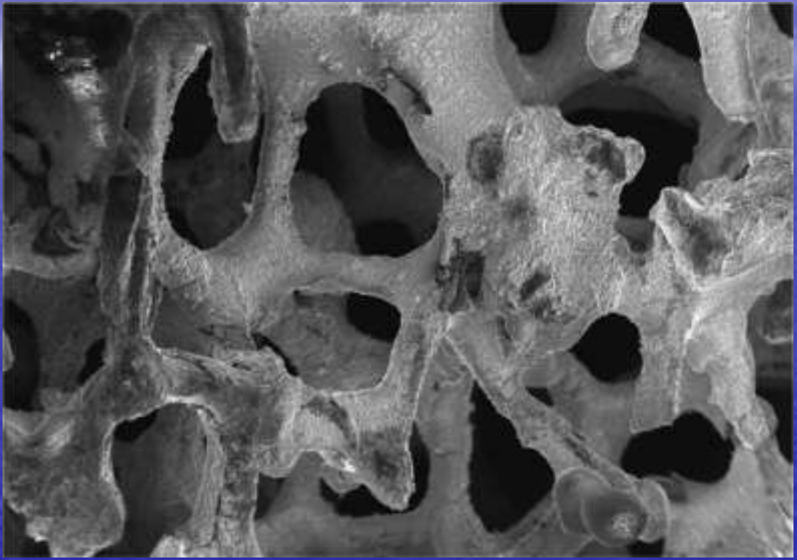


Fig. 1. Experimental apparatus: (1) housing, (2) aluminum contacts, (3) metal foam specimen, (4) voltmeter, (5) IR camera, (6) holder, (7) amperemeter, and (8) power supply.



Experimental data on heat transfer enhancement in the strip of metal foam at natural convection in both vertical and horizontal plates demonstrates that heat transfer is increased dramatically (up to 18–20 times for metal foam of 20 ppi) relative to the flat plate of the same overall dimensions.

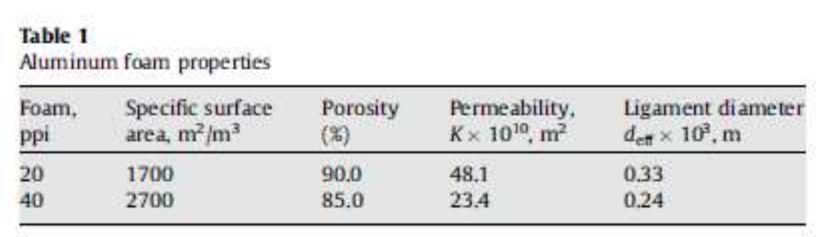


Table 1
Aluminum foam properties

Foam, ppi	Specific surface area, m^2/m^3	Porosity (%)	Permeability, $K \times 10^{10}$, m^2	Ligament diameter $d_{eff} \times 10^3$, m
20	1700	90.0	48.1	0.33
40	2700	85.0	23.4	0.24

EHD Convection

EHD stands for Electro Hydro Dynamics which is the study of the flow of a fluid under the effect of an electric field.

When a Newtonian, impressible fluid is subject to the presence of electric field, the Navier-Stokes equation becomes:

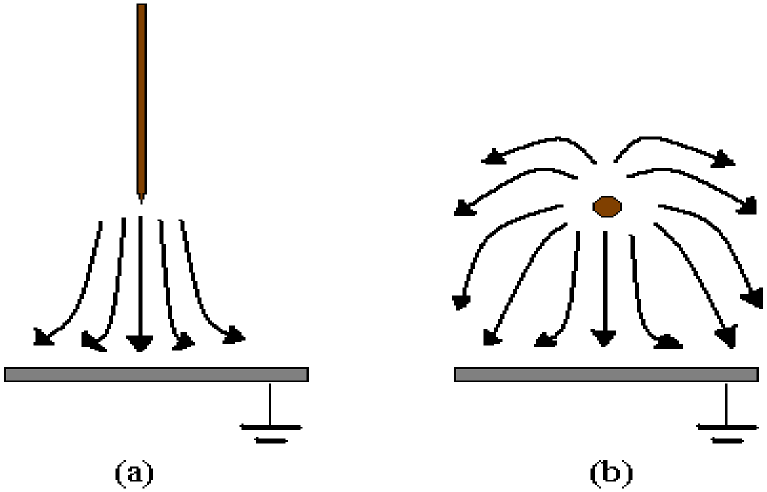
$$\rho \frac{D\vec{V}}{Dt} = -\nabla p + \mu \nabla^{2}\vec{V} + \rho \vec{g} + \boxed{\vec{f}_{e}}$$

EHD body force

Corona wind

A fluid motion driven by an electric field is termed a coron wind or an ionic wind.

Electrode geometry



Needle electrode

Wire electrode

Typical electrode types.

Corona region (Ionization region)

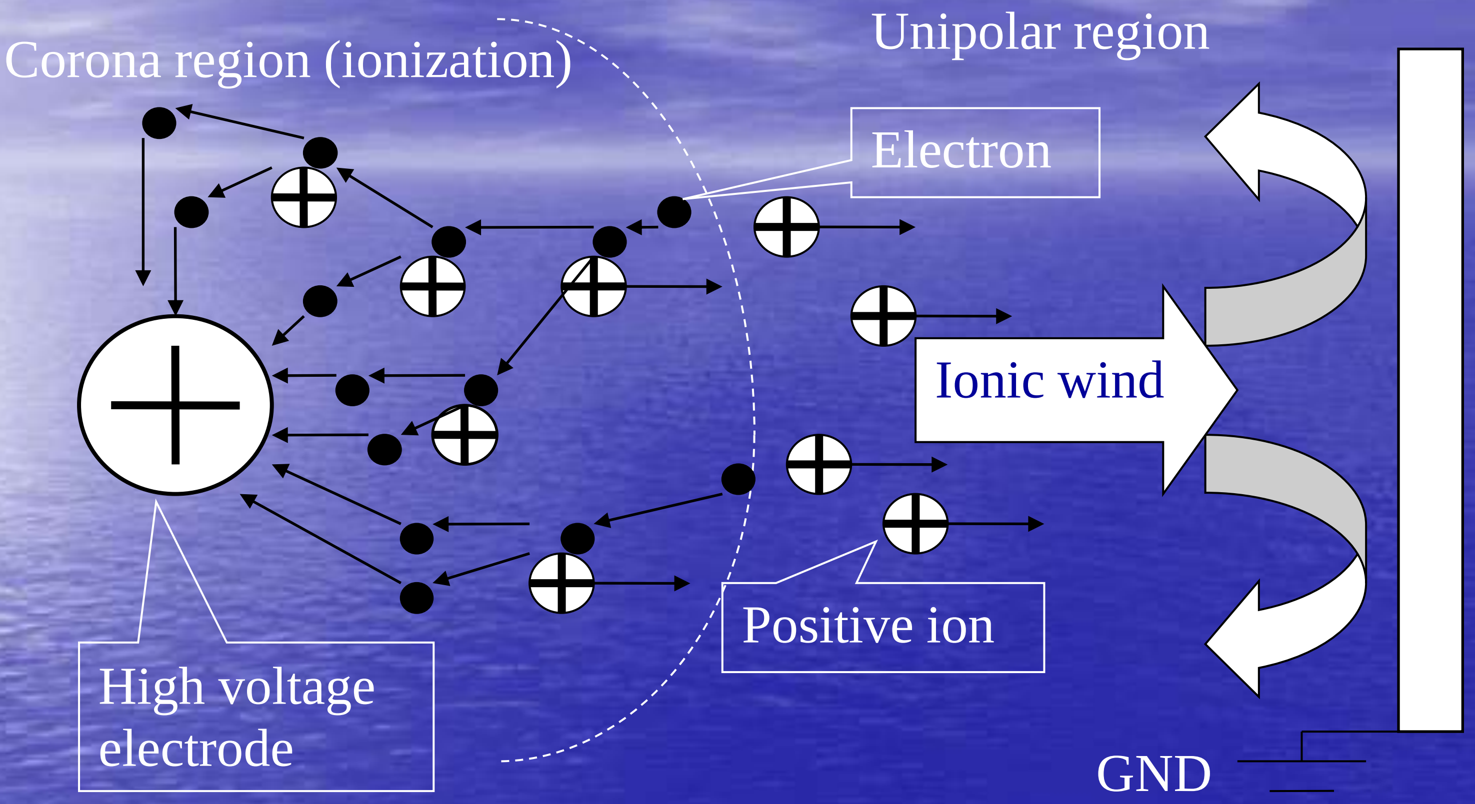
Near the charged electrode, ionization occurs and creates positive ions and free electrons in a process known as the electron avalanche. The positive ions are attracted toward or repelled away from the curved electrode (depending on the polarity). The electrons thus migrate in the opposite direction.

Unipolar region

The energized electrons, accelerated by the electric field, inelastically collide with the neutral atoms, entraining the stagnant fluid from the ambience to the grounded surface.

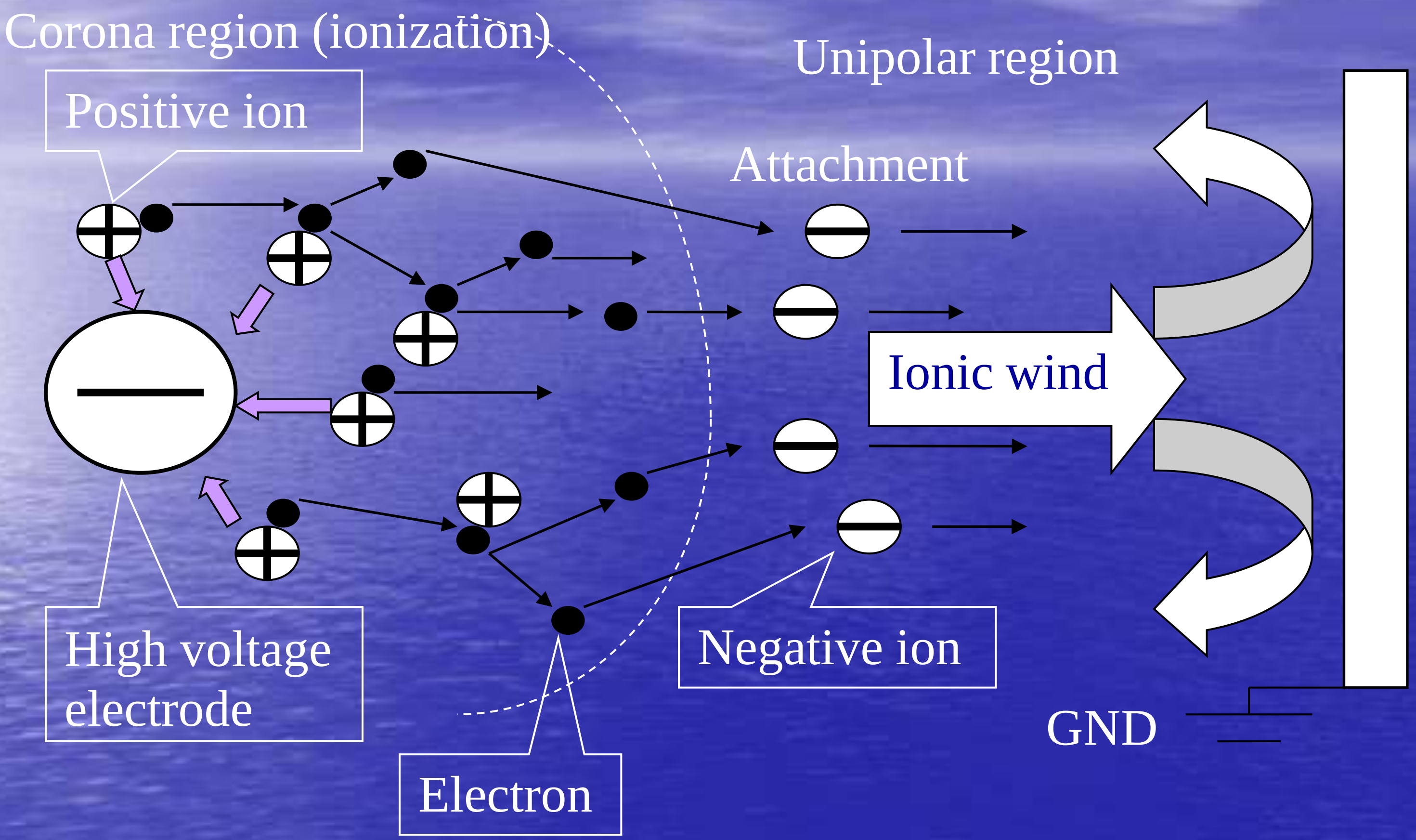
$$E_p = \frac{2V}{R\log_{10}\left(\frac{4x}{R}\right)}$$

Positive Corona

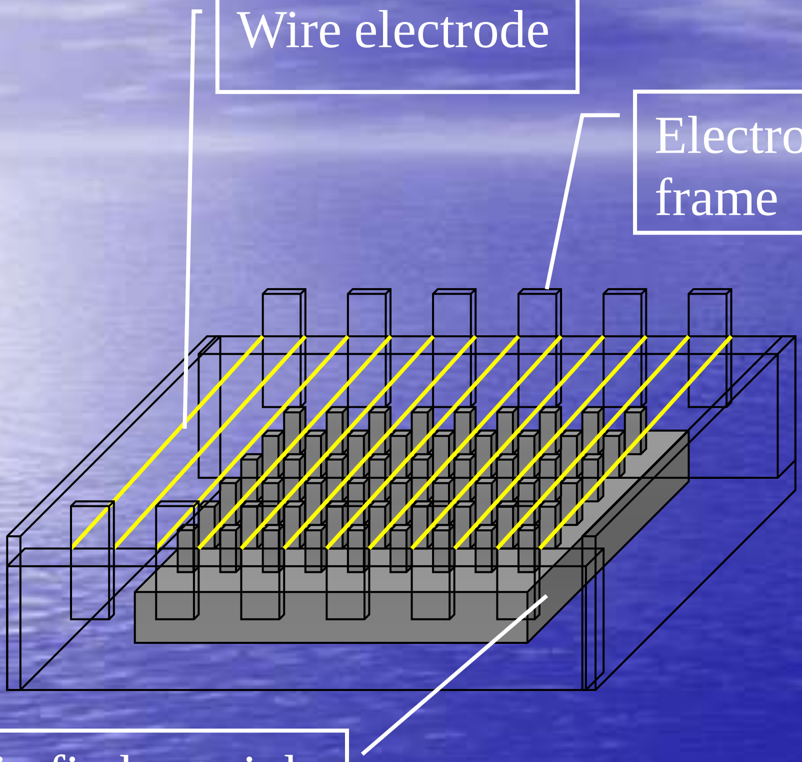


Positive corona generation and ionic wind.

Negative Corona



Negative corona generation and ionic wind.

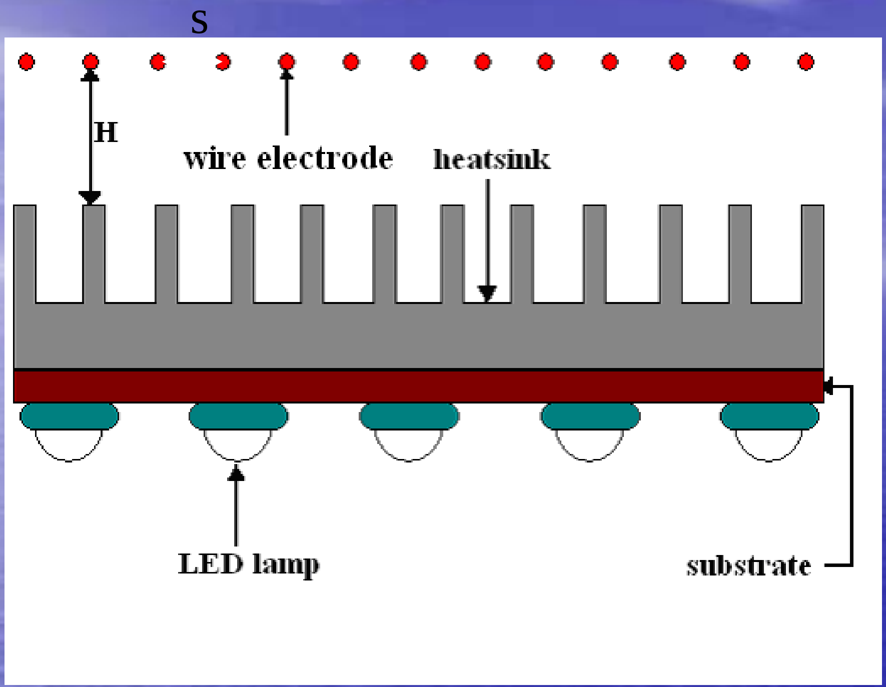


Wire electrode

Electrode frame

Pin fin heat sink

The electrode arrangement.

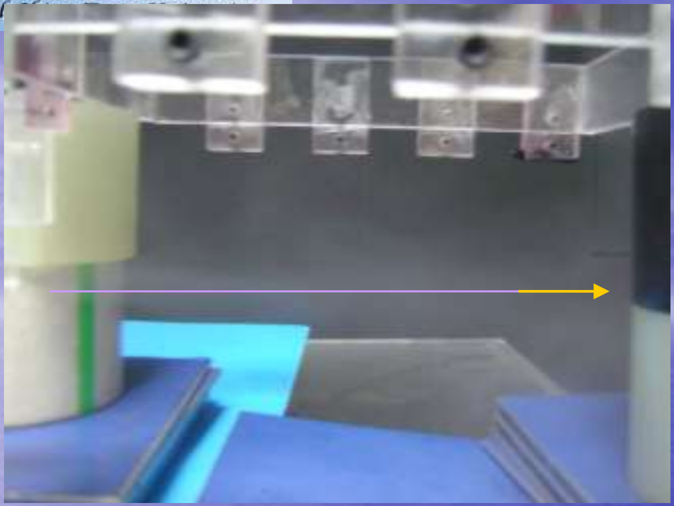


H: separation distance
S: electrode spacing

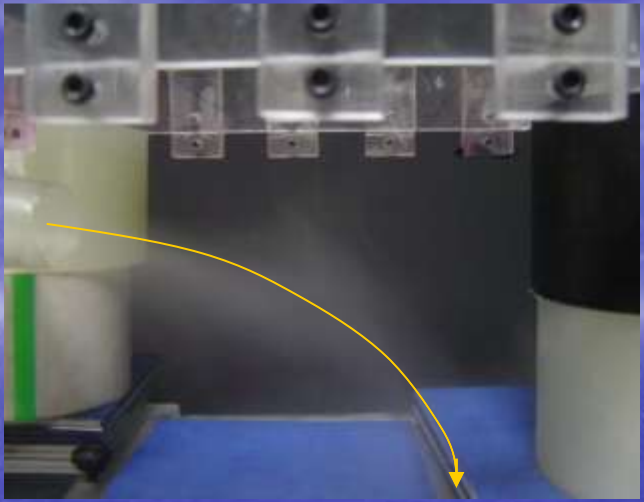
Illustration of the electrode arrangement.

