

典型蒸發與冷凝管

Shell side boiling

Shell side condensation

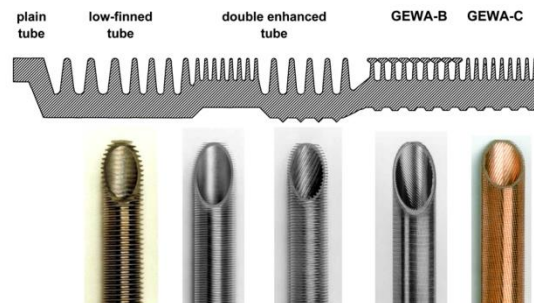
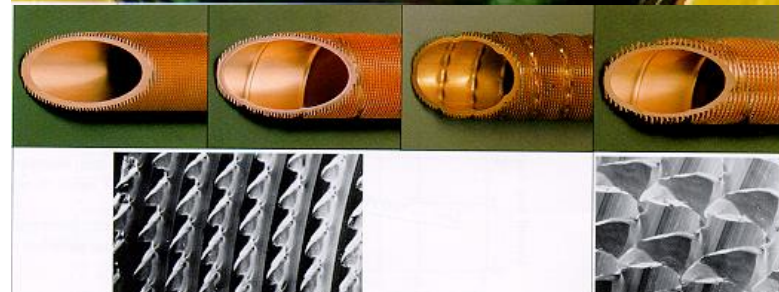
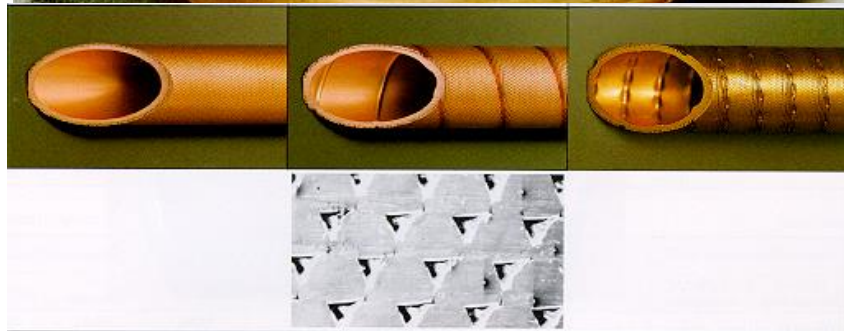
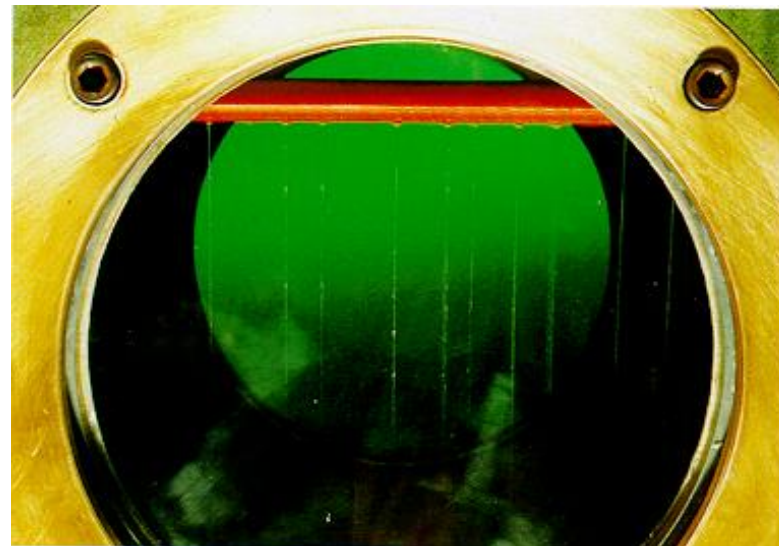
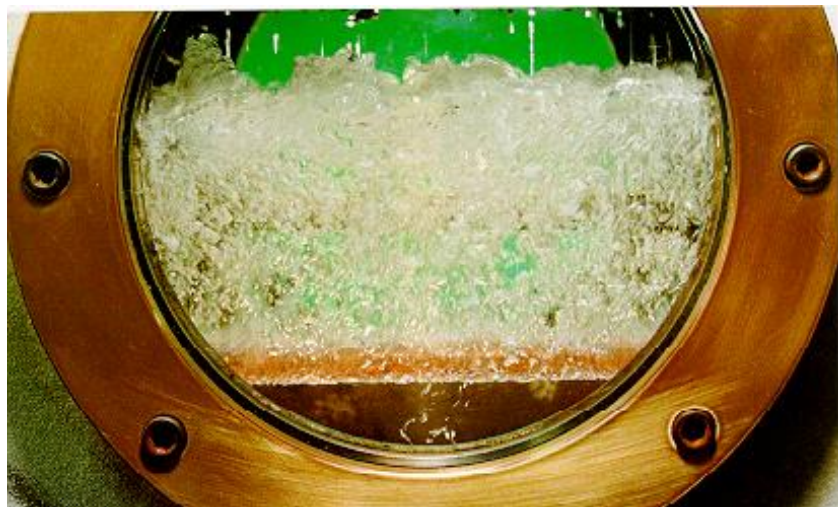
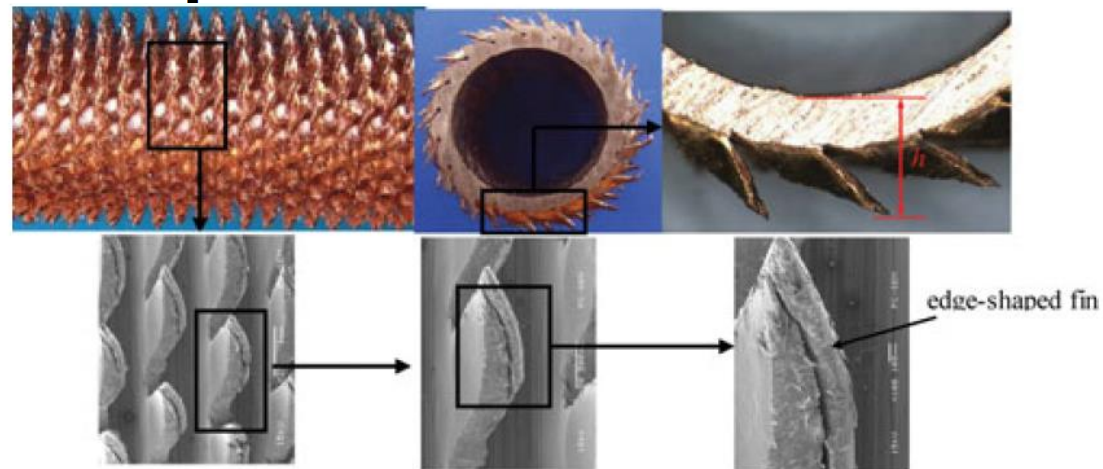