Enhancement ratio $E_R = \dfrac{Nu_{EHD}}{Nu_{natural}}$

$$h' = \frac{h_{EHD}}{h_{natural}} = \frac{(T_b - T_a)_{natural}}{(T_b - T_a)_{EHD}}$$



(a)



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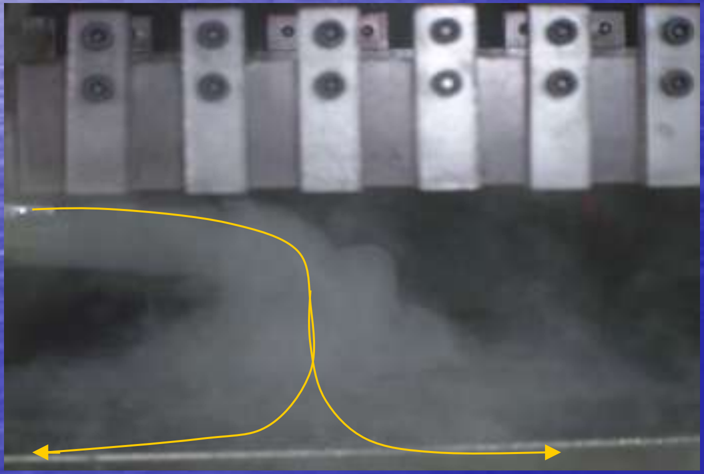
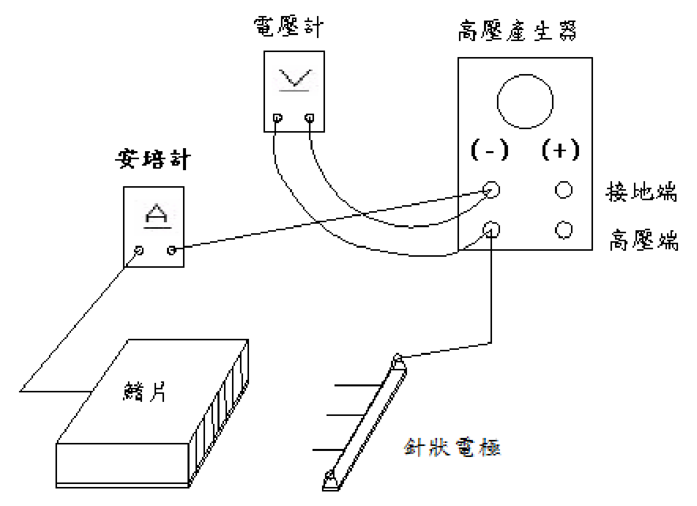
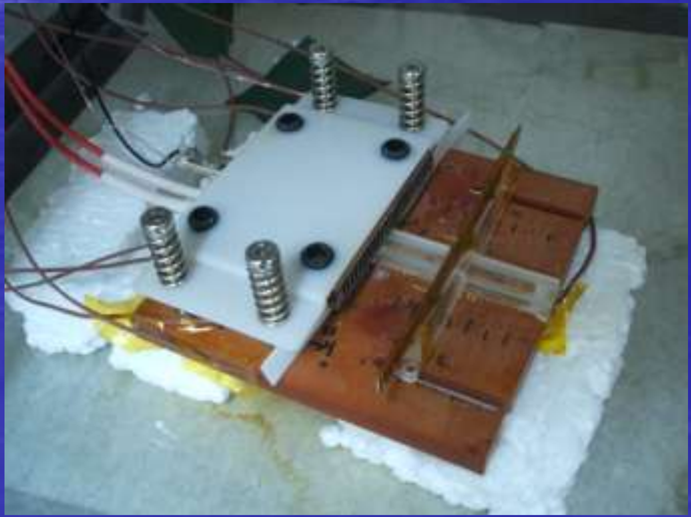


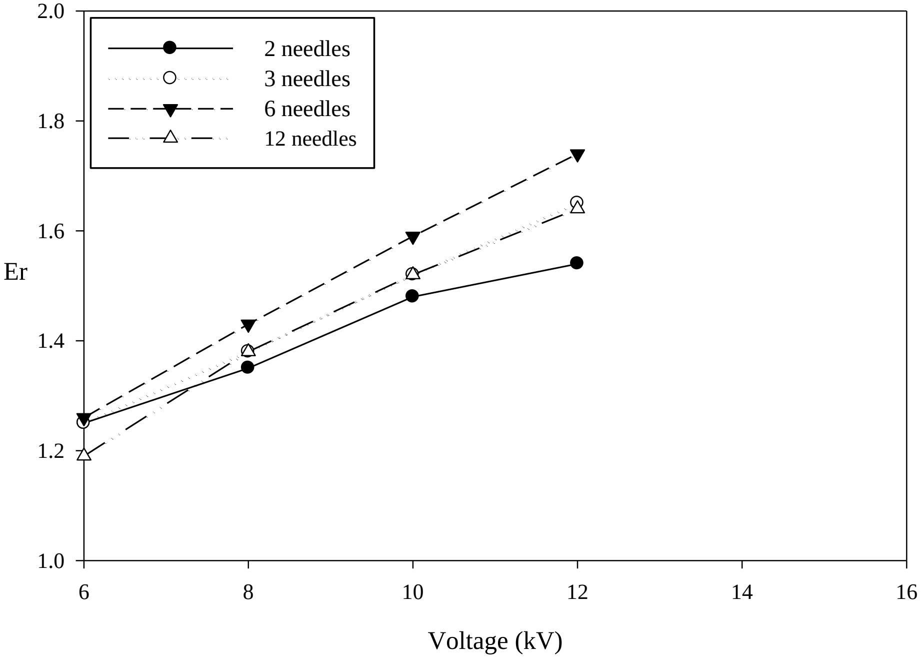
(c)

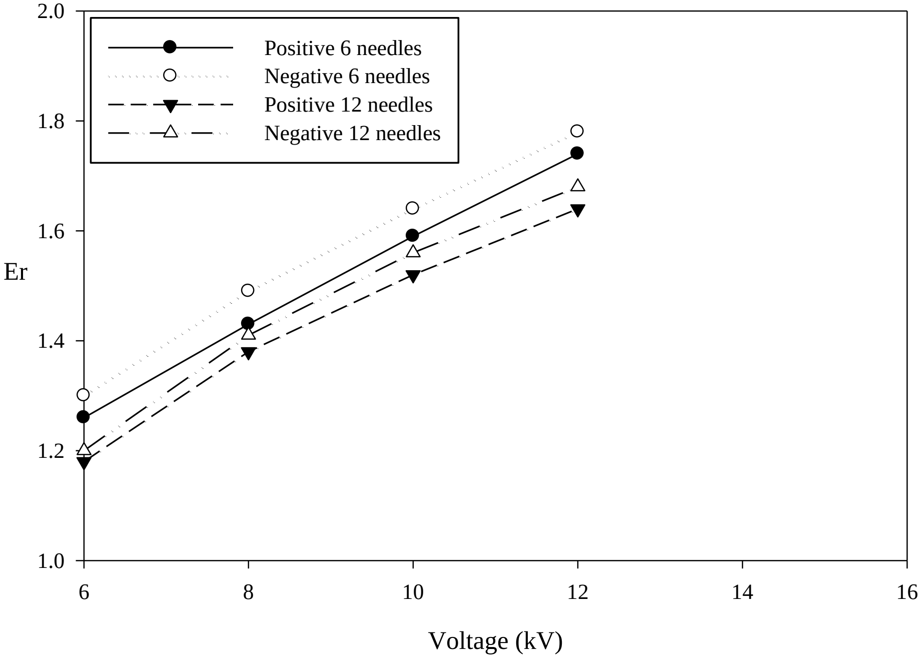
Flow visualization, (a) 0 kV,
(b) +5 kV, (c) +10 kV.


電壓計
V
高壓產生器
(-)
(+)
接地端
高壓端
安培計
A
鰭片
針狀電極

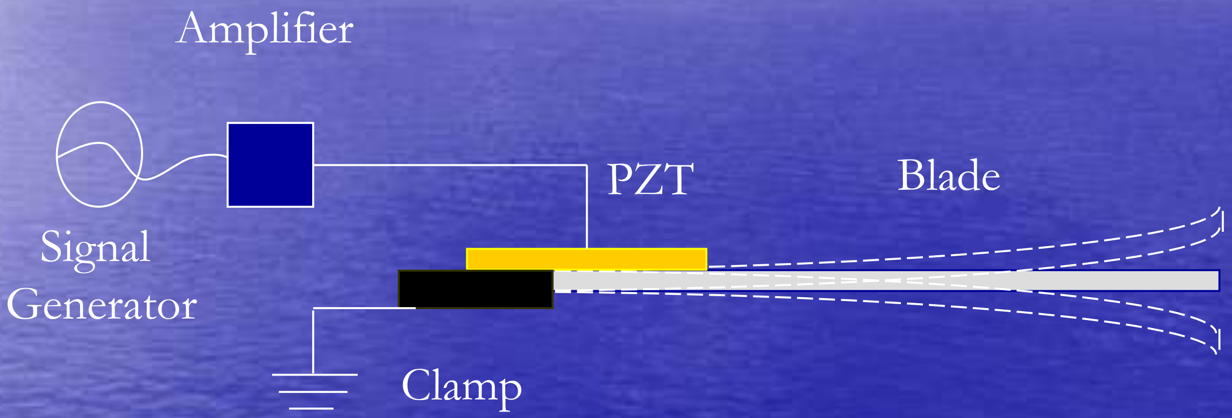


不同電極數目與電極極性及對熱傳性能的影響
距離10mm

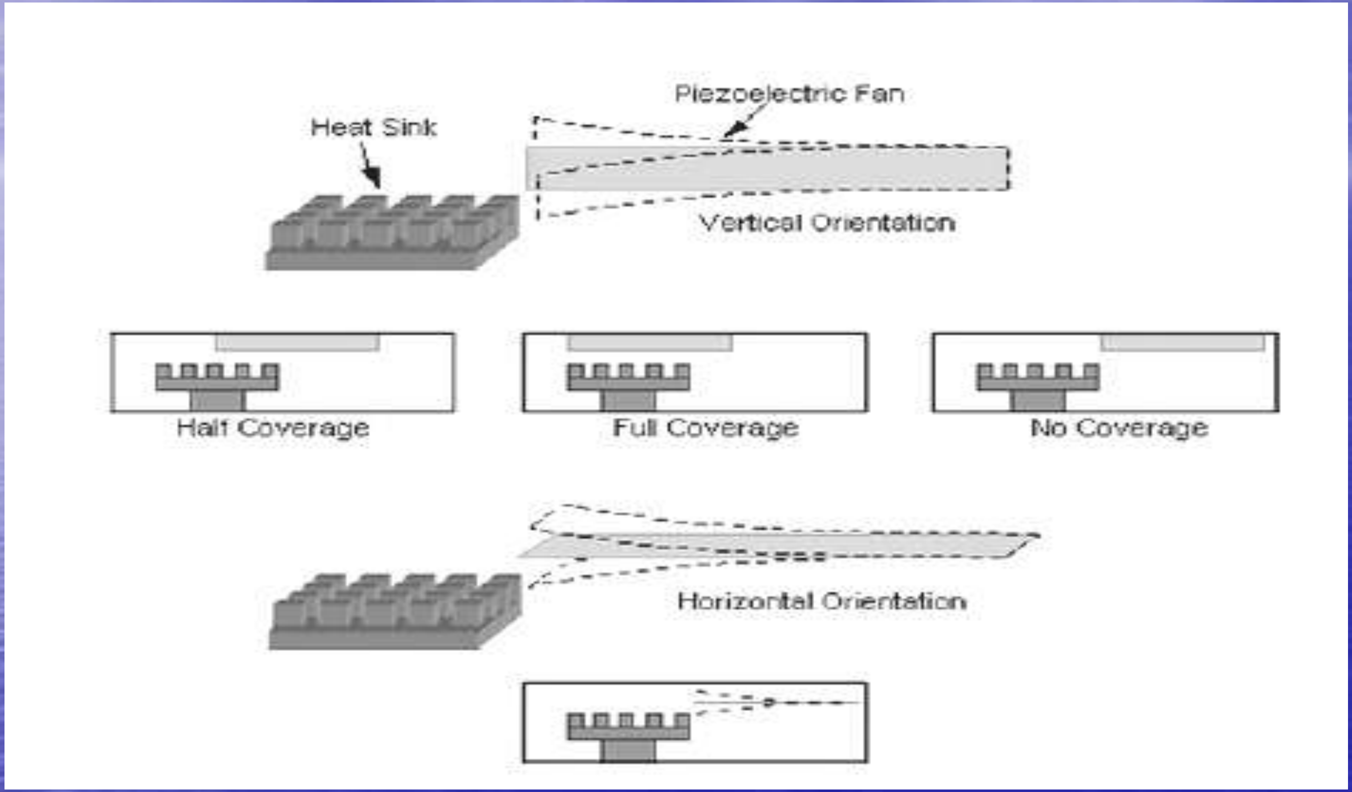




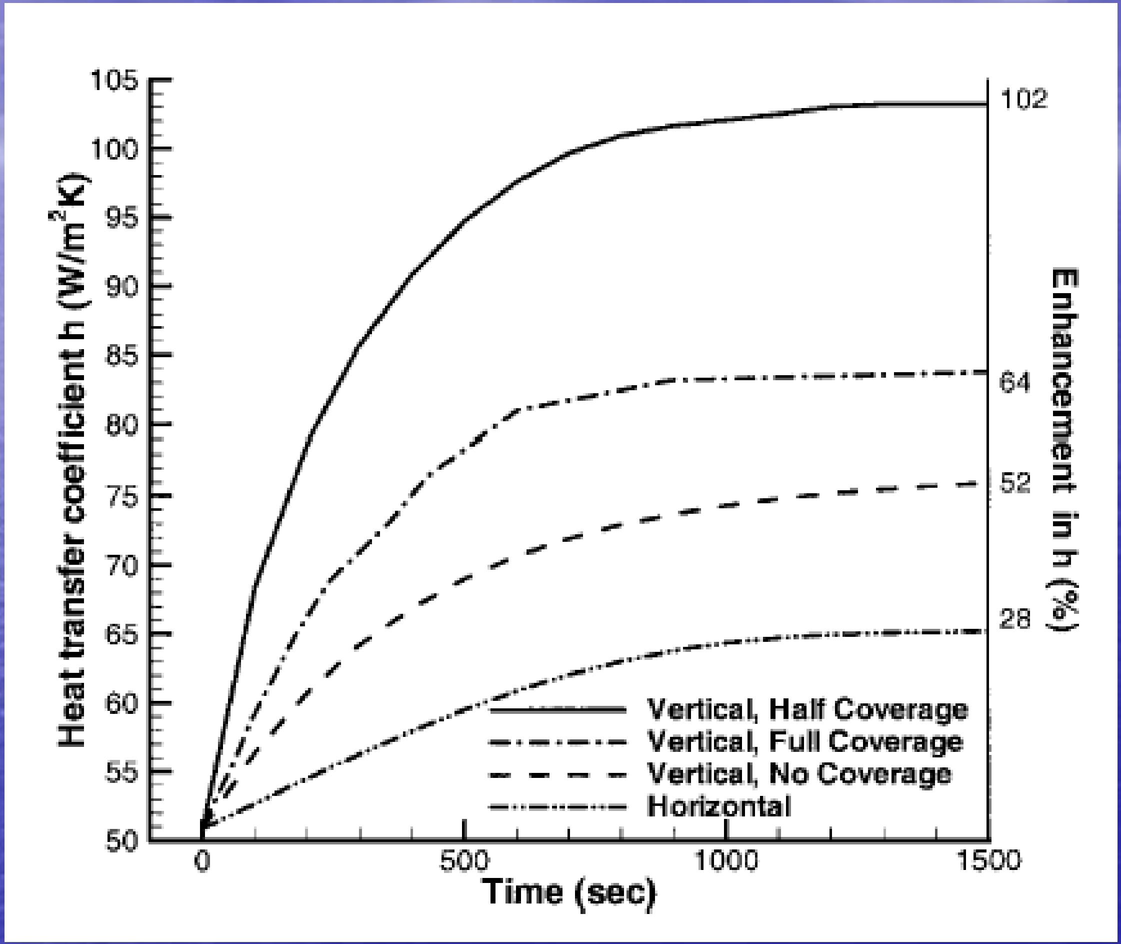
Piezoelectric Fans



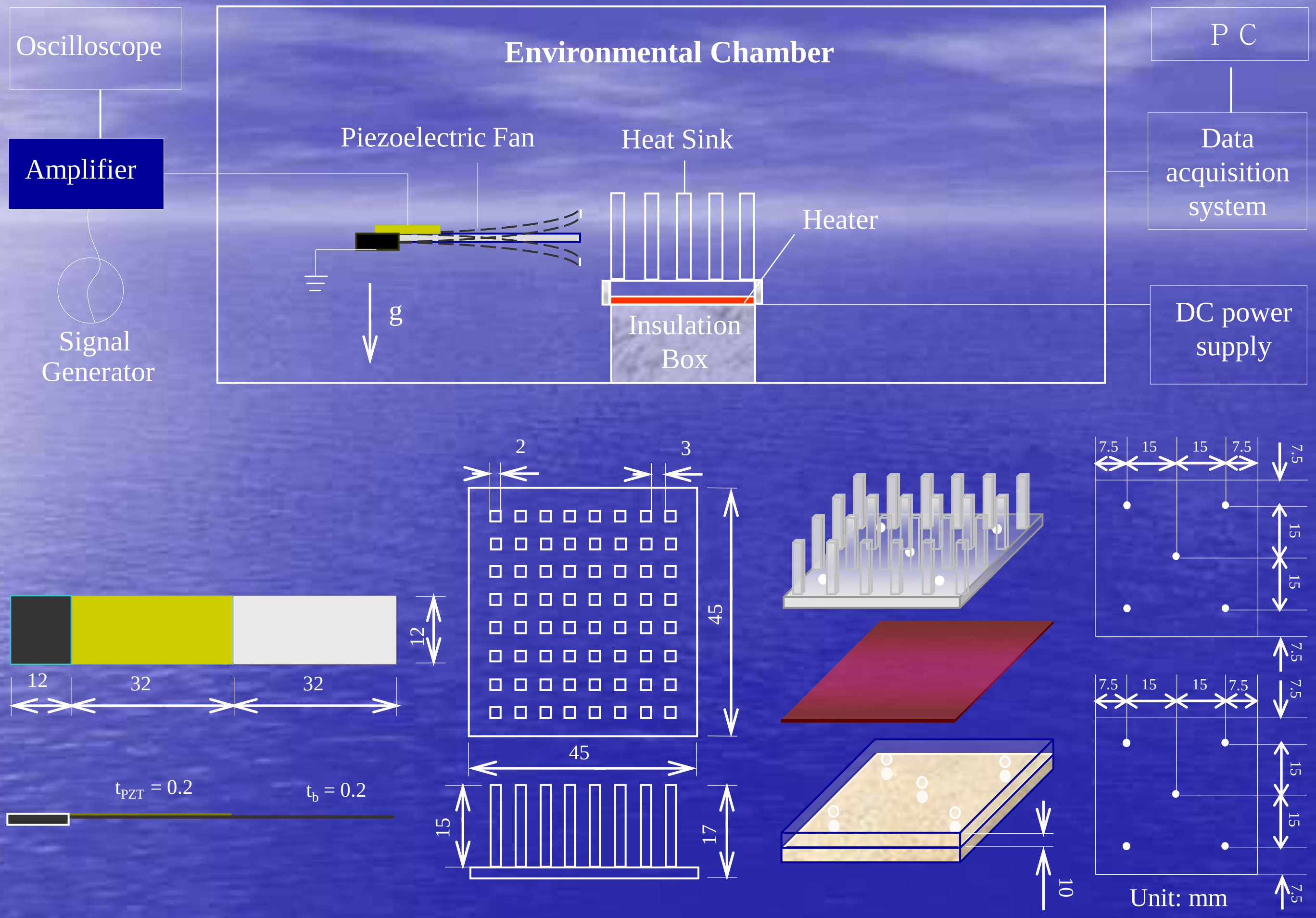
Schematic of a piezoelectric fan.



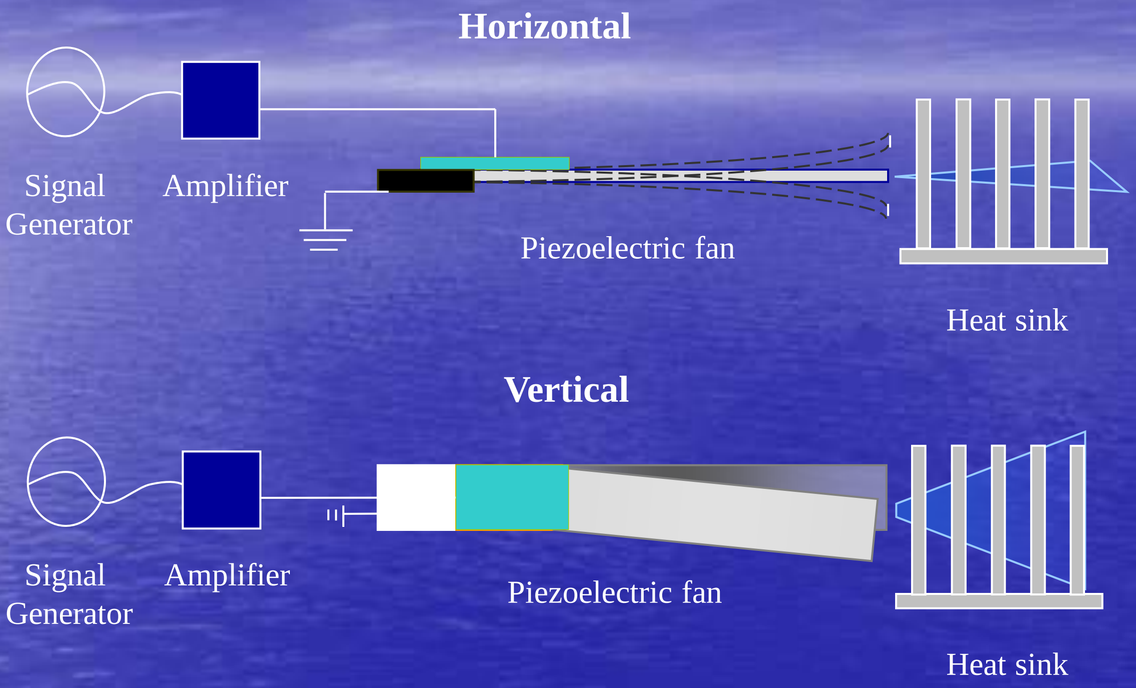
Different arrangements of the piezoelectric fans.



Enhancement in heat transfer coefficient for the four different fan positions in the enclosure. The area is based on the heat source [4].



Schematic of the experimental setup



Horizontal

Signal Generator

Amplifier

Piezoelectric fan

Heat sink

Vertical

Signal Generator

Amplifier

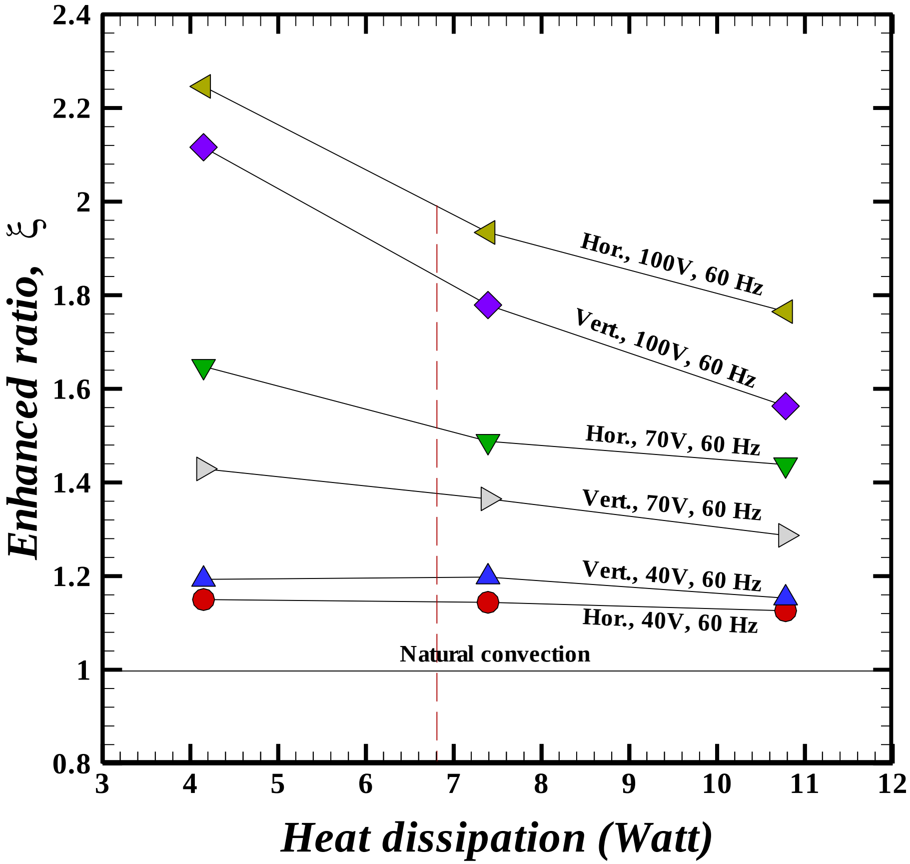
Piezoelectric fan

Heat sink

Arrangement of the piezoelectric fans

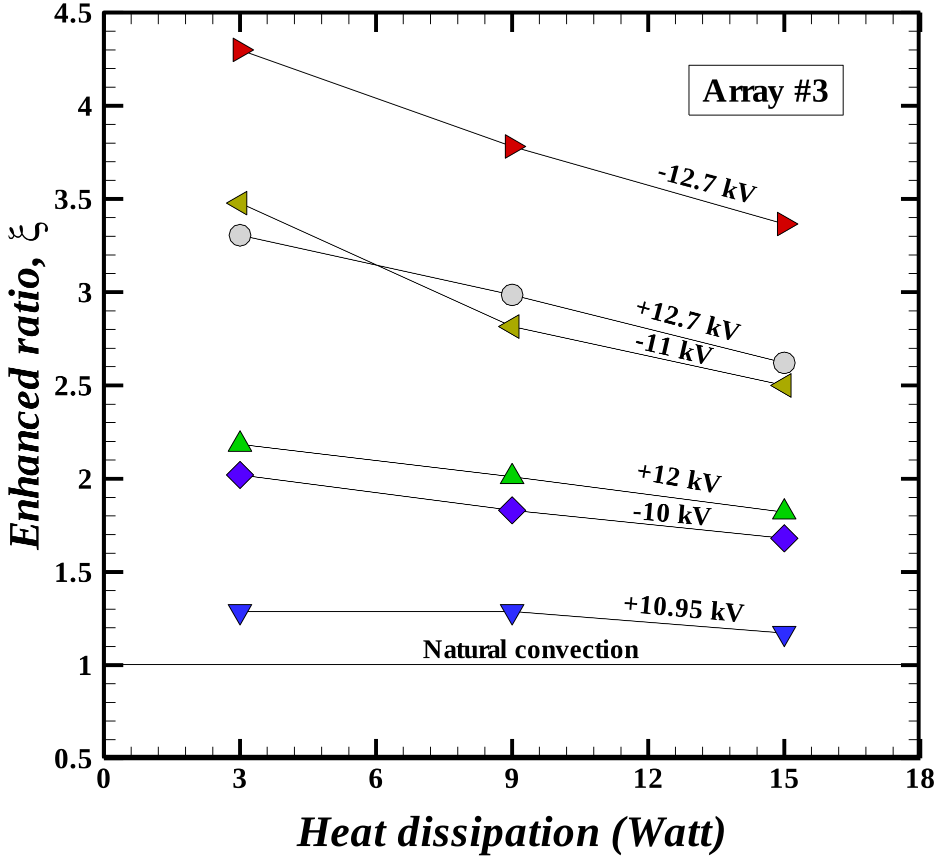
Piezoelectric fan + pin fin

A_t = 0.009705 m^2, ψ= 4.76



EHD needle array + plate fin

A_t = 0.022888 m^2, ψ= 2.20



Dependence of the enhanced ratio on the power dissipation

4.29 W/m^{2}K 5.25 W/m^{2}K 6.11 W/m^{2}K 5.43 W/m^{2}K 6.57 W/m^{2}K 7.39 W/m^{2}K

Natural Convection

- Generally, for both pin fin and plate fin heat sinks, the upward facing orientation yields the highest heat transfer coefficients, followed by the sideward facing and the downward facing orientation.

- With the same fin height of 10 mm and fin spacing of 2mm, the heat transfer coefficient of pin fins are greater than those of plate fins by 0% ~ 23% due to the more open ends for inducing air flow.

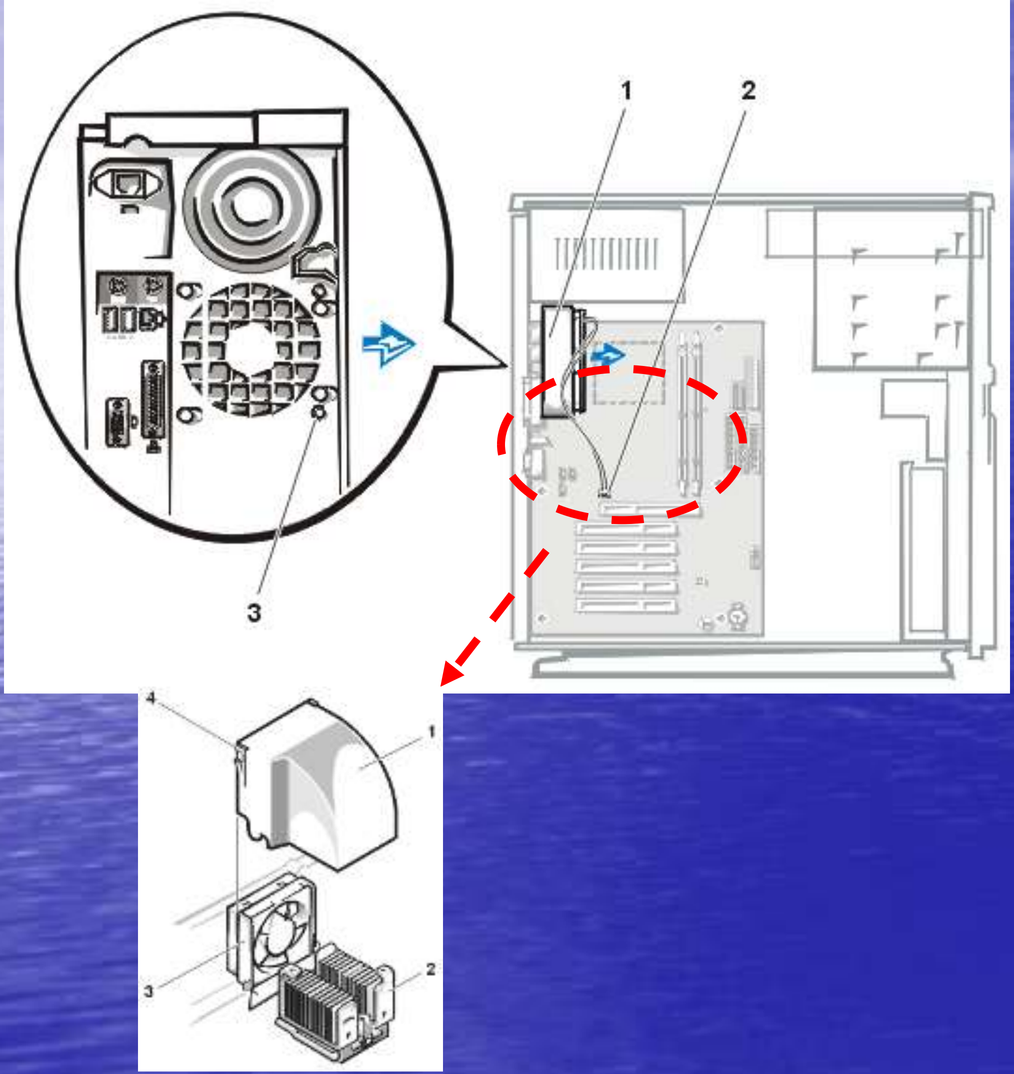
- The orientation effect on the pin fin heat sinks becomes less pronounced as the pin height or as the number density is gradually increased. This interesting result is attributed to the choking phenomenon occurring inside the heat sinks.

EHD Convection

- The electric field intensity is increased with the supplied voltage, and so does the heat transfer coefficient.
- Design criteria shall be taken to avoid flow field interference of the corona wind generated from the individual electrode.
- The negative polarity slightly outperforms the positive one by 6% due to its higher current density and mobility between the electrodes and the grounded surface.
- EHD has been proven as a feasible cooling technique in the present study by showing a threefold heat transfer enhancement at the expense of small power consumption.

Heat Sinks Under Forced Convection

Fundamentals – Semi-Active Heat sink

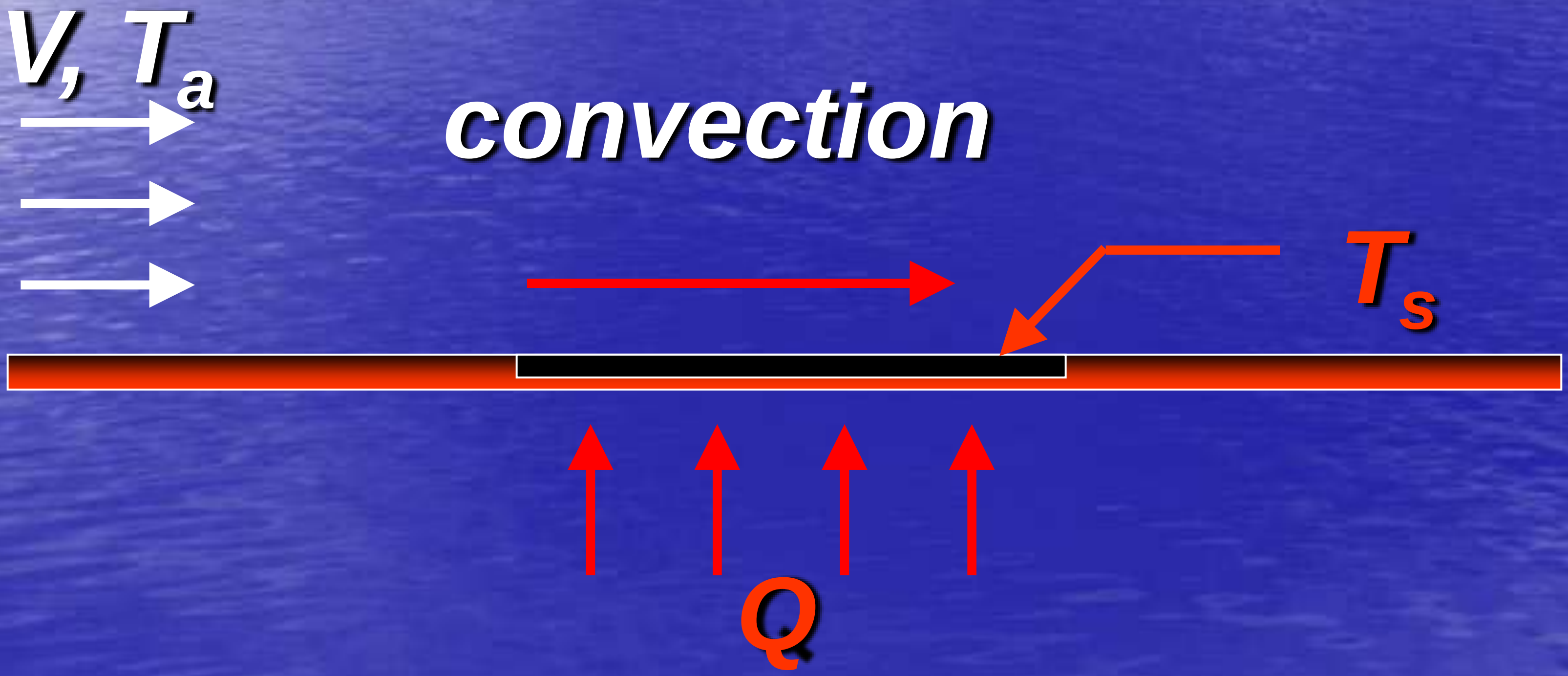


Fundamentals – Active Heat sink

Fundamentals of Heat Transfer

- For a given Q、As、V and Ta：

$$Q = hA_s\left(T_s - T_a\right) \quad \Longrightarrow \quad T_s = T_a + \frac{Q}{hA_s}$$



Fundamentals of Convective Heat Transfer

$$Q = hA_s\left(T_s - T_a\right)$$

- Constraints:

  Usually, T_s is given and must below certain threshold limit. Q is also given, T_a could be specified an upper bound as a constraint.

- For design to meet the constraints. One needs to..

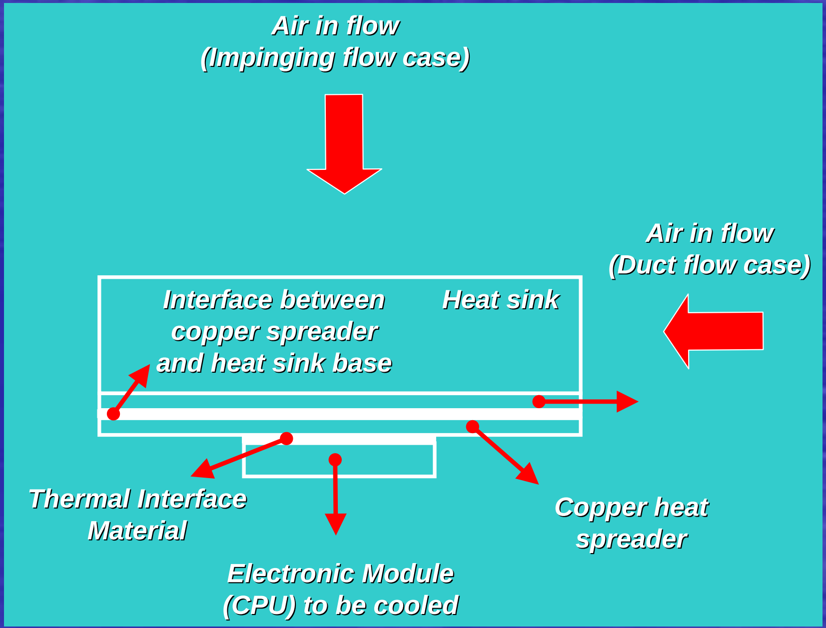
  (a) Increase A?

  (b) Increase h?

  (c) What more can one do?

Objective

- Seeking ways to enhance air-cooling without considerable rise of pressure drop
  - – Focus on cross flow applications
  - – Focus at low velocity operation
  - – Seeking specific fin patterns to tackle the problem



Air in flow
(Impinging flow case)

Air in flow
(Duct flow case)