Fig. 3 - Low-finned and double enhanced tubes.

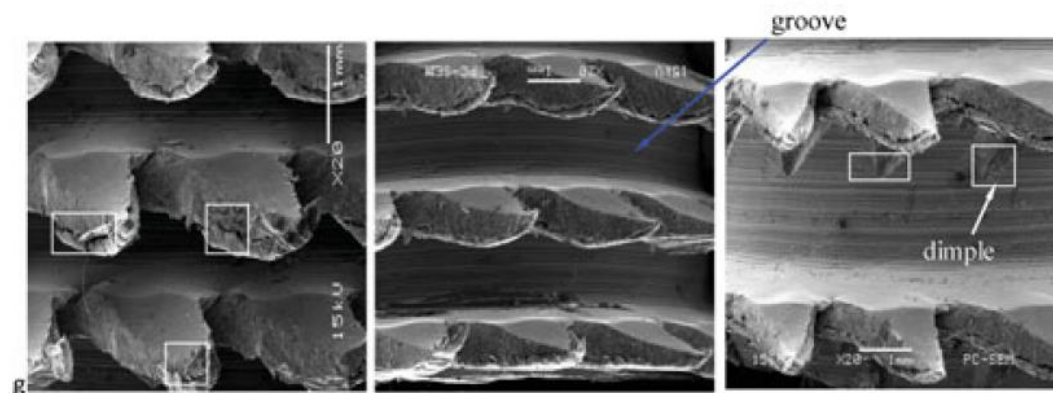


Heat Transfer Performance of an Edge-Shaped Finned Tube

5-7 times increase in
HTC while only 2-3
times increase of HTC
for typical low fin
tube