Interface between
copper spreader
and heat sink base

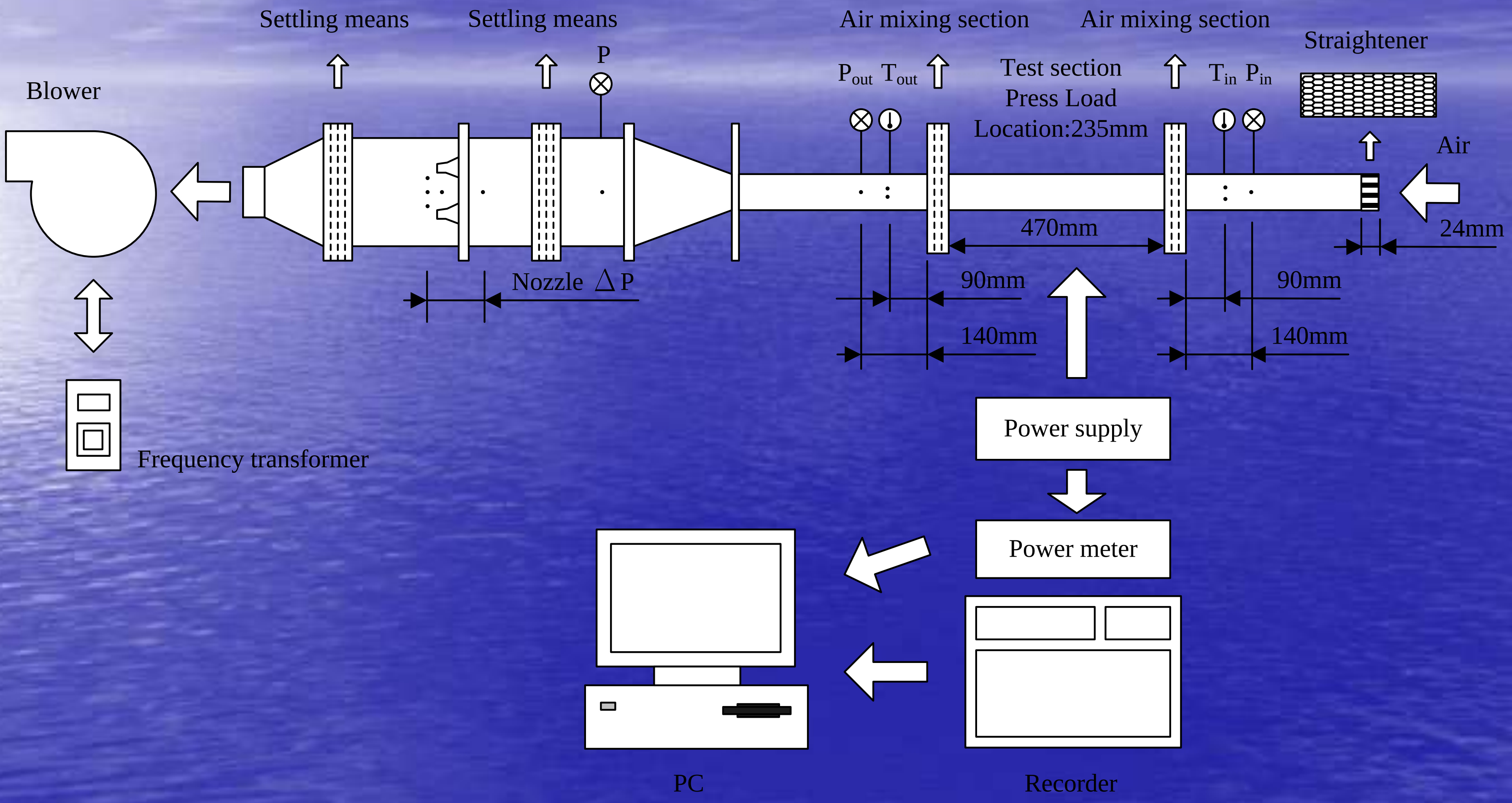
Heat sink

Thermal Interface
Material

Copper heat
spreader

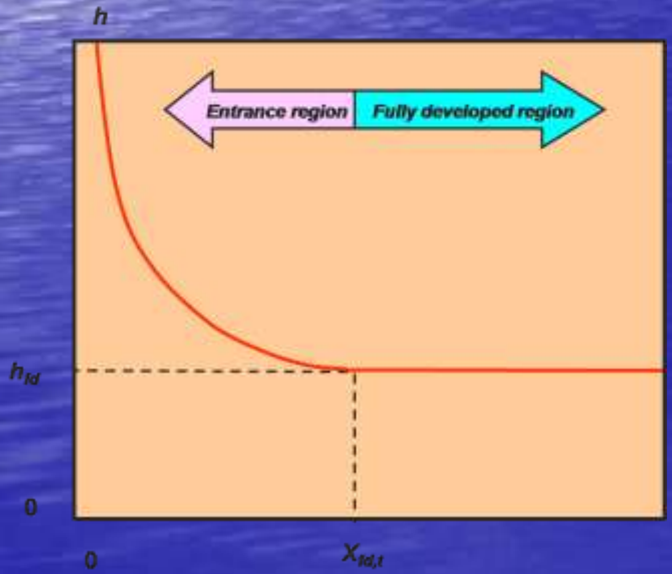
Electronic Module
(CPU) to be cooled

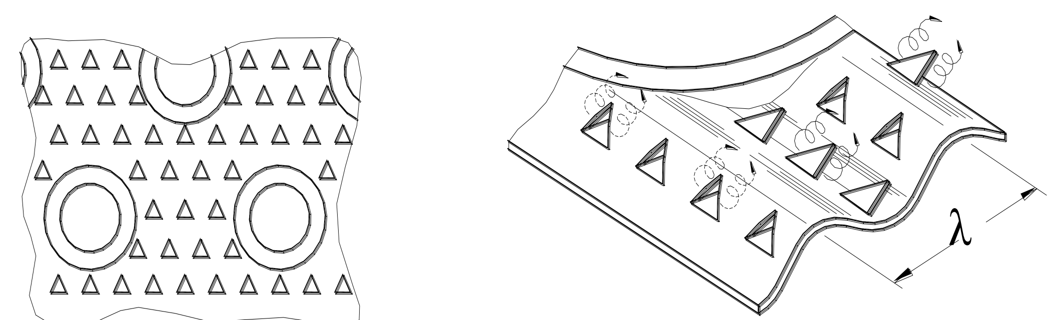
Experimental setup



Some common ways for augmentation

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

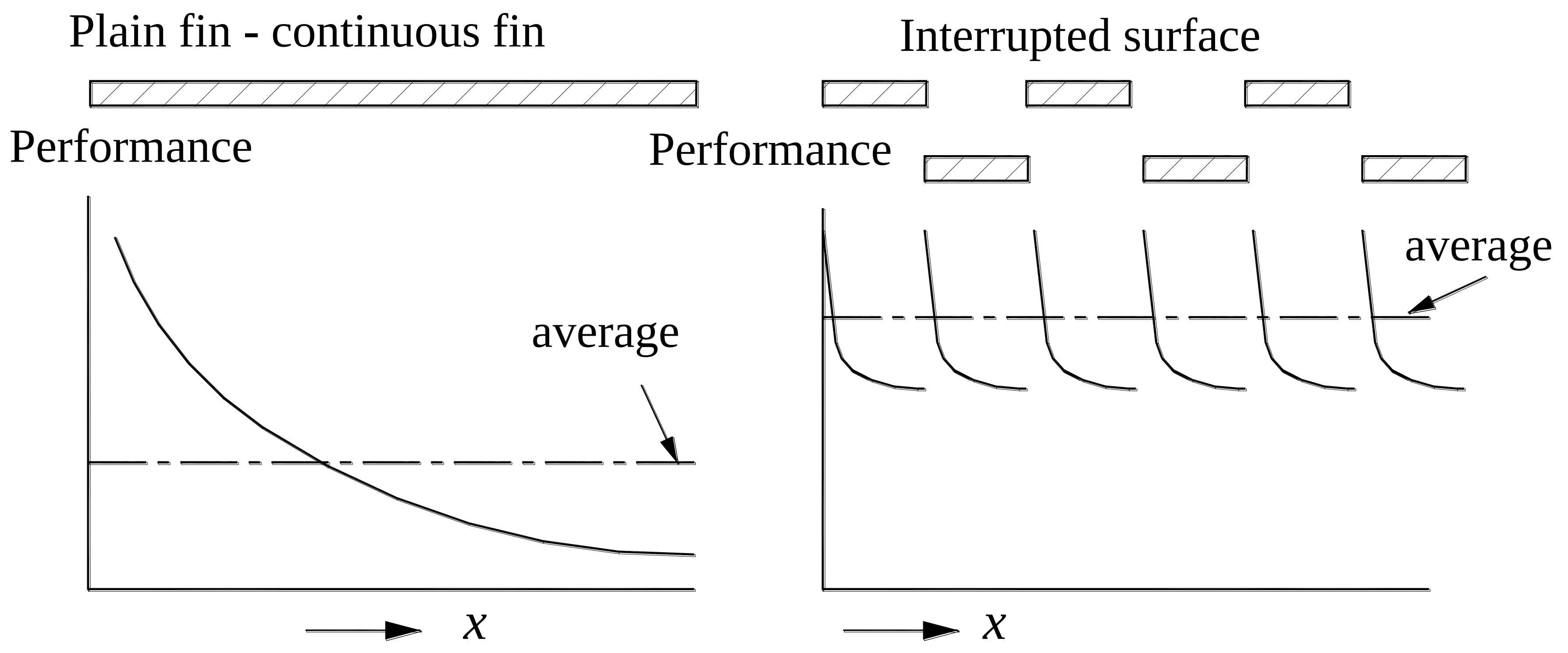




US patent 4817709

Concept of Interrupted surfaces

Boundary restart & Mixing



Various kinds of improvements
- Implementations

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

  Fin spacing can be lower than 1 mm

  (0.8 mm in this study)

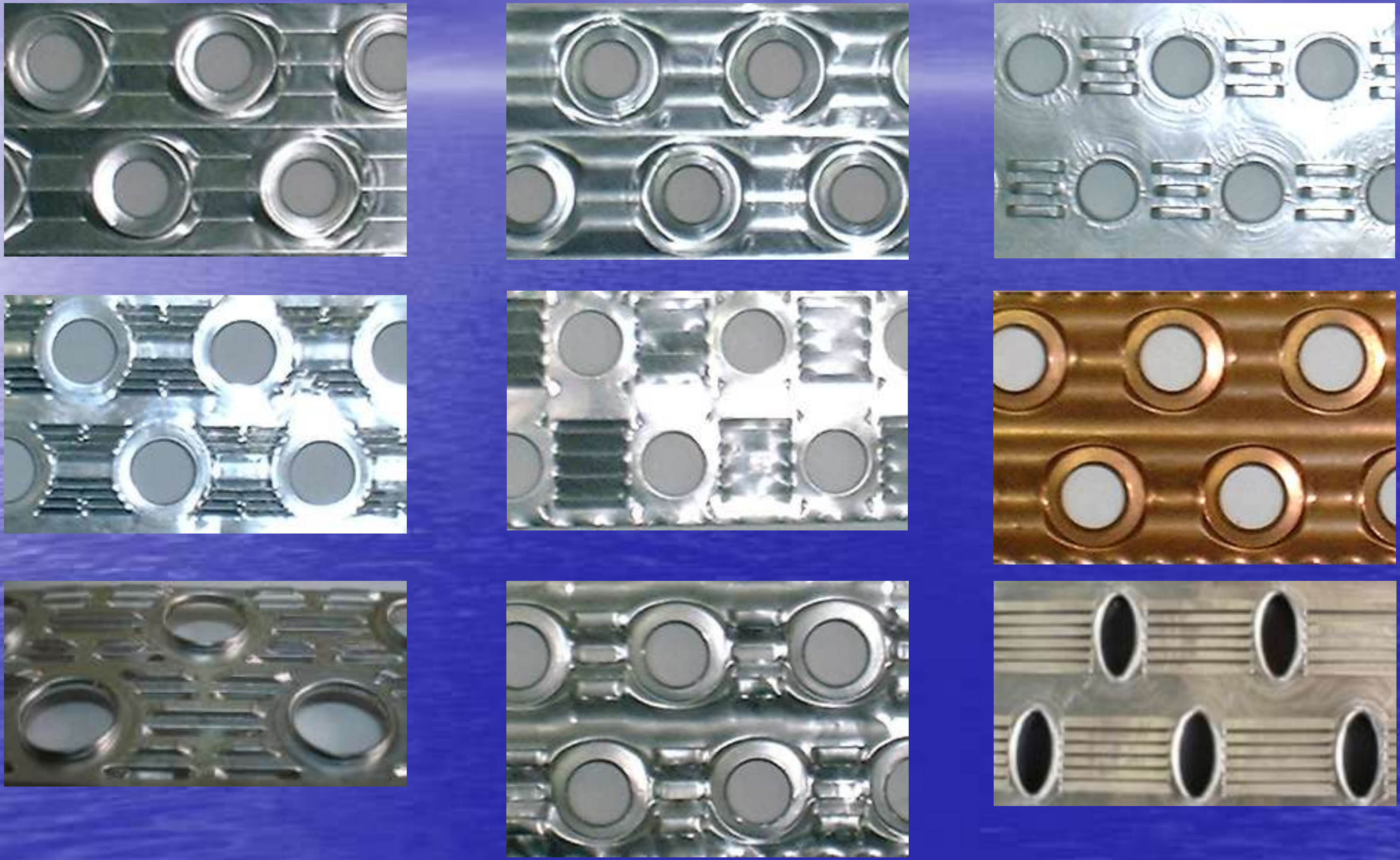
  fin thickness 0.2 mm

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

louver fin

slit fin

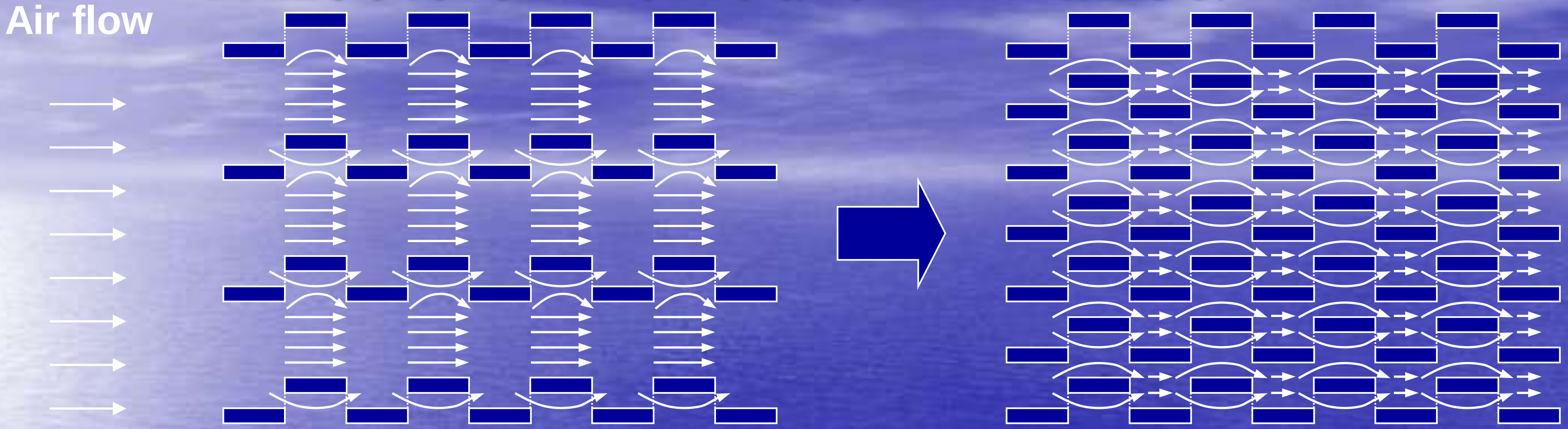
Various Fin Patterns



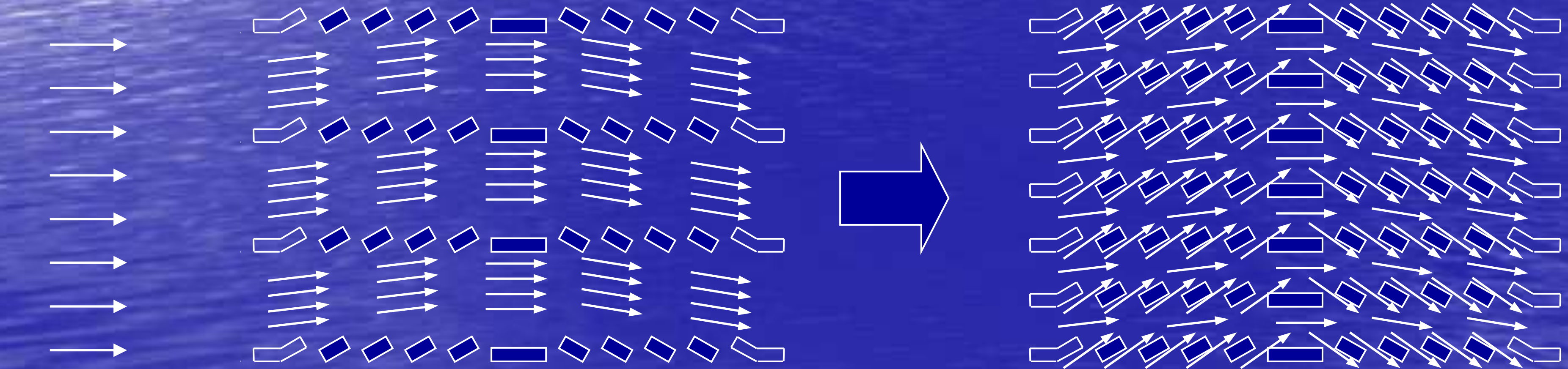
Interrupted surfaces..

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

Effects of Periodic Entrance/Exit



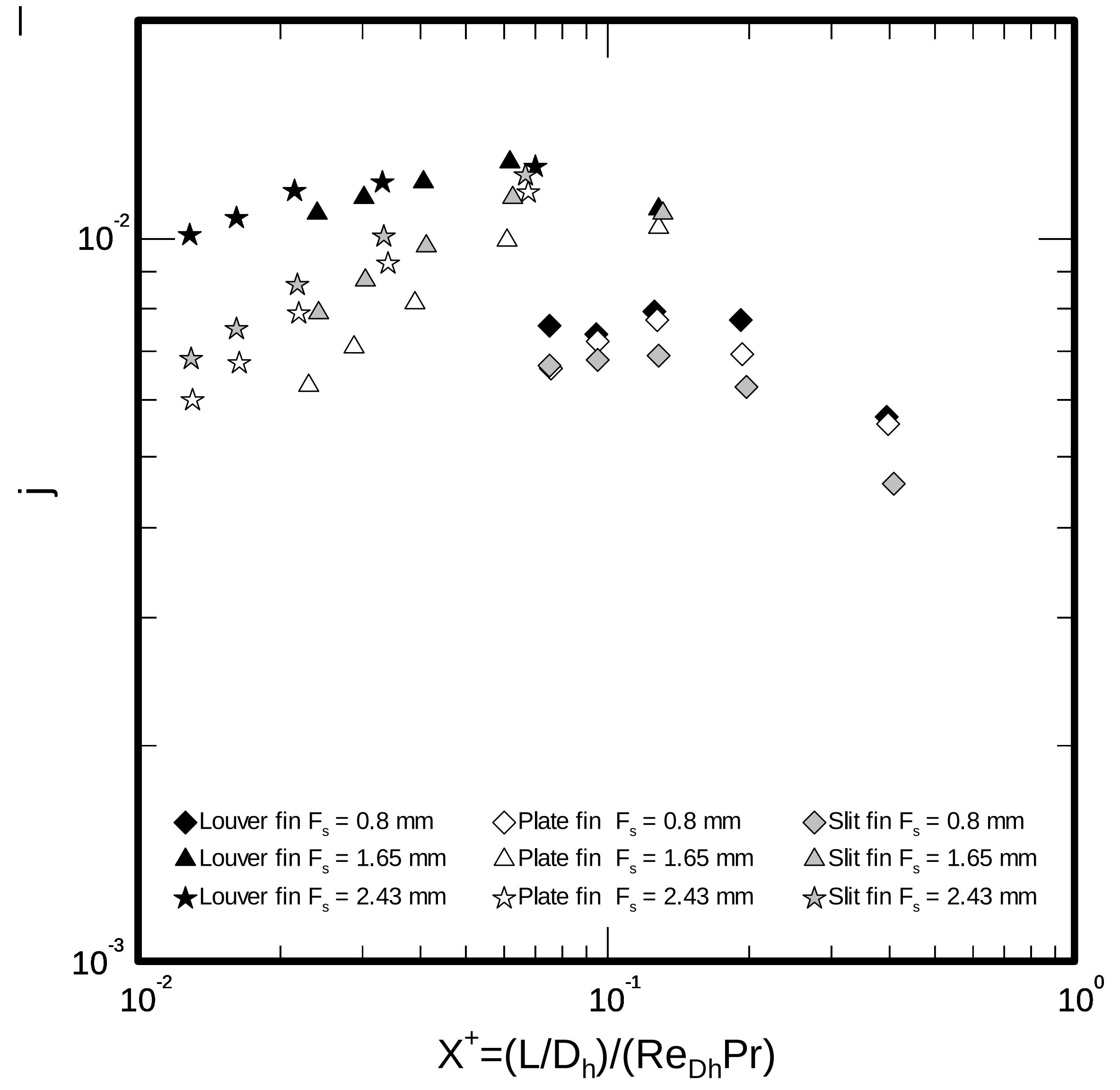
Louver directed vs. fin directed



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

Interrupted surfaces..

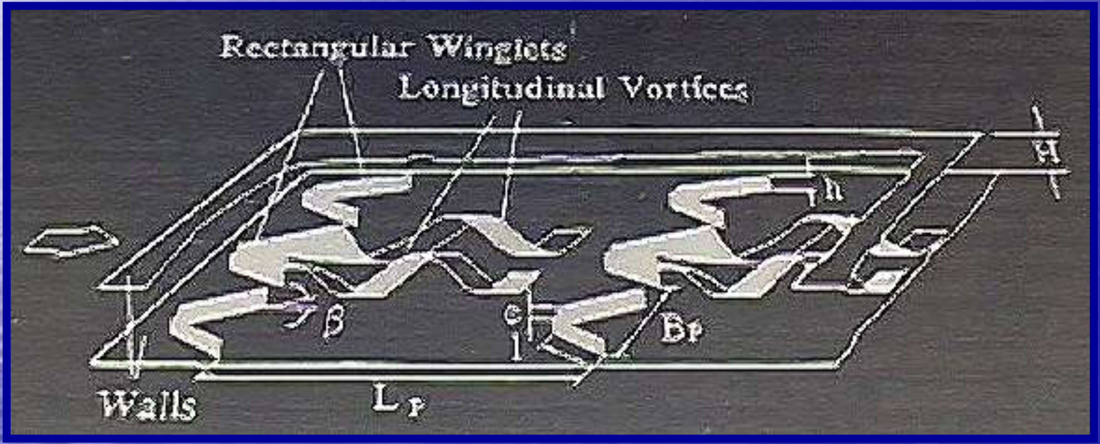
- Smaller fin spacing accentuates the duct flow effect, resulting in fully developed flow and deteriorate the heat transfer performance.