(a) Three-dimensional edge-shaped fins



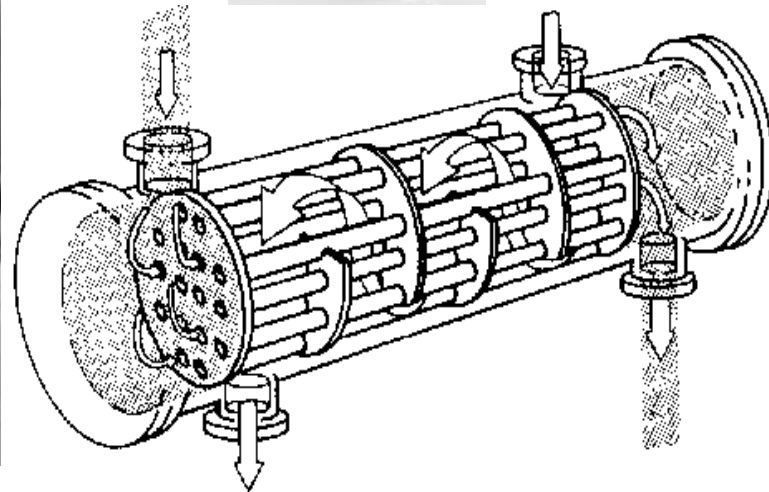
(b) Grooves and dimples

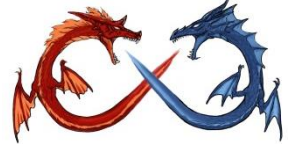
Figure 1 Fins, grooves, and dimples on the tube fabricated by ploughing-extruding process: (a) three-dimensional edge-shaped fins, and (b) grooves and dimples.



Using swirled baffle to improve 殼管式熱交換器 (Shell-and-tube HX)

- Typical Shell and tube HX
 - 殼管式熱交換器不是非常的密集 (non-compact)。
 - 在外型上，殼管式熱交換器的設計非常的「強壯」，因此非常適用於高壓的應用上。
 - 可廣泛地適用於不同的應用場合，例如惡劣的工作環境與特殊的工作流體。
 - 可適用高溫與高壓的應用場合。





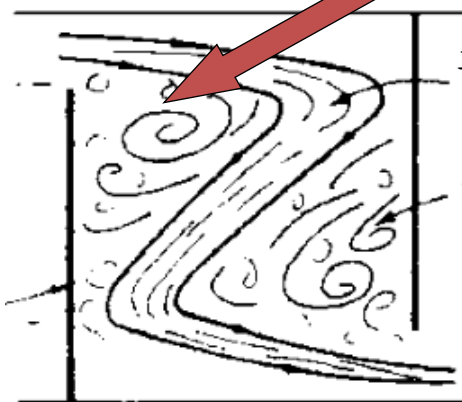
傳統式阻流板流場示意圖



流體以180°轉折方式前進

阻流板

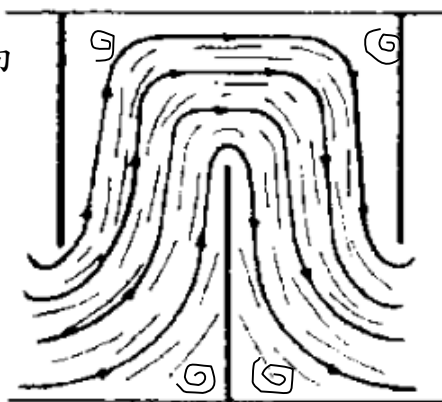
轉折處會出現大量漩渦造成熱傳效率差、積垢與沉積腐蝕



開口率小

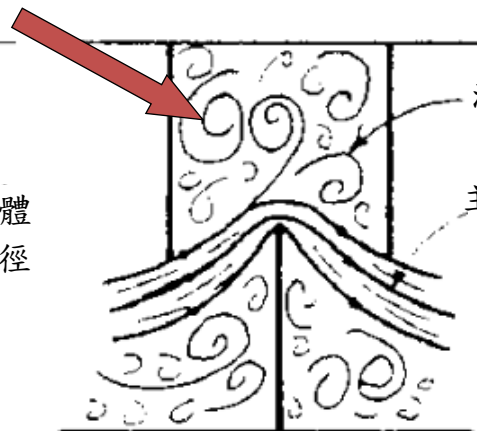
主要流向
(B流)

漩渦



開口率適中

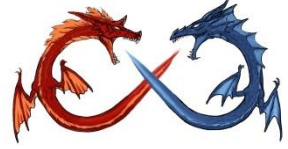
殼體
內徑



開口率大

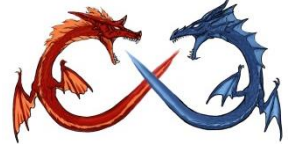
漩渦

主要流向
(B流)



傳統殼管式的缺點

- 殼側高壓損 (due to the sudden contraction and expansion of flow, and fluid impinging on the shell wall caused by segmental baffles) ，需要較大的Pump與操作功耗
- 較低的熱傳性能 (due to the flow stagnation in the so-called “stagnation regions,”)
- 較低的殼側流速 (due to the leakage between baffles and shell wall caused by inaccuracy in manufacturing tolerance and installation)
- 管群的流動震盪 (due to the vibration caused by shell-side flow normal to tube bundle)



Enhancement via directing..

HELIXCHANGER® 的優點與效益

降低殼側積垢速率與
增加操作時間3~4倍

平均可節省
整體成本
20%

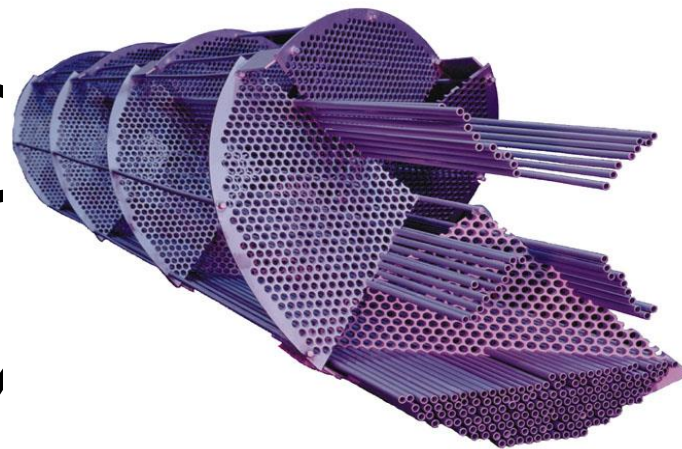
減少所需
熱傳面積最
高35%

製程流量負
荷提升最高
40%

可降低殼側
壓降值，減少
PUMP系統成本

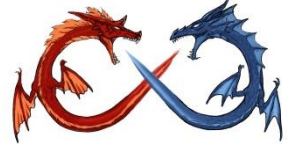
可避免傳熱管
共振損壞

減少設備
停車保養
時間



■ **Lummus**獨家專利技術

■ 專利名：**HELIXCHANGER®**



(19) **United States**

(12) **Patent Application Publication**
Wang et al.

(10) **Pub. No.: US 2008/0190593 A1**

(43) **Pub. Date: Aug. 14, 2008**

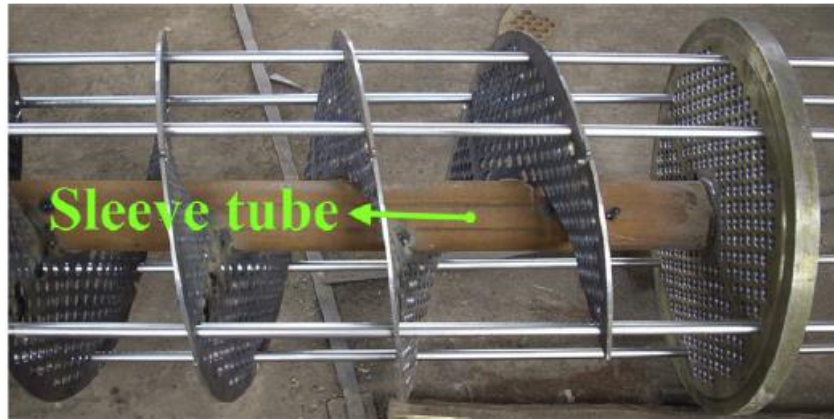


Fig. 1. Central sleeve tube of a continuous helical baffled STHX [30].

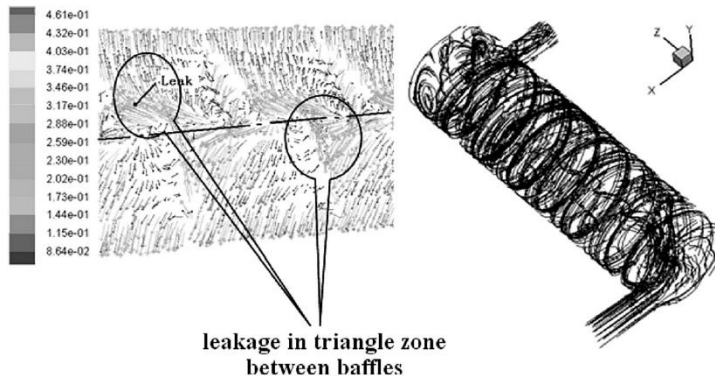
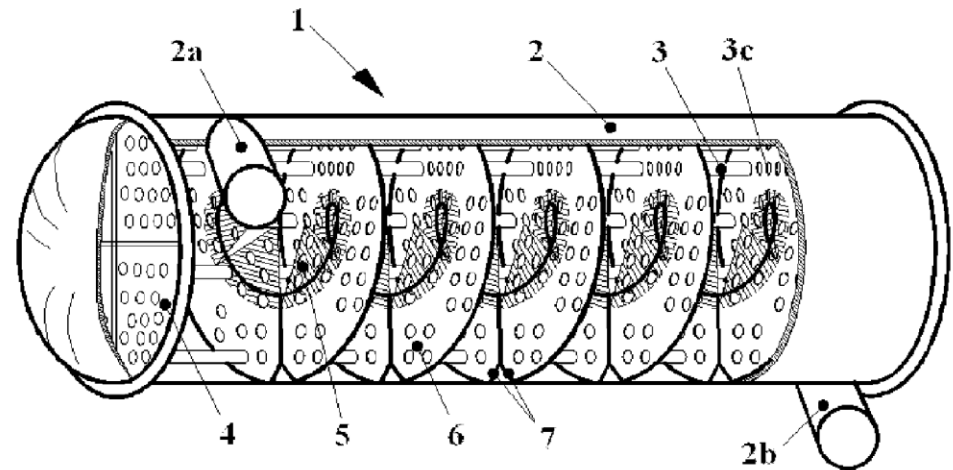


Figure 8 Flow field in symmetrical interface of heat exchangers with overlap helical baffles [32].