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

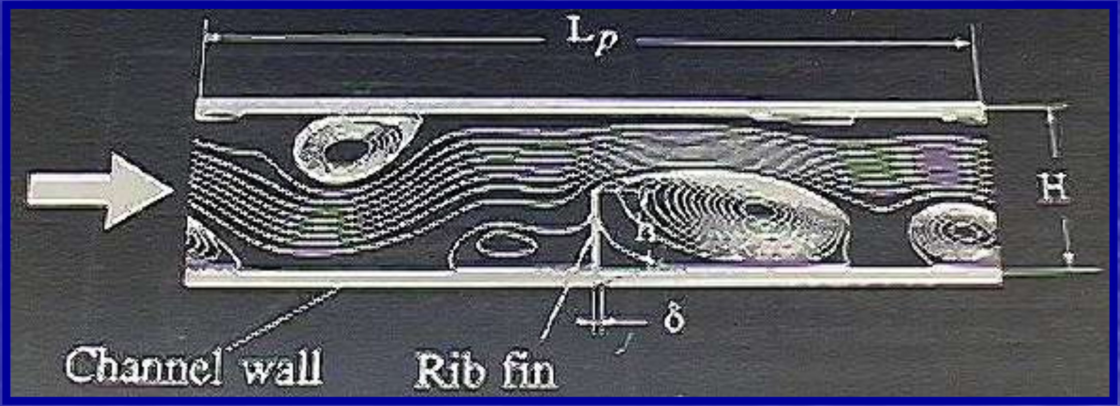
Type of vortex generators

Longitudinal vortex outperforms the transverse vortex



Longitudinal vortex

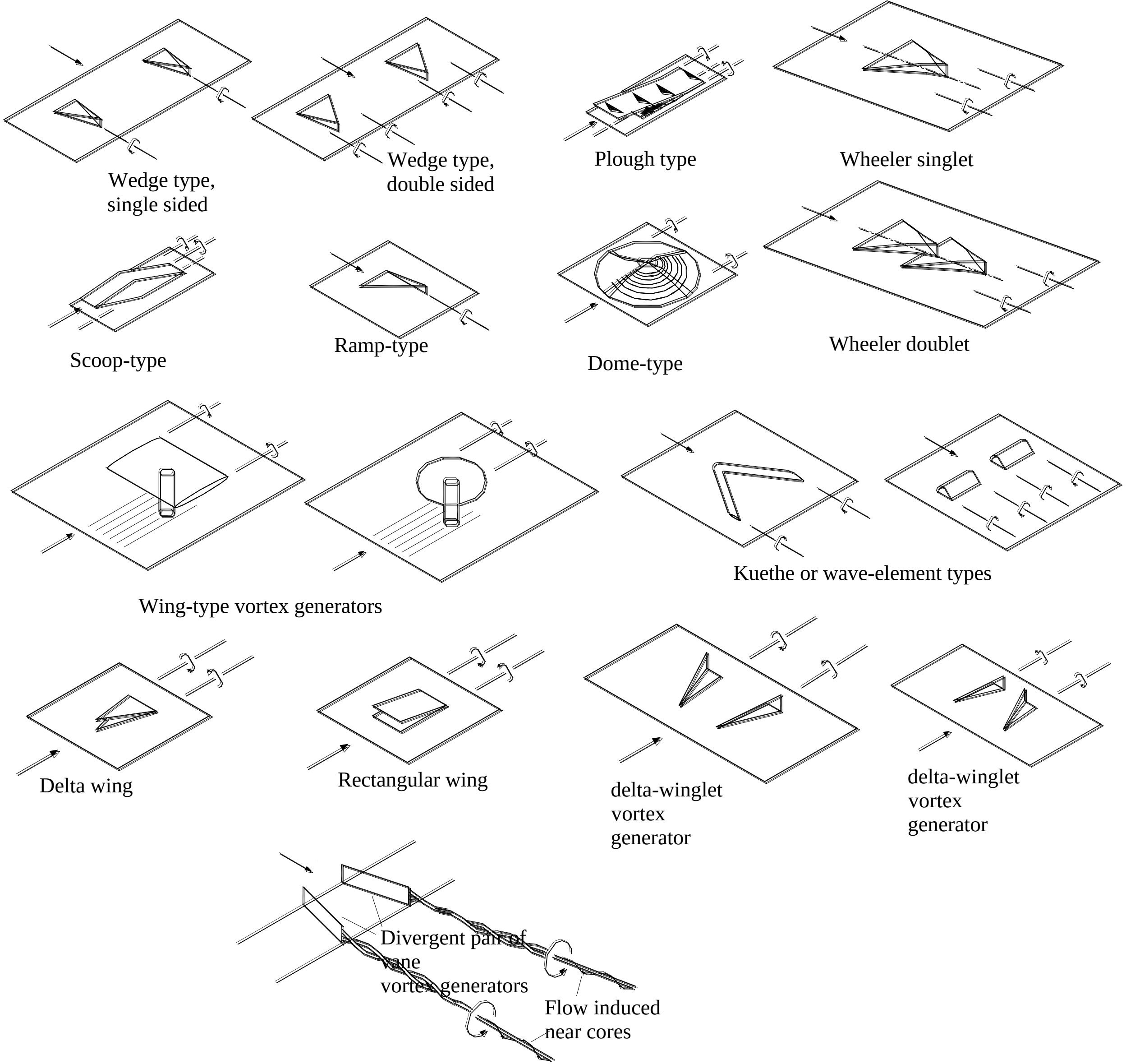
Transverse vortex



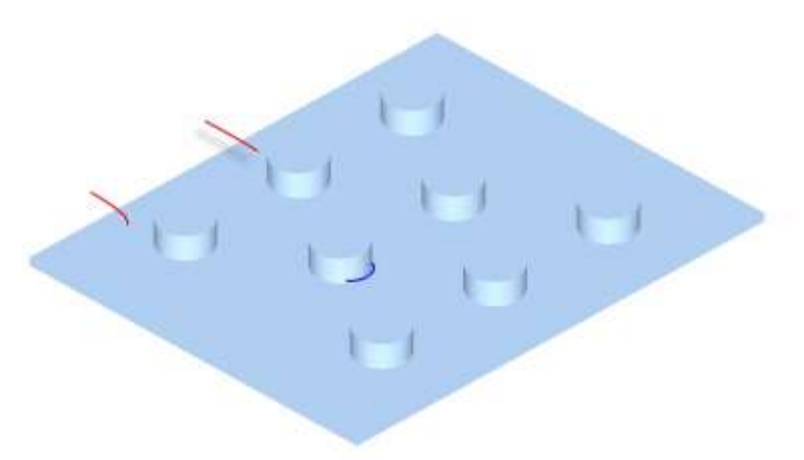
Typical LVGs

Benefits of vortex generator

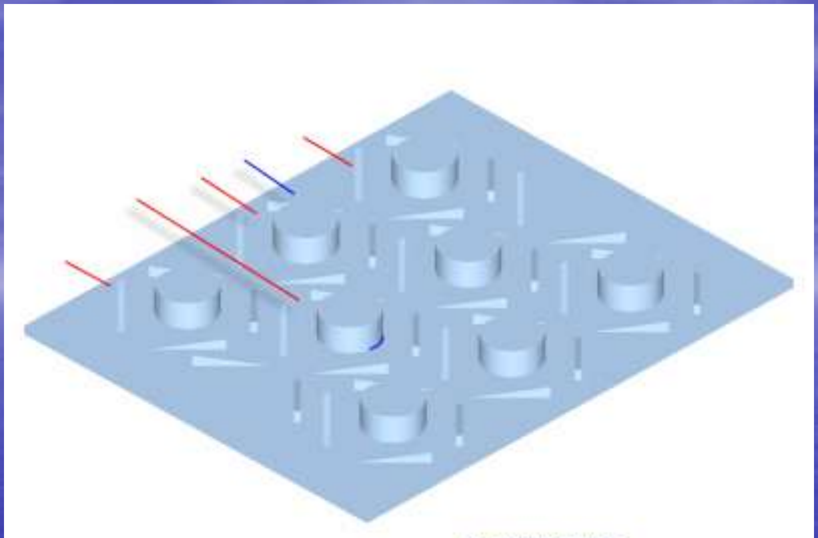
- Prevent Boundary Layer separation
- Improve heat transfer performance with acceptable pressure drop



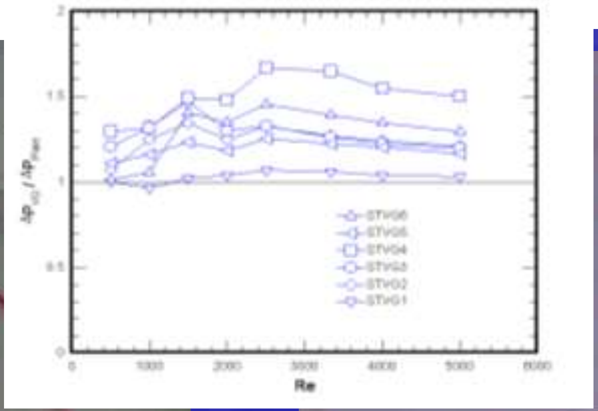
Int. J. of Heat and Mass Transfer, Vol. 45, pp. 1933-1944.
Int. J. of Heat and Mass Transfer, Vol. 45, pp. 3803-3815.

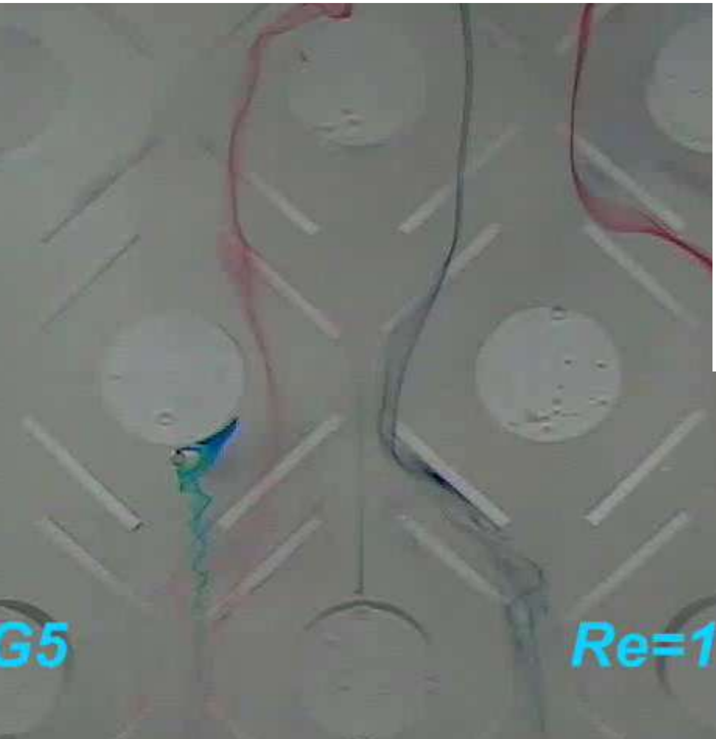


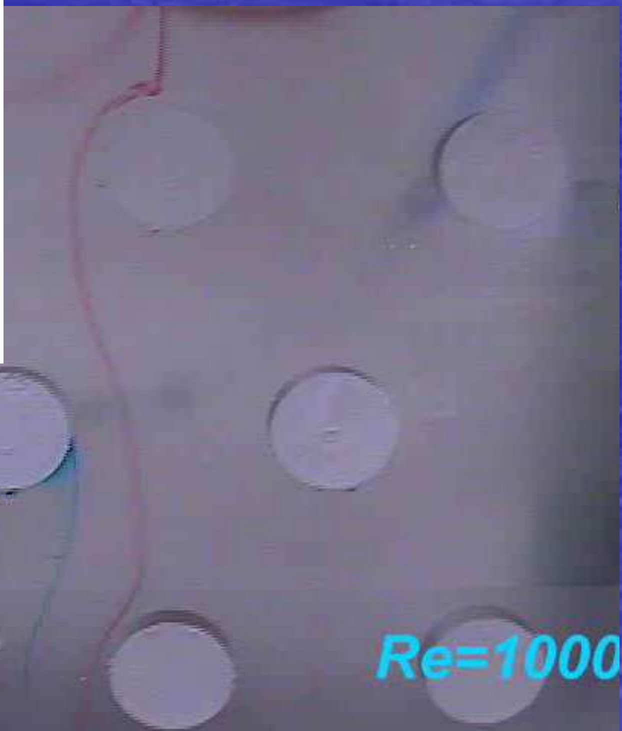
Re=1500,STPL



Re=1000,STVG5

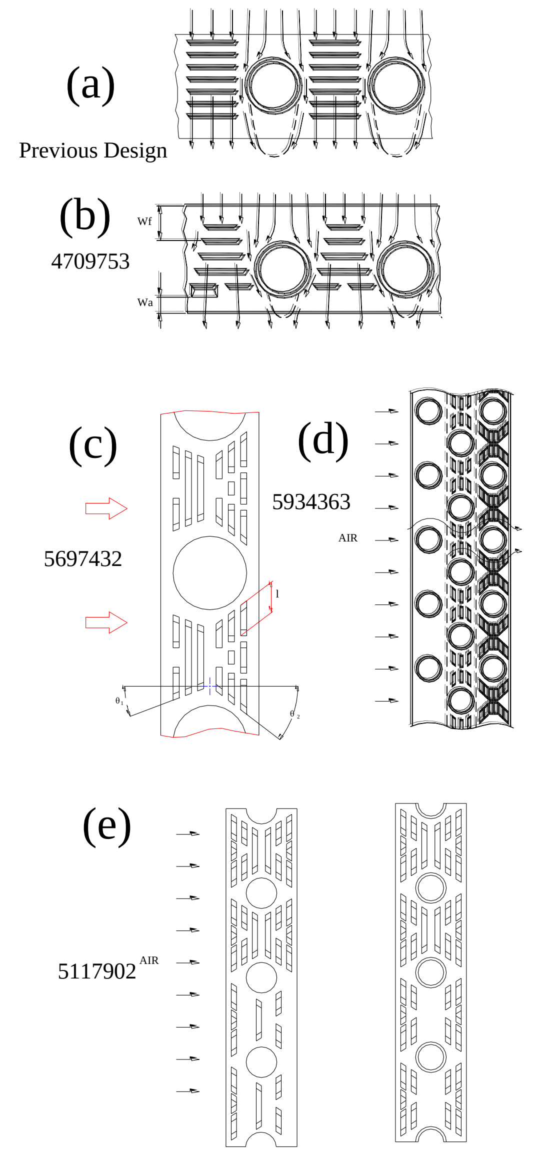






Design by Non-uniformity

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

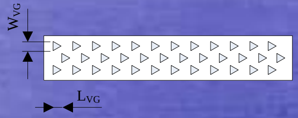
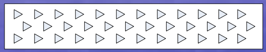
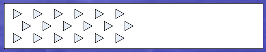
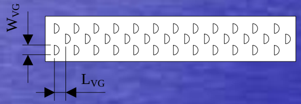
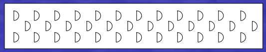
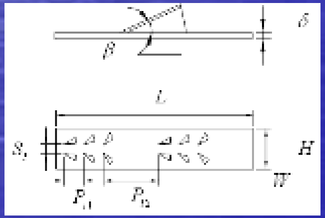
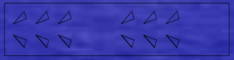
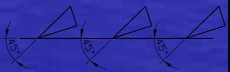
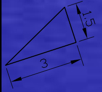
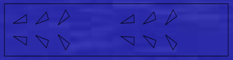
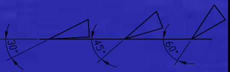
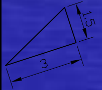
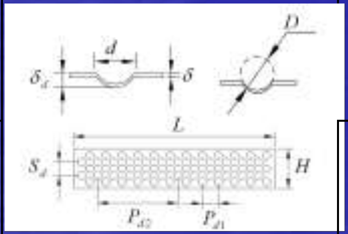
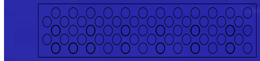
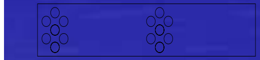


Vortex Generators..

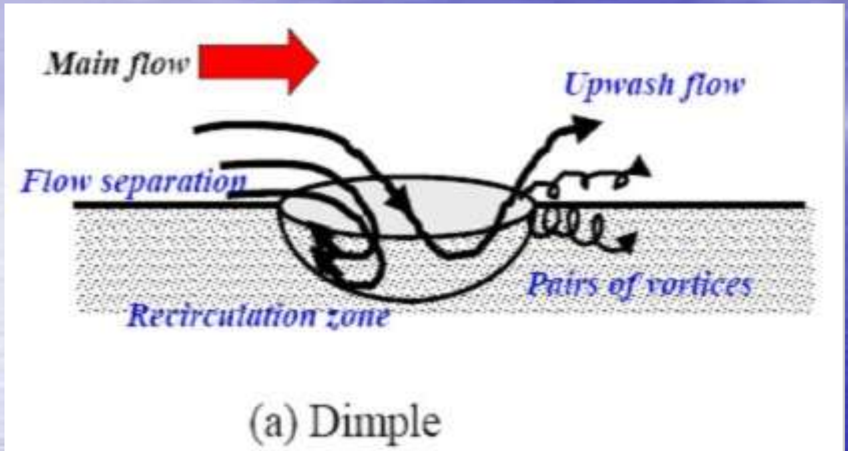
- Implementations

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

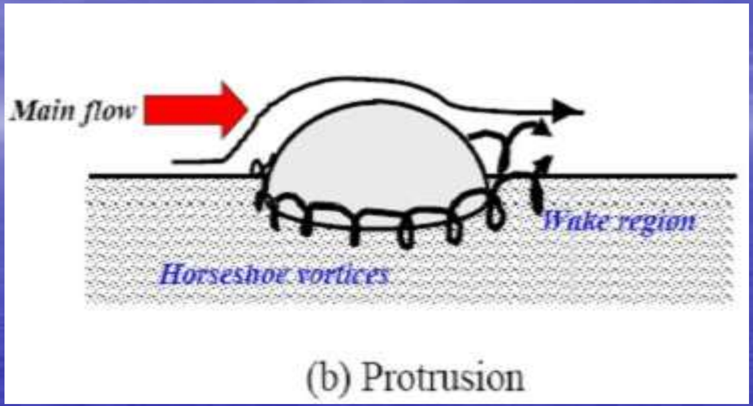
Type IV: Heat sink with loose vortex generator: The enhancements of this category are still vortex generators or dimple/protrusion structure but with sparse arrangement of vortex generator.

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

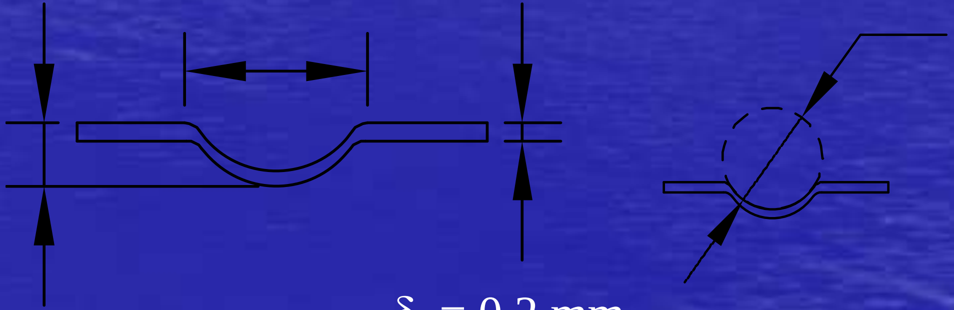
The original concept of Using dimple..



(a) Dimple



(b) Protrusion



d = 2.0 mm

D = 3.05 mm

δ_d = 0.5 mm

δ_f = 0.2 mm

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

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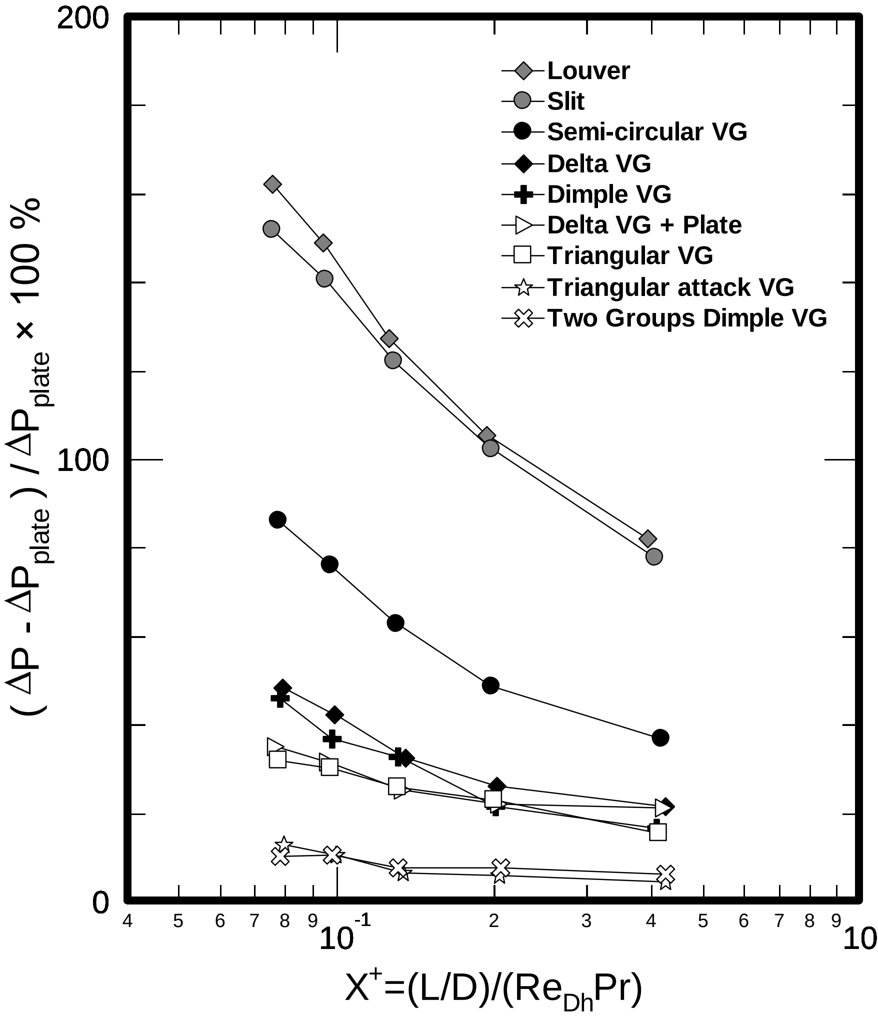
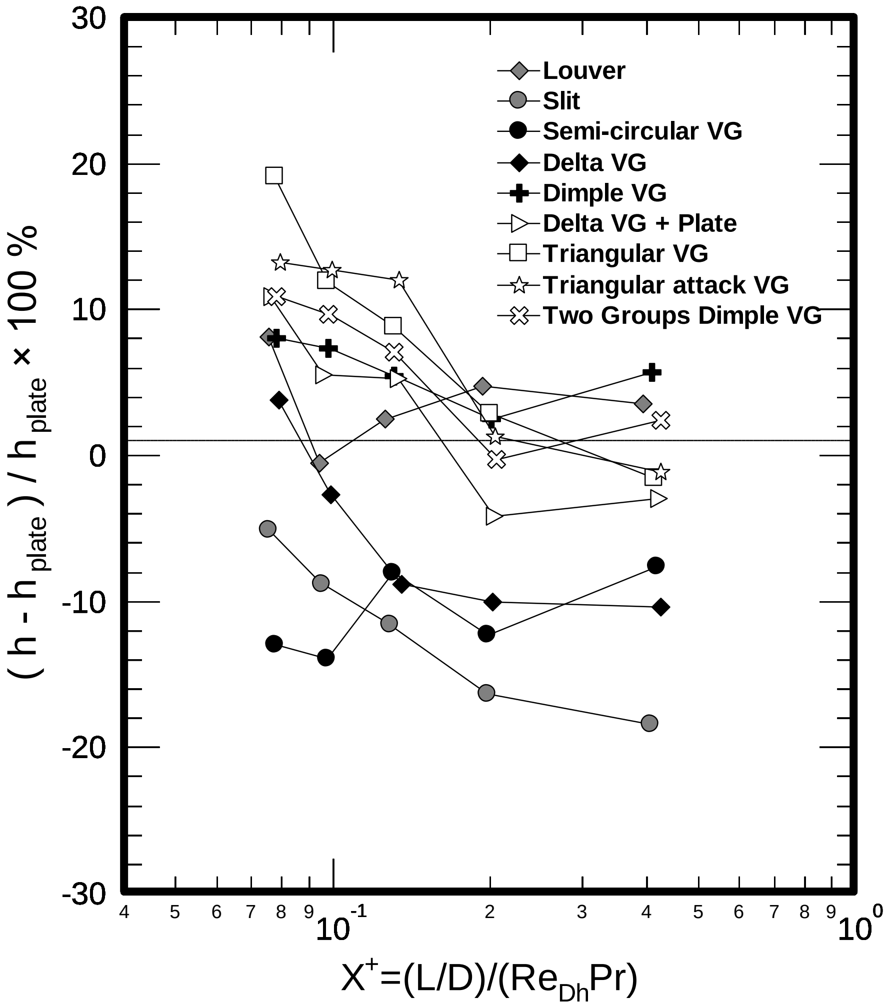
National Chiao Tung University