Continuous Helix baffle



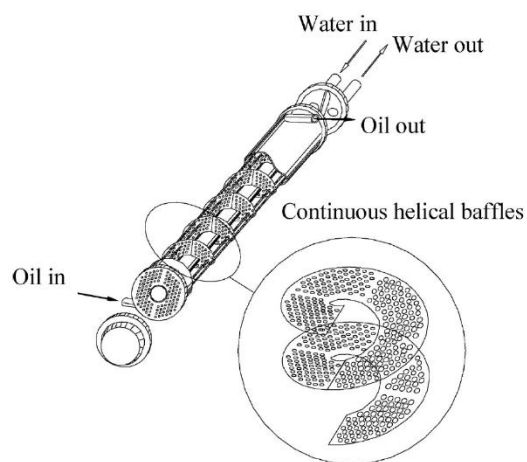


Figure 10 Heat exchanger with continuous helical baffles [44].

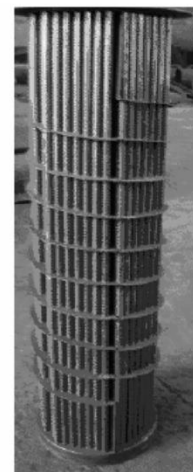
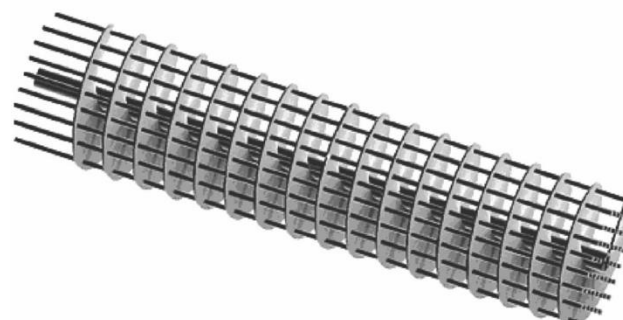
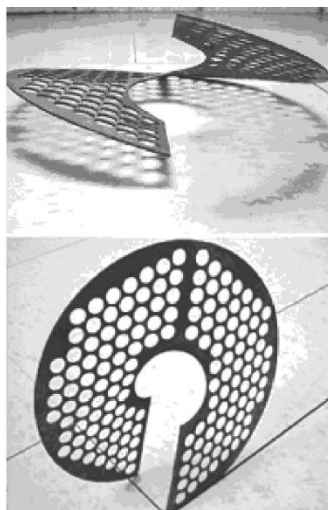


Figure 11 Tube bundle of STHX with continuous helical baffles [45].

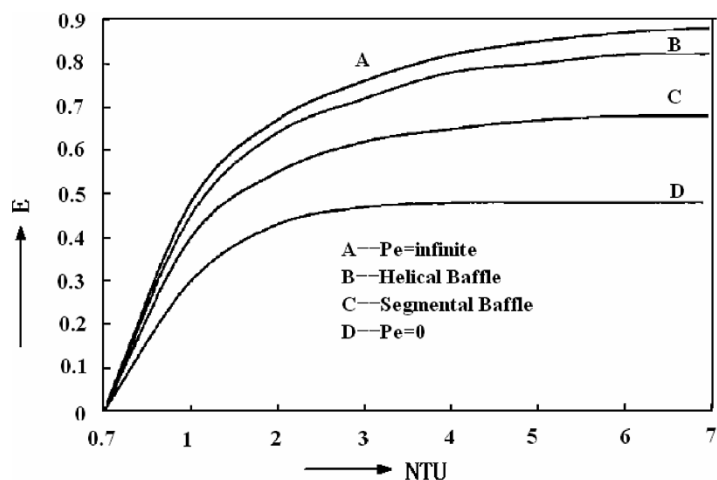


Figure 4 Heat exchanger effectiveness for segmental baffles and helical baffles [17].

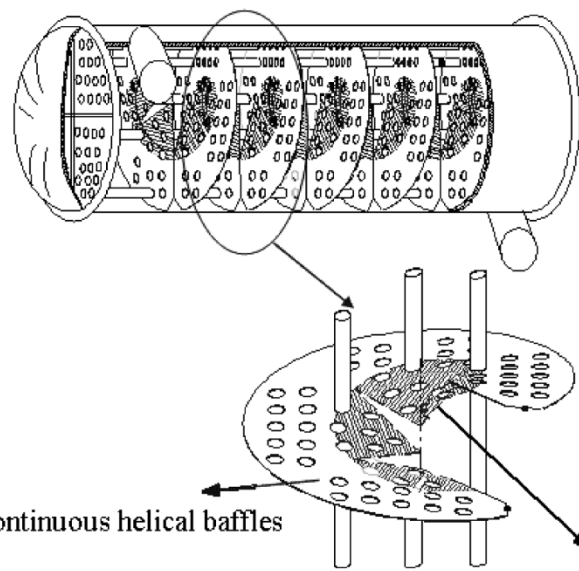


Figure 16 Schematic of combined single shell-pass heat exchangers with helical baffles [54].



Table 6 Main characteristics of shell-and-tube heat exchangers for preheating crude oil [5, 17]

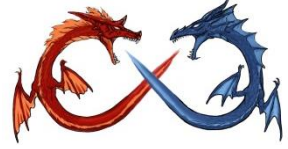
Characteristic	Shell-and-tube heat exchangers with			
	Segmental baffles		Helical baffles (40°)	
	Tube side	Shell side	Tube side	Shell side
Process fluid	Kerosene	Crude oil	Kerosene	Crude oil
Mass flow rate (kg/h)	2.8	25	2.8	25
Inlet temperature (°C)	200.7	10	200.7	10
Outlet temperature (°C)	46.5	32.1	46.5	32.1
Dynamic viscosity (Pa-s)	0.61011e-3	6.96288 e-3	0.6066e-3	6.9841 e-3
Shell diameter (mm)	491		491	
Tubes				
OD (mm)	20		20	
Length (mm)	3,985		4,171	
Number	220		220	
Number of passes	4	1	4	1
Heat duty (kW)	1,015.8		1,006.4	
Area (m ²)	55.08		57.66	
HTC (W/m ² -K)	448.21	997.15	452.02	729.7
Pressure drop (kPa)	2.52	43.55	2.6	17
Capital cost				
\$	45,522		46,251	
\$/yr	8,116		8,246	
Operation cost (\$/yr)	1,189		470	
Total cost (\$/yr)	9,305		8,716	

Review of Improvements on Shell-and-Tube Heat Exchangers With Helical Baffles

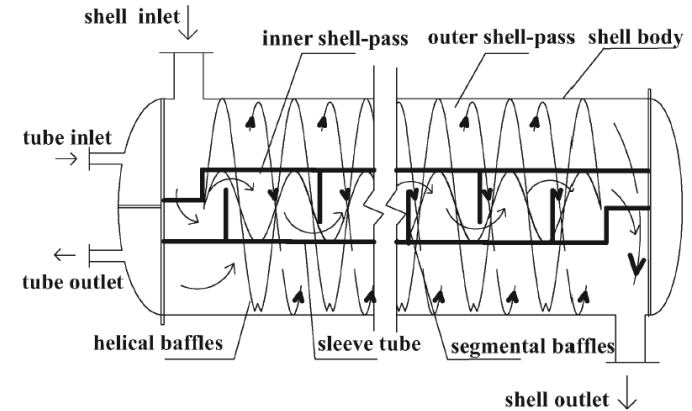
QIUWANG WANG, GUIDONG CHEN, QIUYANG CHEN, and MIN ZENG

Table 2 Compared results of direct expanding evaporators with continuous helical and segmental baffles [47]

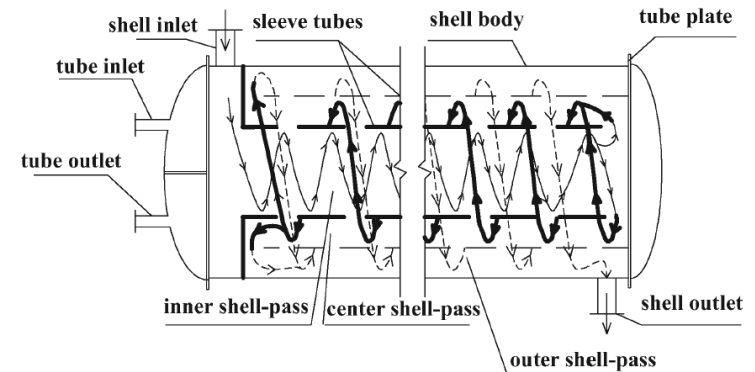
Item	LS195Z	LS195Z II
Compressor type	SRF-3	
Number of condenser tubes	90	90
Total heat transfer area of condenser /m ²	10.52	
Baffle style	Segmental	Continuous helical
Number of evaporator tubes	196	196
Heat transfer area, m ²	23.99	23.91
Refrigeration capacity, kW	185.8	195.0 (+4.9%)
Coefficient of performance (COP)	3.57	3.75 (+5.4%)
Heat flux on the outer surface, kWm ⁻²	7.74	8.23
Total heat transfer coefficient, W m ⁻² K ⁻¹	1,545	1,613 (+4.4%)