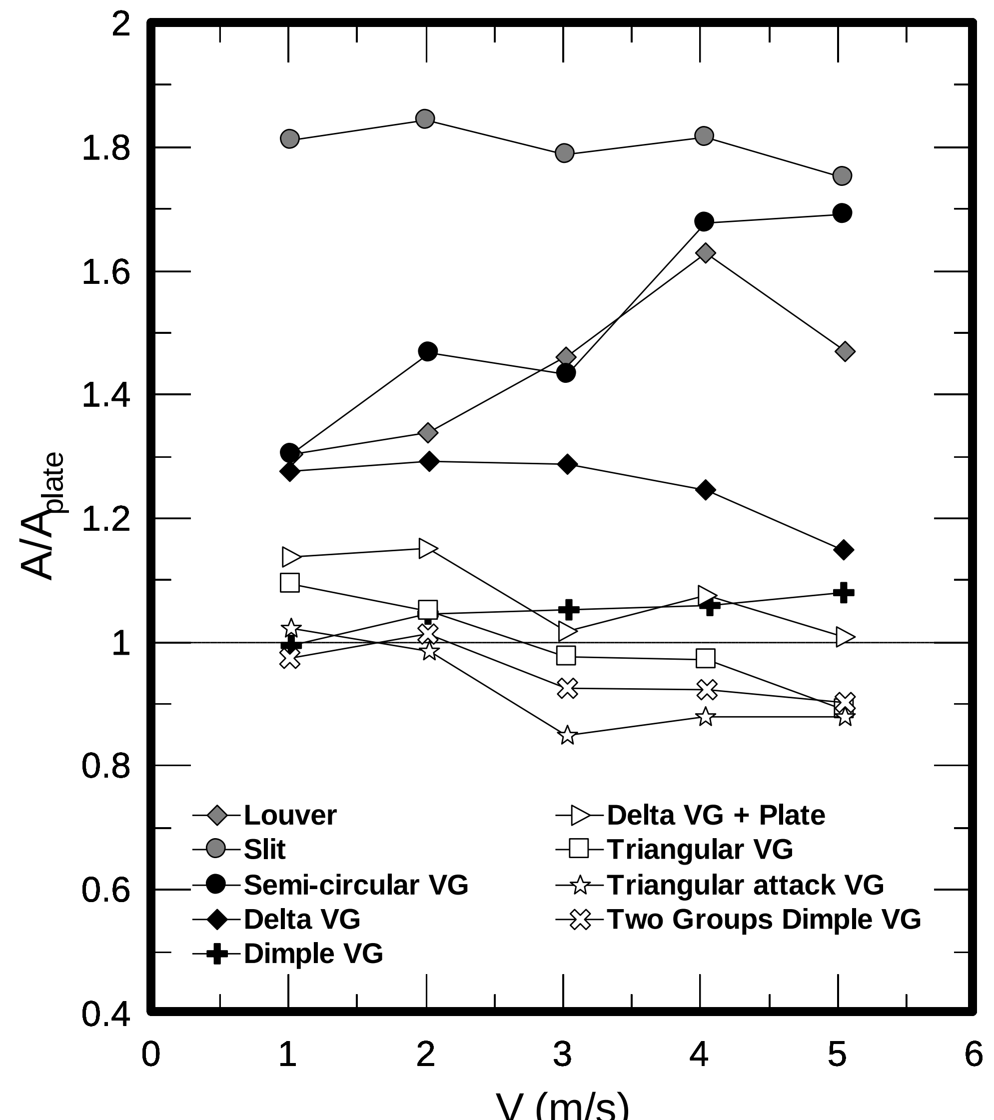
Performance comparison Fin spa



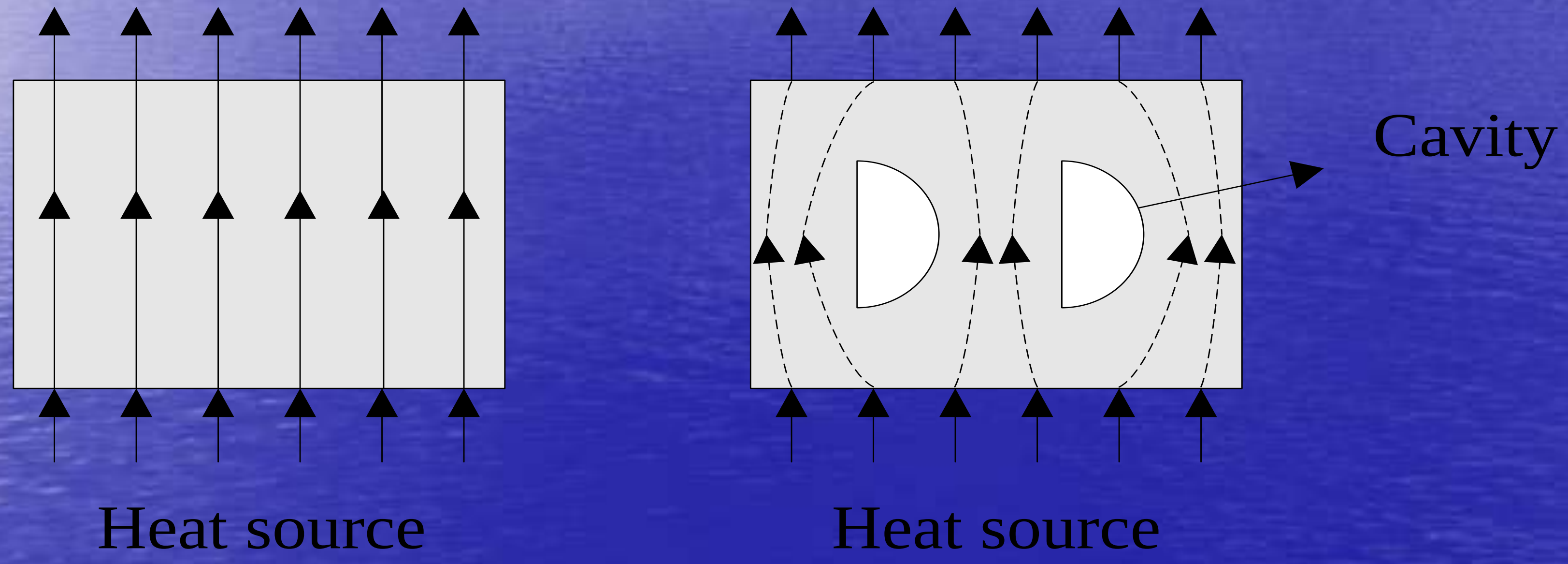


Performance evaluation based on VG-1 Criterion

- Vortex generators fin operated at a higher frontal velocity and arrangeme of loose vortex generator is more beneficial.
- The results show that when frontal velocities as 3~5 m/s and the fin with enhancement as triangular, triangular attack and two-groups dimple effectively reduce required surface are The type II and type III fin geometry possesses the lower heat transfer coefficient in most situations along wi their significant pressure drops lift the out of the choice of vortex generator subject to the VG-1 criteria.
- The asymmetric combination using heat sink with loose vortex generator (Type IV) fin can be quite effective.



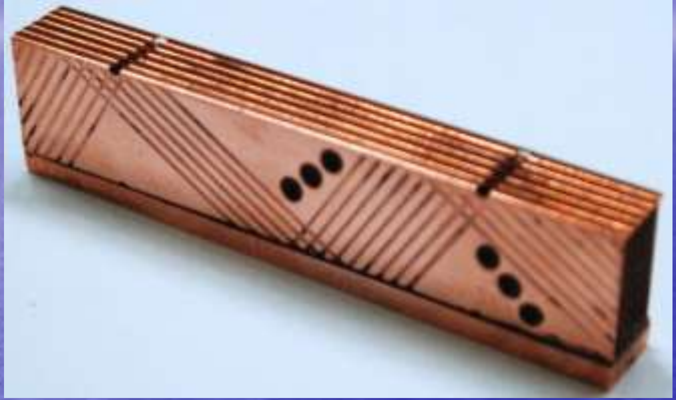
An extra problem for some VG & interrupted surfaces

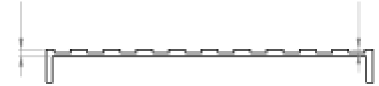


Very small fin spacing also jeopardize the formation of LVG

So, what’s next?

- Oblique Dimples with cannelure structure

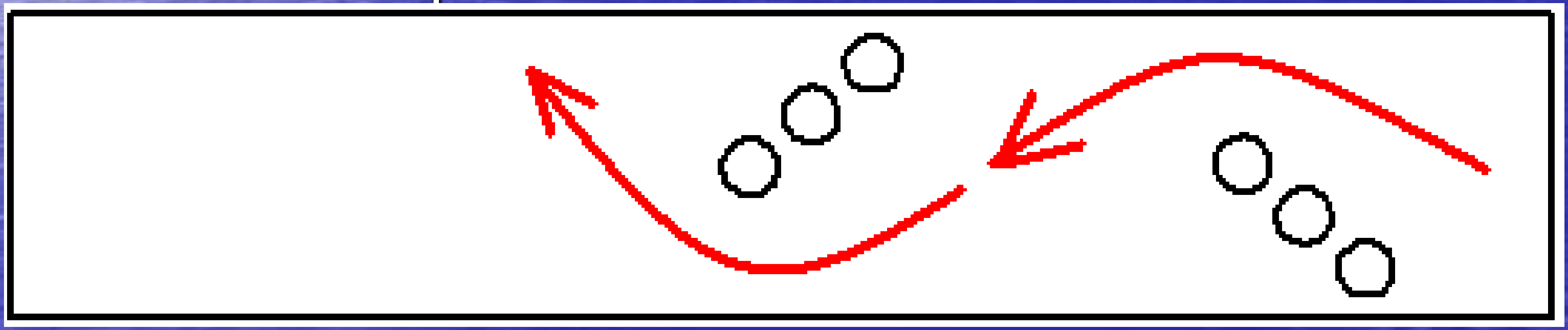
	
（plate fin）	(oblique dimple gap 4-12fin)
	
(oblique dimple gap 6-12 fin)	（cannelure fin I）
	
（cannelure fin II）	(oblique dimple gap 4-12 cannelure fin)
	
(oblique dimple gap 6-12 cannelure fin I)	(oblique dimple gap 6-12 cannelure fin II)
	



Cannelure channel
Depth: 0.1 mm
Width: 0.4 mm

The original idea for oblique dimple..

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



The idea of cannelure channels..

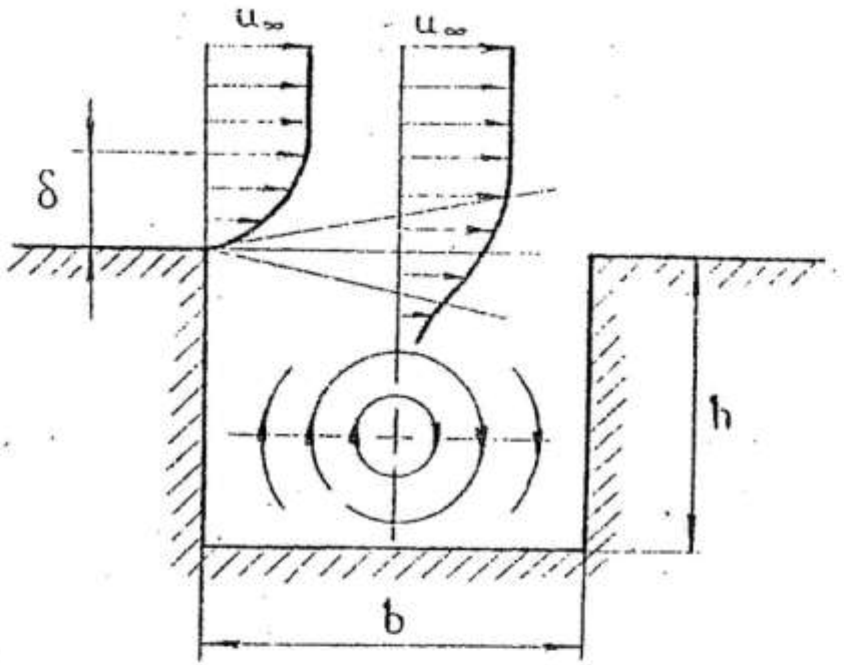


Fig. 1. Model of flow in an isolated rectangular pit [2] in a wall.

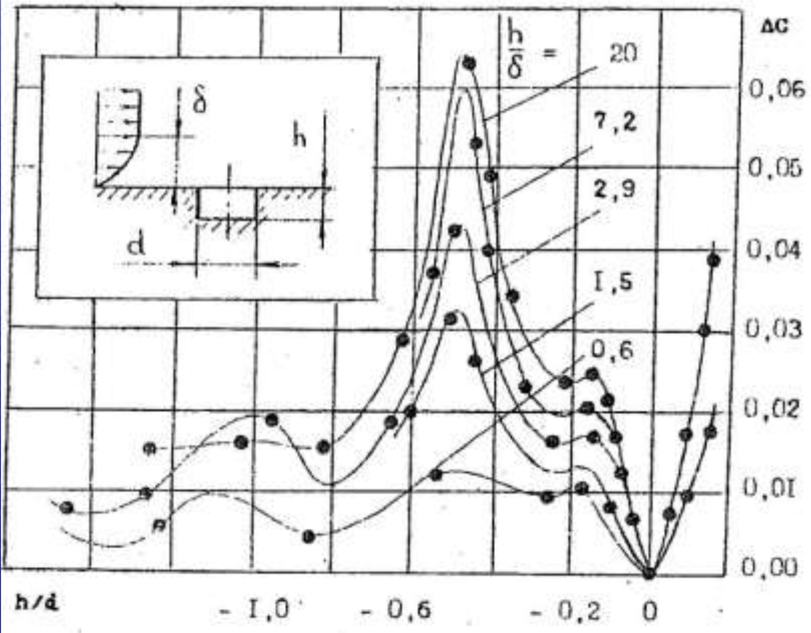


Fig. 2. Increment of the drag coefficient as a function of the dimensionless depth of the pit [4].

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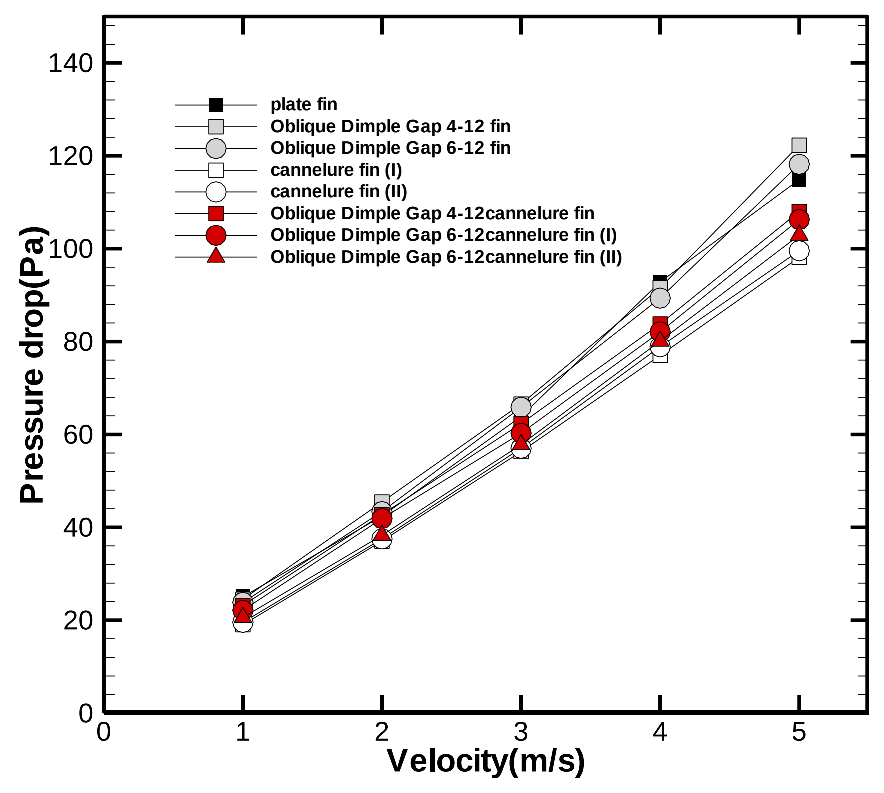
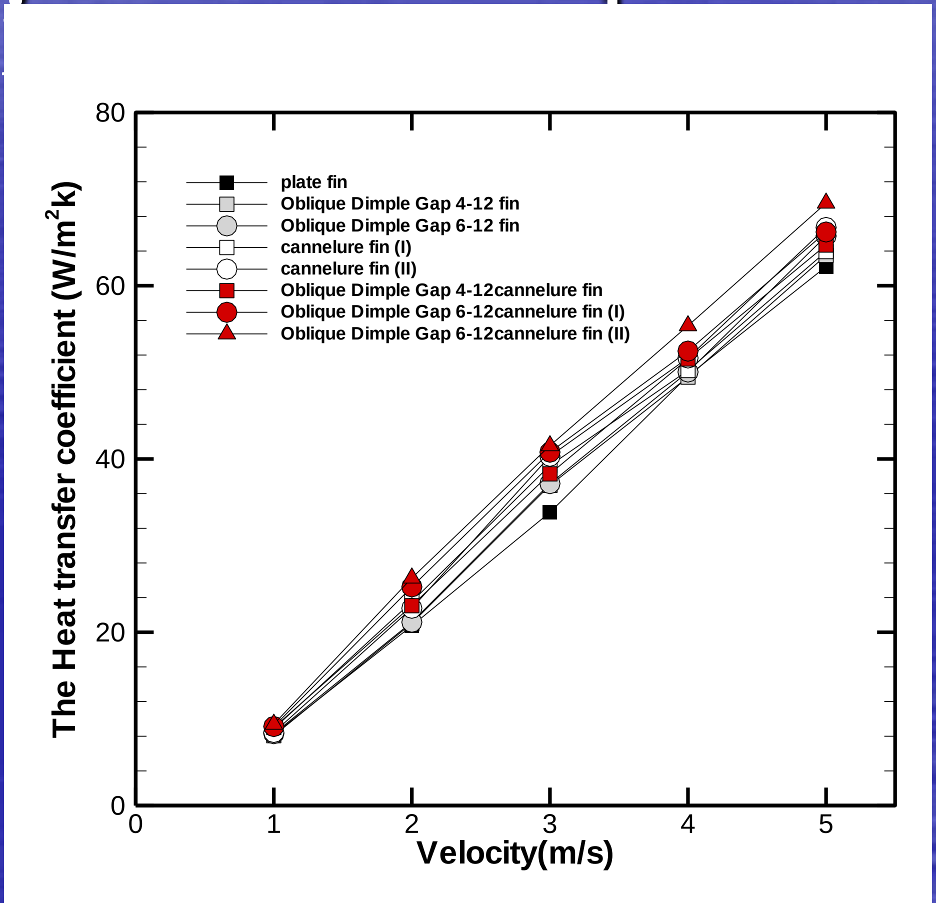
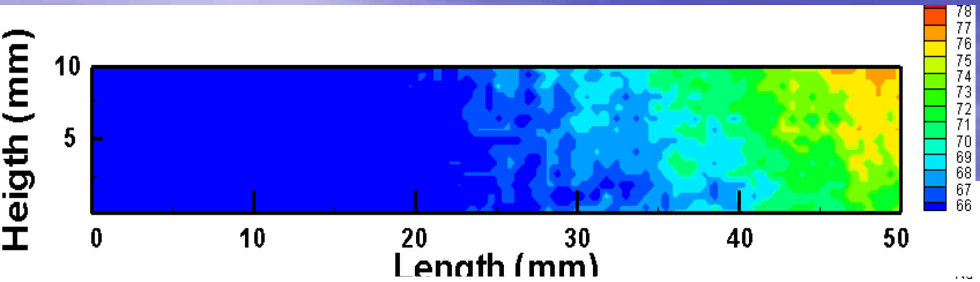
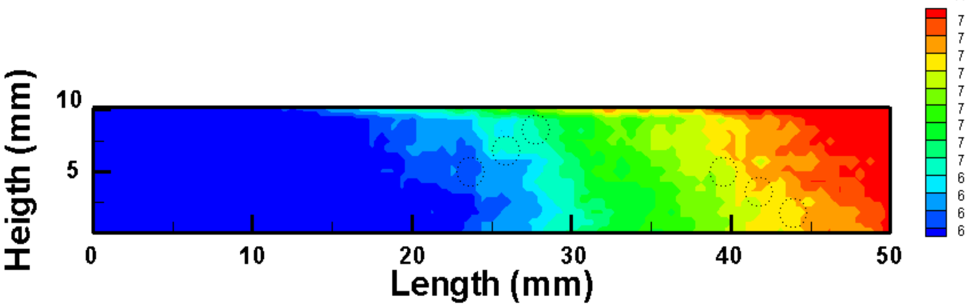
Results:
More than 20% increase HTC &

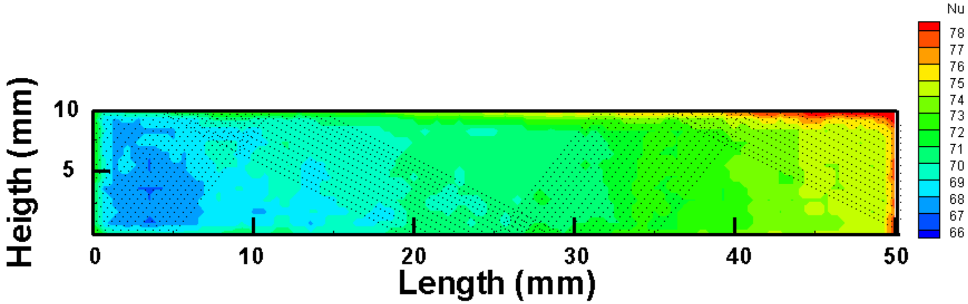


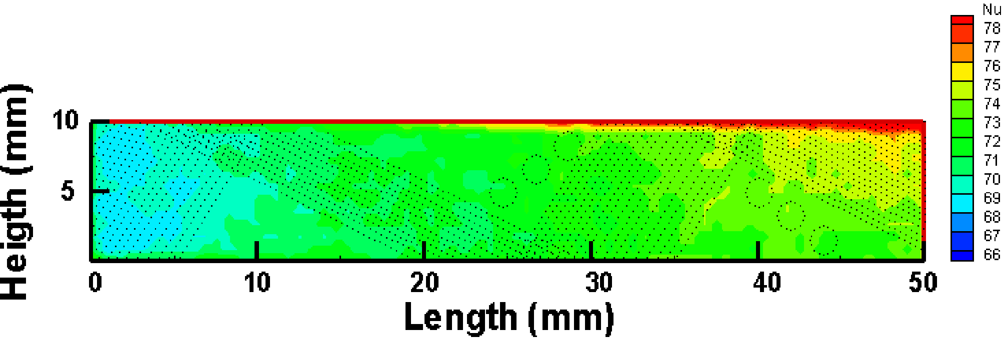


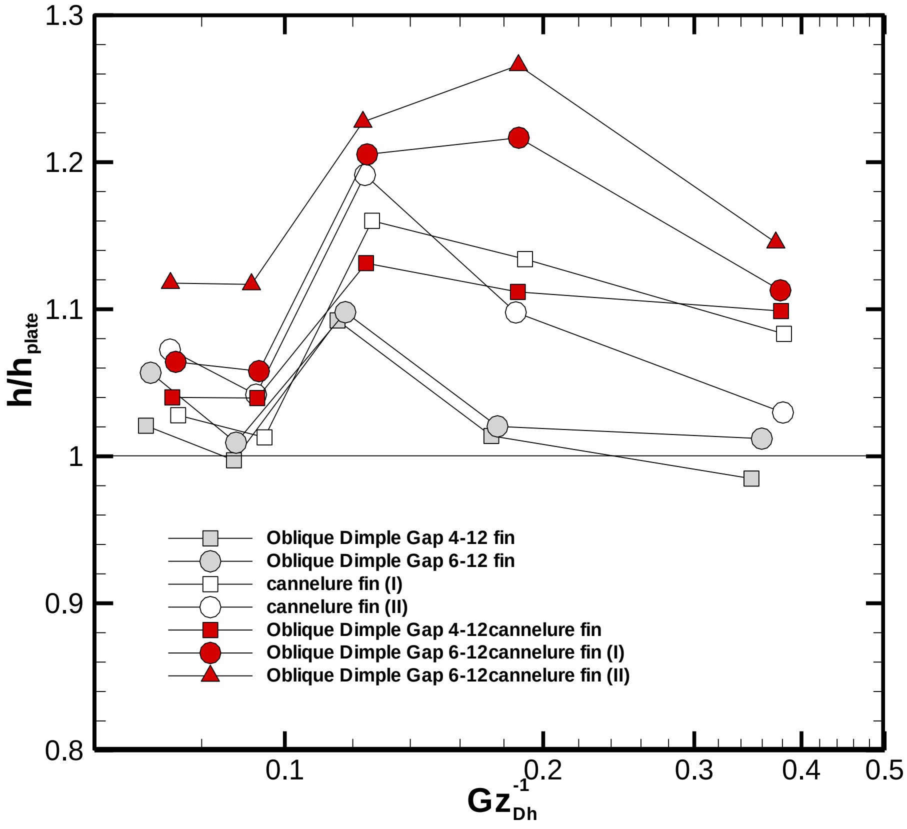
Performance & IR image





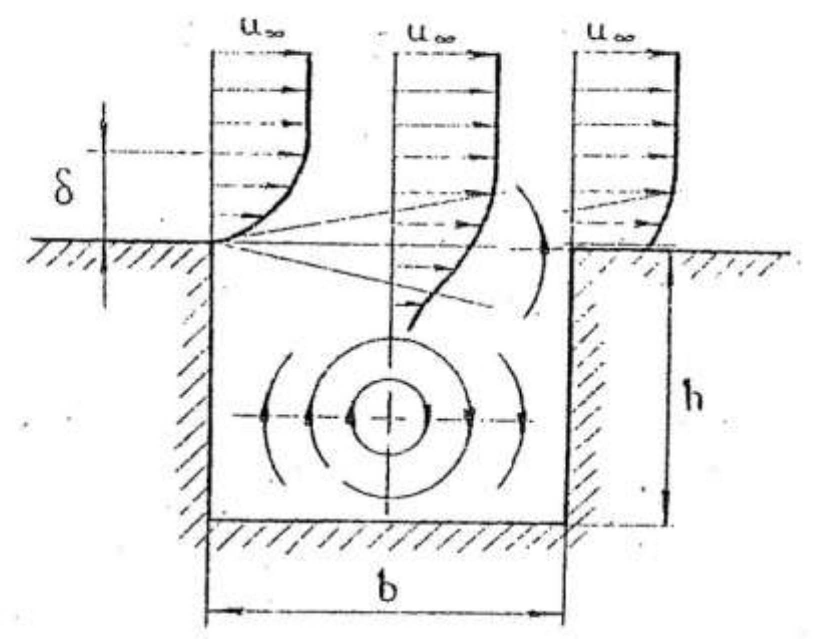






Why cannelure structure is working? – One possible reason

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



Evidence from previous data

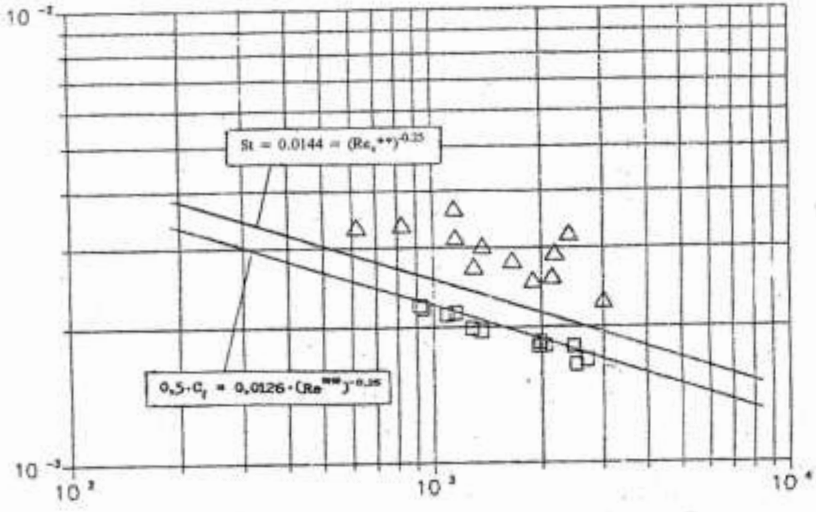


Fig. 13. Friction and heat transfer on indented walls.
□ - the dynamic (velocity) boundary layer; Δ - the thermal boundary layer.

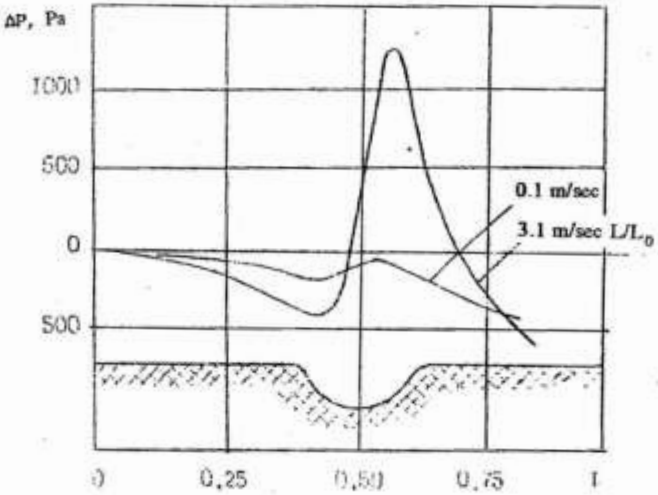
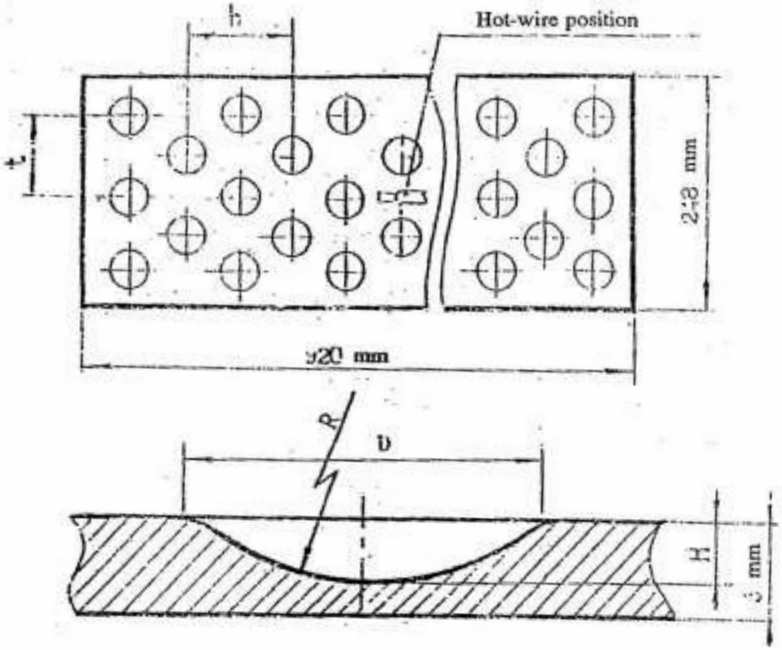
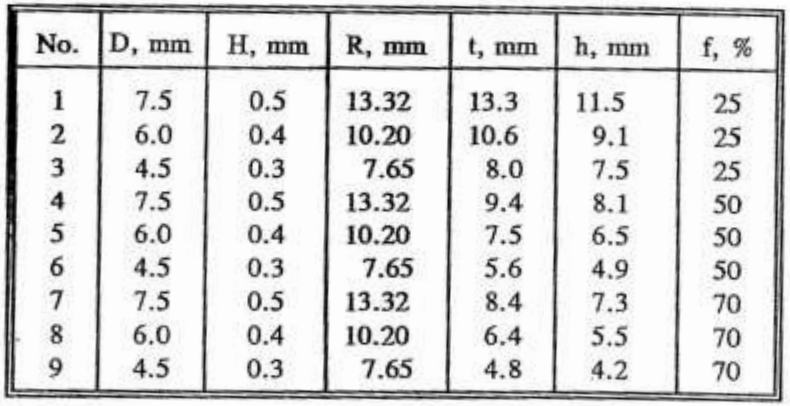


Fig. 10. Distribution of the pressure drop along the axis of a duct with an isolated hemispherical pit in the wall [15].





No.	D, mm	H, mm	R, mm	t, mm	h, mm	f, %
1	7.5	0.5	13.32	13.3	11.5	25
2	6.0	0.4	10.20	10.6	9.1	25
3	4.5	0.3	7.65	8.0	7.5	25
4	7.5	0.5	13.32	9.4	8.1	50
5	6.0	0.4	10.20	7.5	6.5	50
6	4.5	0.3	7.65	5.6	4.9	50
7	7.5	0.5	13.32	8.4	7.3	70
8	6.0	0.4	10.20	6.4	5.5	70
9	4.5	0.3	7.65	4.8	4.2	70

Fig. 12. Geometric parameters of the walls studied.

Conclusions

- The test fin patterns can be classified into four categories, namely the base plain fin heat sink (Type I), interrupted fin geometry (Type II), dense vortex generator (Type III), loose vortex generator (Type IV) and their combinations.
- It is found that the heat transfer performance is strongly related to the developing/fully developed flow characteristics. The result from the present experiment suggests that the asymmetric combination using loose vortex generator arrangement (Type IV) can be quite effective.
- The triangular attack VG is regarded as the optimum enhancement design for it could reduce 12~15% surface area at a frontal velocity of 3 m/s~5 m/s. The asymmetric design is still applicable even when the fin spacing is reduced to 0.8 mm.

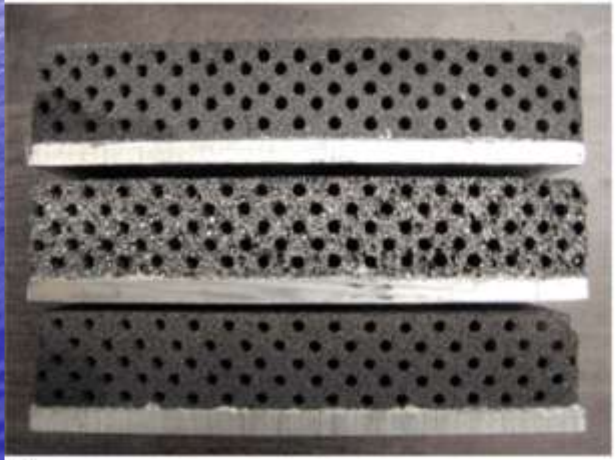
Conclusions

- Combined Con-cavity and dimple is quite effective in heat transfer and pressure drop reduction, provided the numbers are low.
- Cannelure structure may reduce the boundary layer thickness to further reduce pressure drop.
- The cannelure structure is especially effective at fully developed region.
- In the best condition, more than 20% increase in HTC and 20% reduction of pressure drop is achieved.

Carbon & metal Foam

J. of Heat Transfer, 2010, Vol. 132 / **121901-1**

(a)



(b)

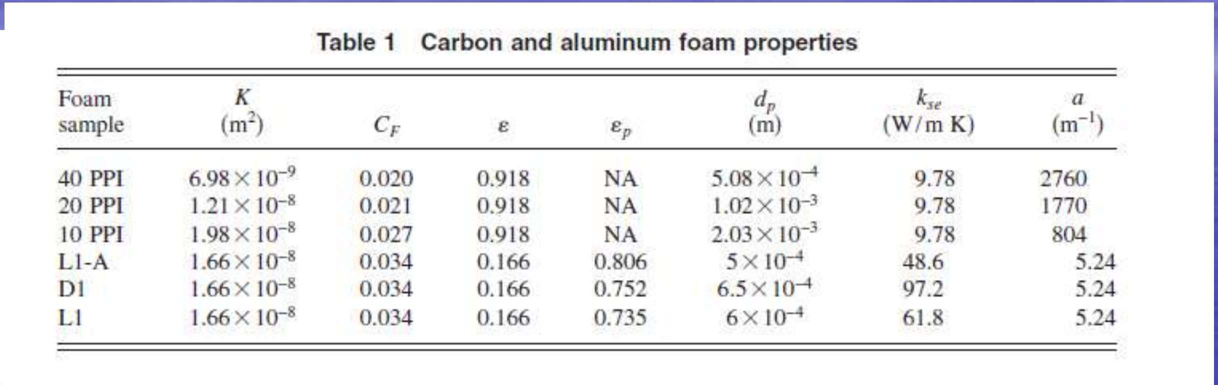


Table 1 Carbon and aluminum foam properties

Foam sample	K (m^2)	C_F	ε	ε_p	d_p (m)	k_{se} (W/m K)	a (m^{-1})
40 PPI	6.98×10^{-9}	0.020	0.918	NA	5.08×10^{-4}	9.78	2760
20 PPI	1.21×10^{-8}	0.021	0.918	NA	1.02×10^{-3}	9.78	1770
10 PPI	1.98×10^{-8}	0.027	0.918	NA	2.03×10^{-3}	9.78	804
L1-A	1.66×10^{-8}	0.034	0.166	0.806	5×10^{-4}	48.6	5.24
D1	1.66×10^{-8}	0.034	0.166	0.752	6.5×10^{-4}	97.2	5.24
L1	1.66×10^{-8}	0.034	0.166	0.735	6×10^{-4}	61.8	5.24

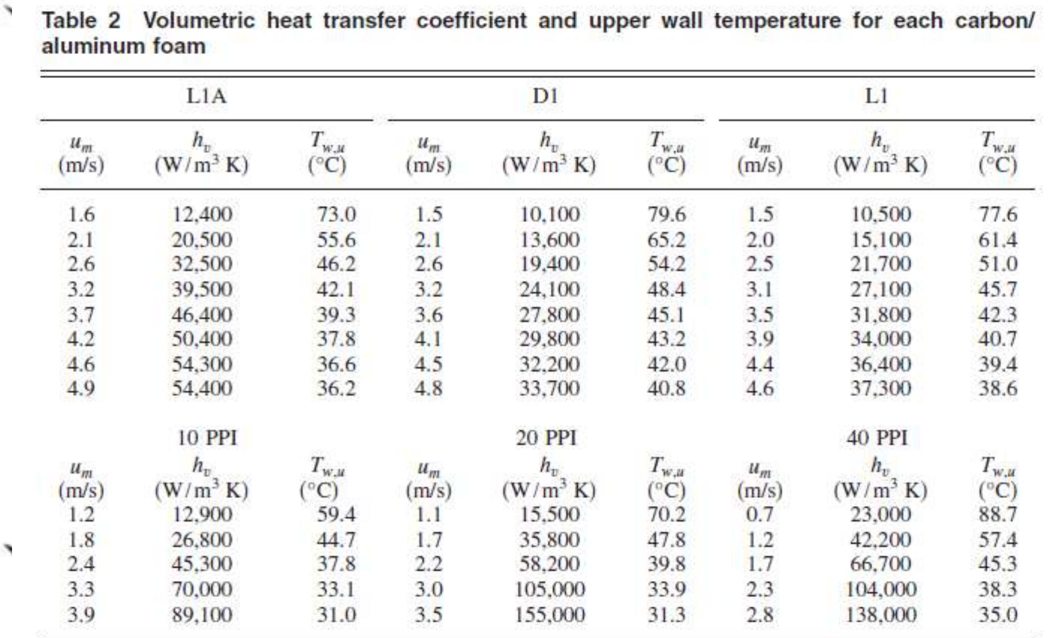


Table 2 Volumetric heat transfer coefficient and upper wall temperature for each carbon/aluminum foam