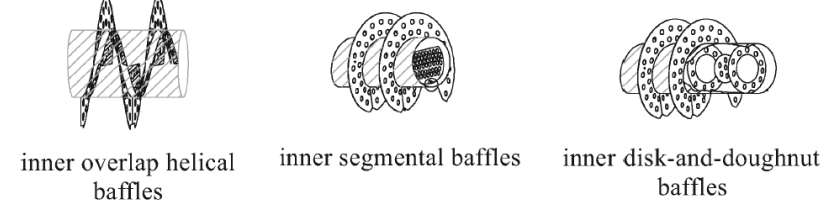
- Continuous helical baffles and combined helical baffles have higher heat transfer coefficients per pressure drop $h/\Delta p$ than the segmental baffled STHXs. STHXs still have higher heat transfer coefficients and much higher pressure drop at the same mass flowrate.
- Continuous helical baffles is superior to discontinuous helical baffle but is more difficult to manufacture (especially when the size is becoming larger). Combined helical baffle is more easier to produce.



(a) combined parallel multiple shell pass helical baffled STHX [58]

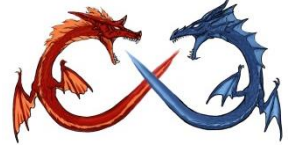


(b) combined serial multiple shell pass STHX with helical baffles [59]



(c) different baffles in inner shell pass [58, 59]

Figure 18 Schematic of multiple shell pass STHXs with helical baffles.



Surface engineering for phase change heat transfer: A review, *MRS Energy & Sustainability*, pp. 1-40, 2014

- High performance (efficiency) design requires low ΔT

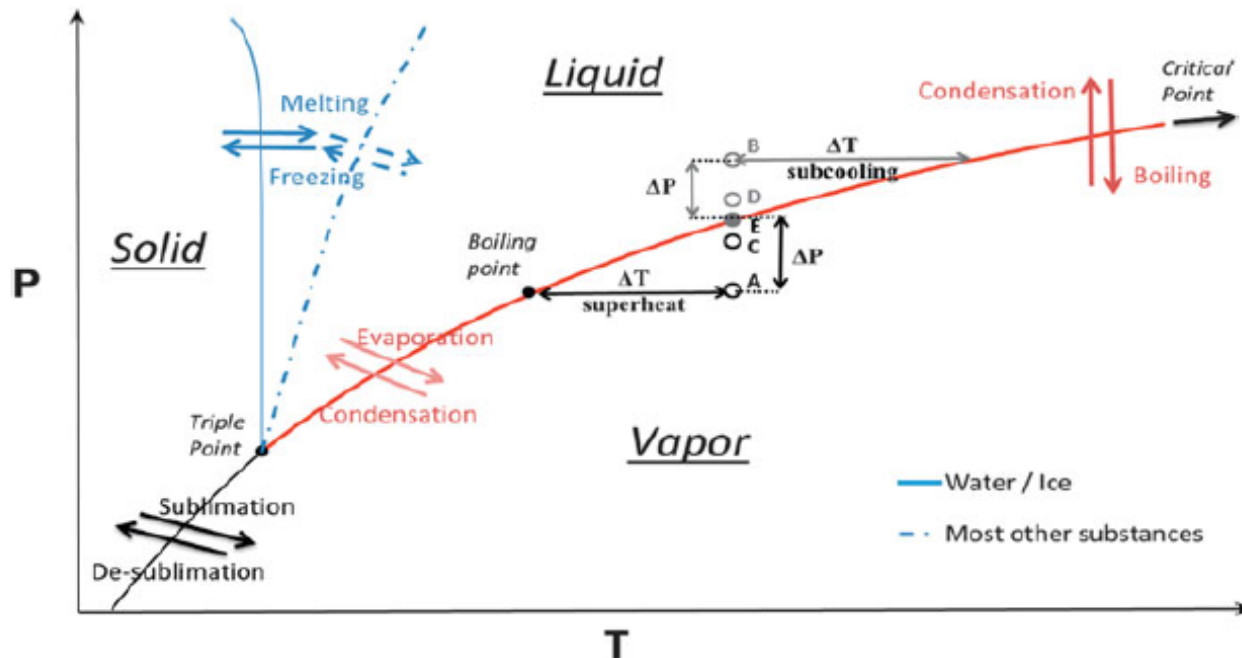


Figure 3. The phase change processes considered in this review can all be described in a pressure (log scale) – temperature diagram. The points A, C, and E illustrate the thermodynamic discussion in boiling, section “Characteristics of optimum surfaces for boiling heat transfer”, whereas the points B, D, and E are relevant to condensation, section “Characteristics of optimum surfaces for condensation heat transfer.”