L1A			D1			L1		
u_m (m/s)	h_v (W/m^3 K)	$T_{w,u}$ (°C)	u_m (m/s)	h_v (W/m^3 K)	$T_{w,u}$ (°C)	u_m (m/s)	h_v (W/m^3 K)	$T_{w,u}$ (°C)
1.6	12,400	73.0	1.5	10,100	79.6	1.5	10,500	77.6
2.1	20,500	55.6	2.1	13,600	65.2	2.0	15,100	61.4
2.6	32,500	46.2	2.6	19,400	54.2	2.5	21,700	51.0
3.2	39,500	42.1	3.2	24,100	48.4	3.1	27,100	45.7
3.7	46,400	39.3	3.6	27,800	45.1	3.5	31,800	42.3
4.2	50,400	37.8	4.1	29,800	43.2	3.9	34,000	40.7
4.6	54,300	36.6	4.5	32,200	42.0	4.4	36,400	39.4
4.9	54,400	36.2	4.8	33,700	40.8	4.6	37,300	38.6
10 PPI			20 PPI			40 PPI		
u_m (m/s)	h_v (W/m^3 K)	$T_{w,u}$ (°C)	u_m (m/s)	h_v (W/m^3 K)	$T_{w,u}$ (°C)	u_m (m/s)	h_v (W/m^3 K)	$T_{w,u}$ (°C)
1.2	12,900	59.4	1.1	15,500	70.2	0.7	23,000	88.7
1.8	26,800	44.7	1.7	35,800	47.8	1.2	42,200	57.4
2.4	45,300	37.8	2.2	58,200	39.8	1.7	66,700	45.3
3.3	70,000	33.1	3.0	105,000	33.9	2.3	104,000	38.3
3.9	89,100	31.0	3.5	155,000	31.3	2.8	138,000	35.0

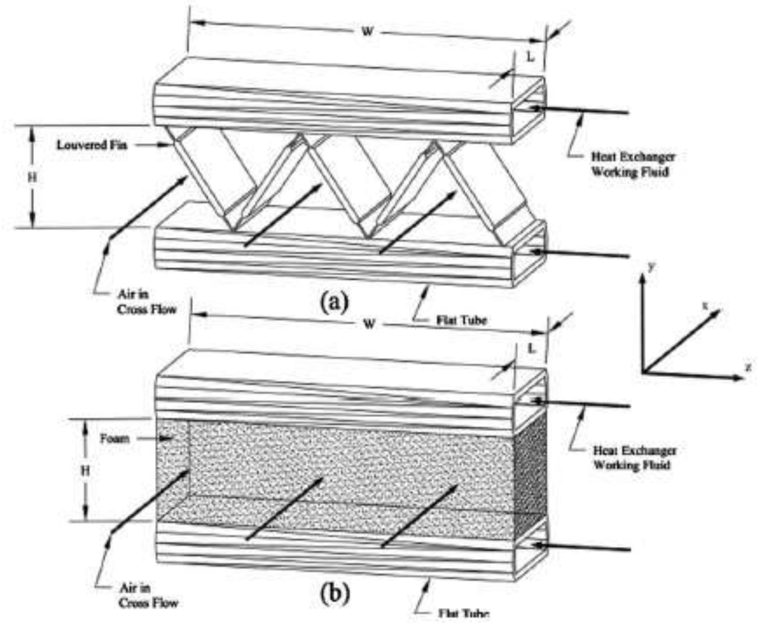


Fig. 13 Hypothetical heat exchanger in cross flow with (*a*) louvered fin and (*b*) carbon/aluminum foam configurations

$$COP = \frac{\dot{Q}_{\mathrm{removed}}}{\dot{E}_{\mathrm{input}}}$$

$$CF = \frac{\dot{Q}_{\mathrm{removed}}}{V}$$

$$PD = \frac{\dot{Q}_{\mathrm{removed}}}{m}$$

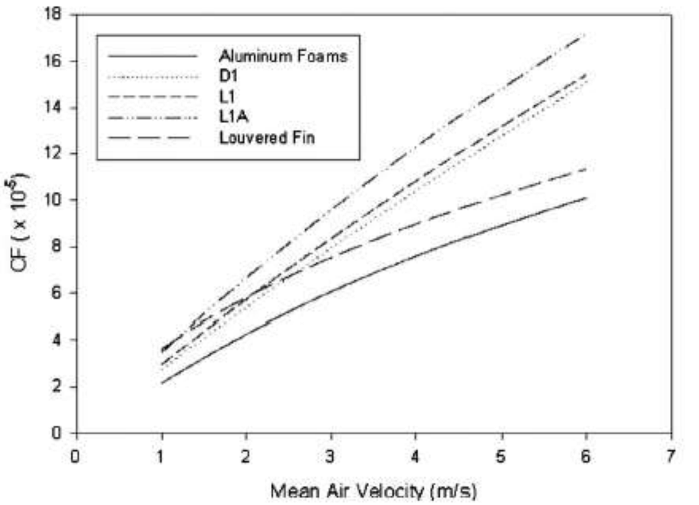


Fig. 16 Comparison of compactness factor for louvered fin and foam configurations

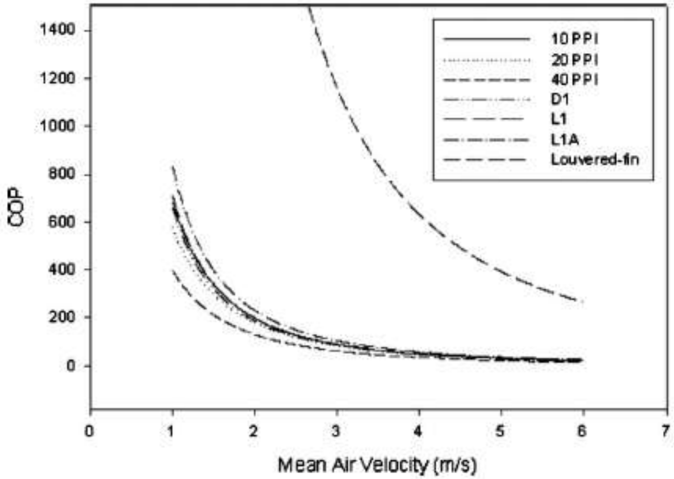


Fig. 15 Comparison of coefficient of performance for louvered fin and foam configurations

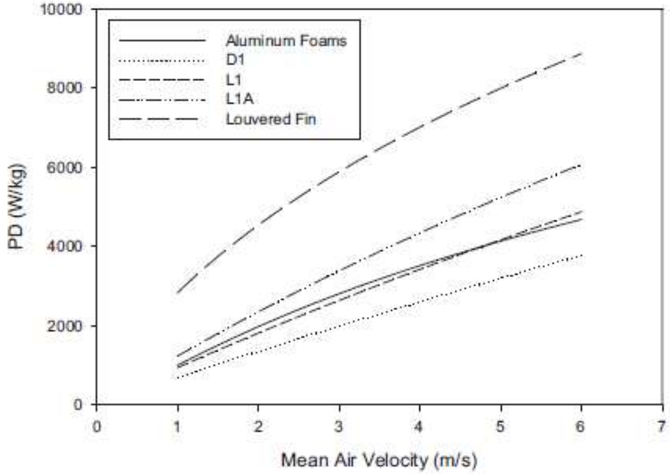
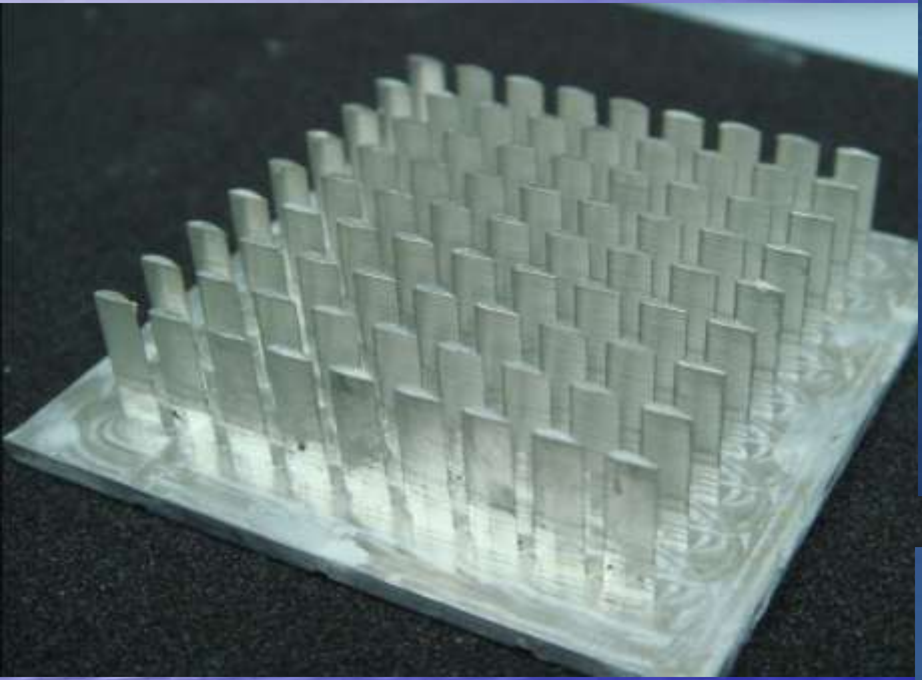
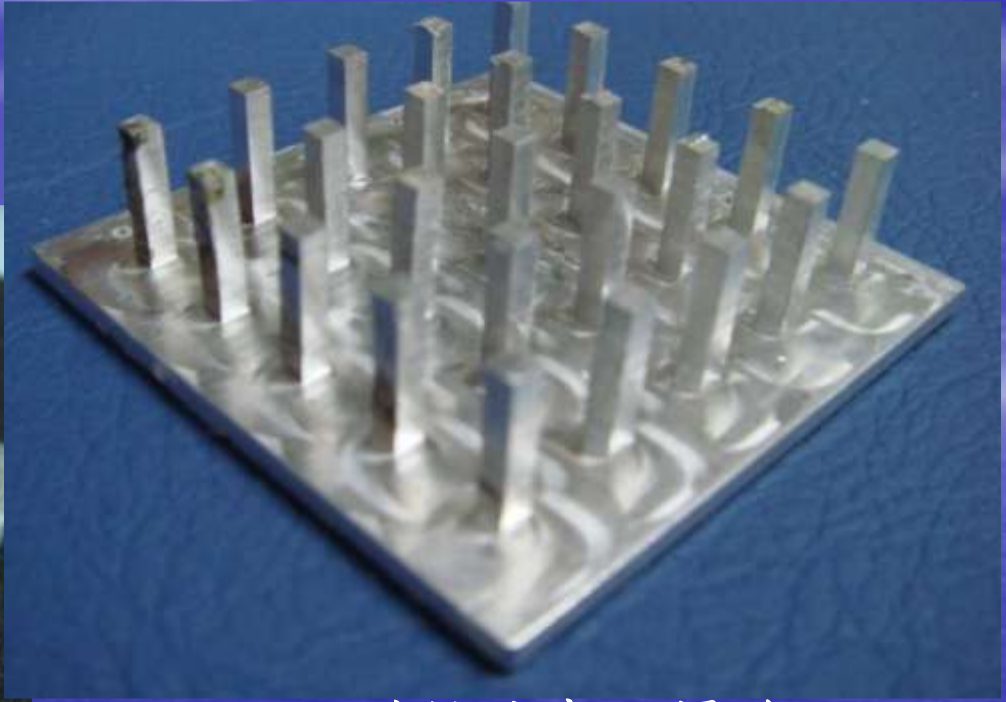


Fig. 17 Comparison of power density for louvered fin and foam configurations

機翼型針鰭9*9

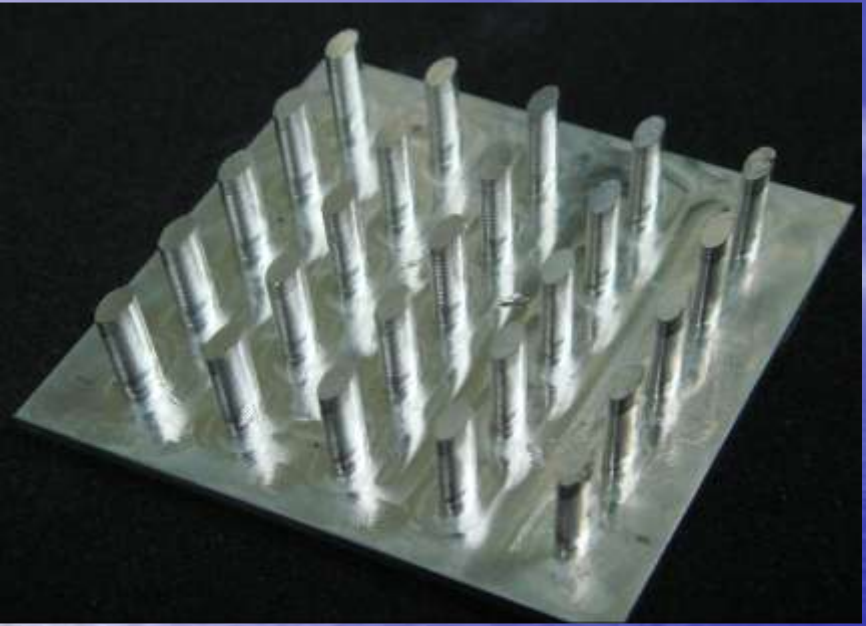




方型針鰭未加擾流

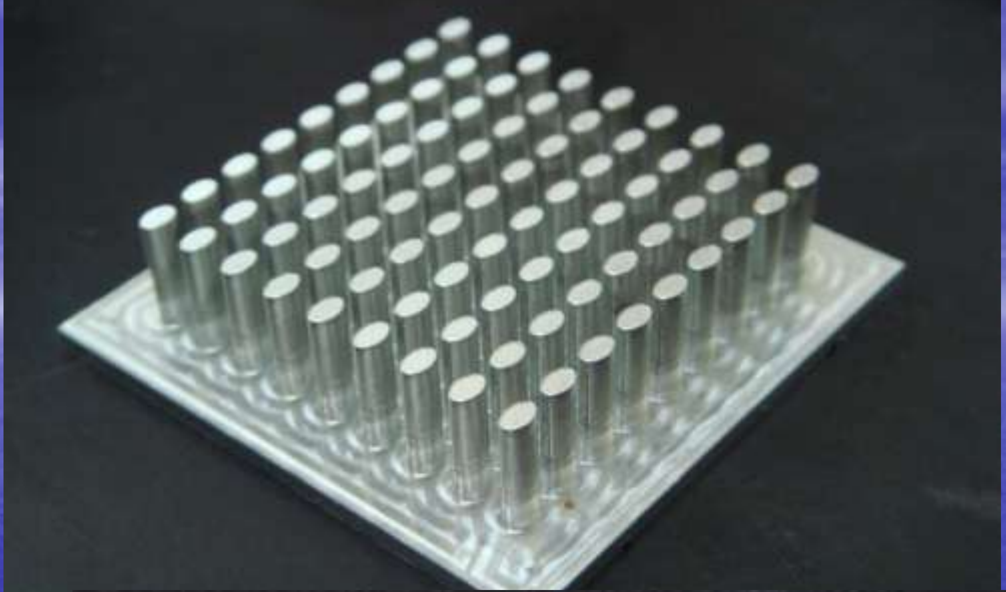
- Pin Fins …

方型針鰭加擾流



橢圓型針鰭5*5

橢圓型針鰭9*9

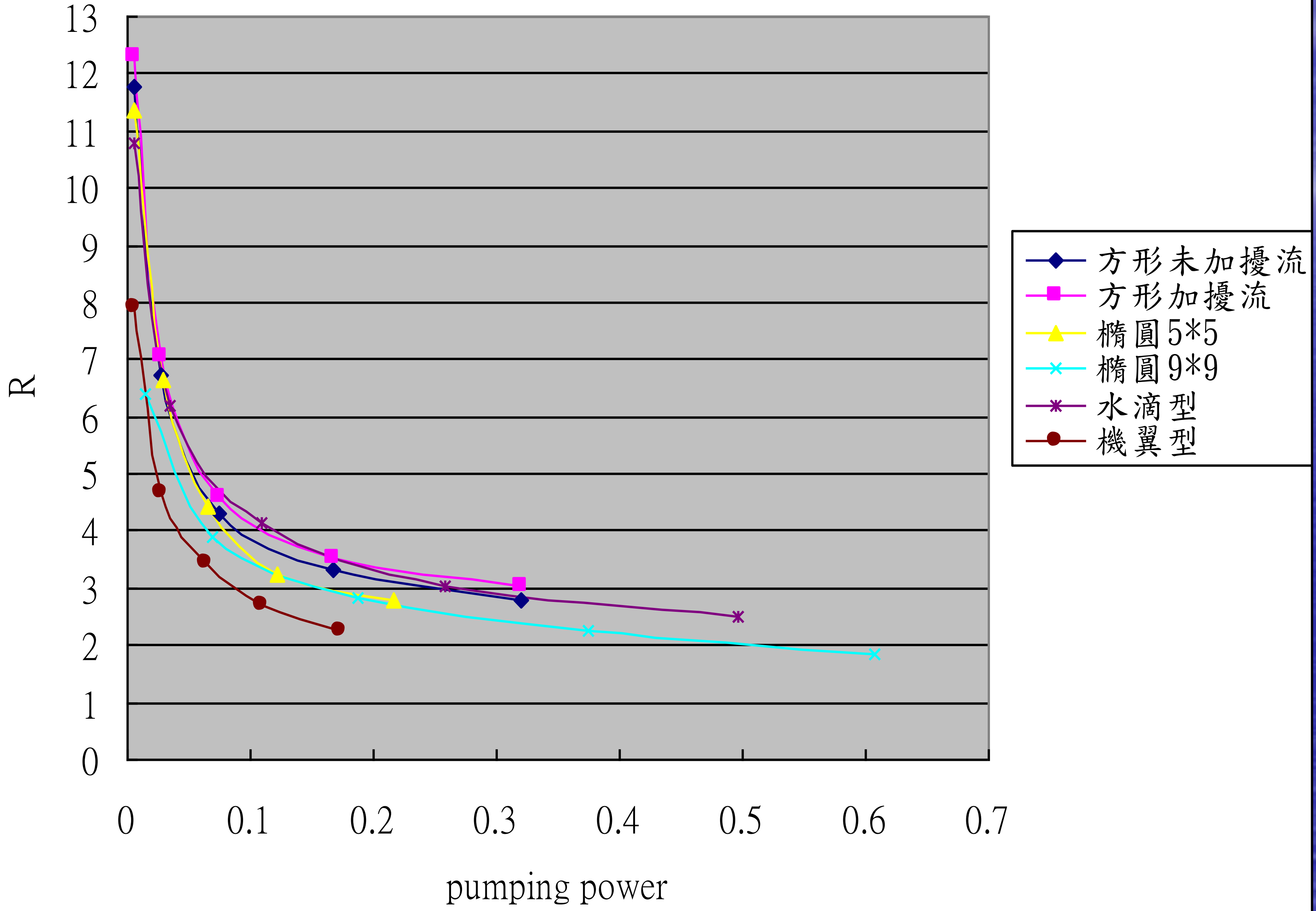


水滴型針鰭5*5

所有形狀尺寸

	鰭片數	進入排數	底面長	底面寬（W）	水滴半圓（ r）	水滴三角（L）	邊長	鰭片長軸（a）	鰭片短軸（b）	鰭片高（H）	Atotal	Ac	Asingal（fin）
橢圓	9x9	8	0.045	0.035	0	0	0	0.003	0.002	0.01	1.2899E-02	1.8849E-05	1.5865E-04
橢圓	5x5	4	0.045	0.035	0	0	0	0.004	0.002	0.01	5.7908E-03	2.5132E-05	1.9376E-04
方型加擾流	5x5	4	0.045	0.035	0	0	0.002	0	0	0.01	3.2750E-03	4.0000E-06	8.0000E-05
方型未加擾流	5x5	4	0.045	0.035	0	0	0.002	0	0	0.01	3.2750E-03	4.0000E-06	8.0000E-05
水滴型	5x5	4	0.045	0.035	0.0015	0.004	0	0	0	0.01	4.6507E-03	9.5342E-06	1.3256E-04
機翼型	9x9	8	0.045	0.035	0	0	0	0	0	0.01	6.6695E-03	1.5613E-06	6.4456E-05

pumping power和R之比較



Summary

- Natural convection and its augmentation.
- Forced convection concerning passive heat transfer augmentation.
- Influence of special fin patterns are presented and compared.
- For more effective fin design, consider the design by Non-uniformity.

Thanks for Your Attention

Questions?