Table 1. List of the fluid phases for typical phase change heat transfer processes, with the mechanisms and forces involved.

	Boiling	Condensation	Desublimation	Freezing
Continuous phase	Liquid	Vapor	Vapor	Liquid
Discrete phase	Vapor	Liquid	Solid	Solid
Wettability that promotes nucleation	Hydrophobic	Hydrophilic	Hydrophilic	Debated
Wettability that delays transition from high to low thermodynamic efficiency	Hydrophilic	Hydrophobic	Hydrophobic	Debated
Forces that promote discrete phase detachment from substrate	Buoyancy, shear, inertia	Gravity, surface tension, shear	Shear	Inertia, external stress
Forces that promote discrete phase attachment to substrate	Surface tension	Surface tension, gravity, inertia	Gravity, surface tension, adhesion	Adhesion
Dominant heat transfer mechanism at low $q/\Delta T$	Single phase convection	Single phase convection	Single phase convection	Single-phase (liquid) convection
Dominant heat transfer mechanism at high q , below CHF	Multiphase convection	Multiphase convection	Convection (primarily) and conduction	Conduction, convection
Dominant mechanism for q above CHF	Radiation	Single phase convection	Conduction	Conduction

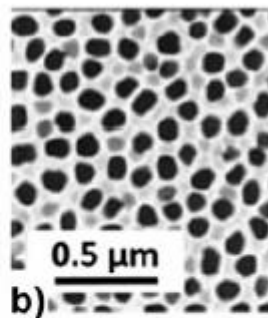
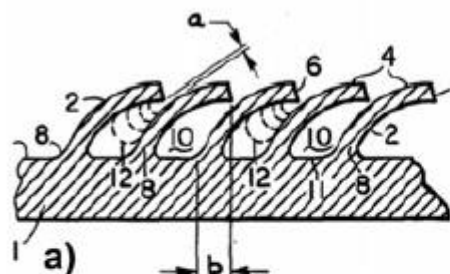


Figure 8. Examples of surface machining and roughening: (a) machined surface with bent fins to enhance nucleate boiling, from patent¹¹⁸; gap *a* is in the 25–125 μm range; (b) electrochemically processed porous alumina (AnodiscTM).

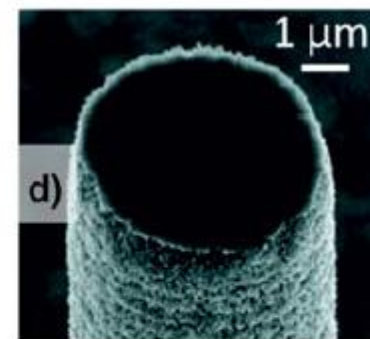
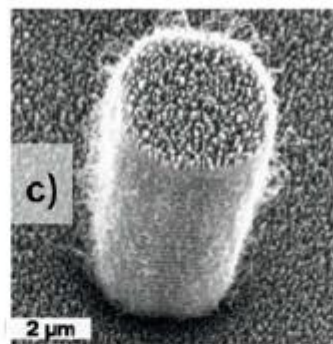
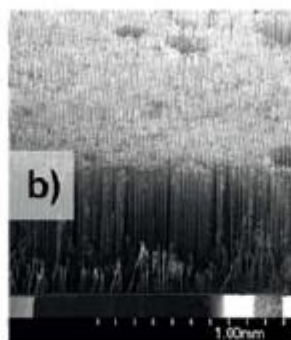
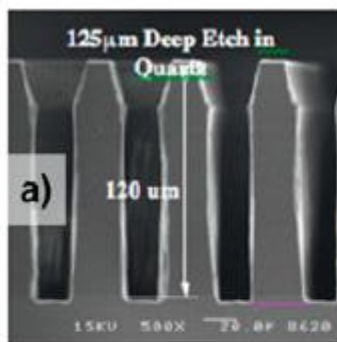


Figure 10. Examples of photolithographic processes to create high aspect ratio geometries and multiscale features on solid surfaces. (a) Cross-sectional SEM picture of glass surface textures fabricated by photolithography and deep reactive ion etching on a glass (quartz) substrate. (b) SEM picture of a dense array of 600 μm -tall freestanding Ni posts (diameter = 5 μm ; pitch = 14 μm ; height-to-diameter aspect ratio = 120) fabricated by electroplating into deep pores formed by photoelectrochemical etching on silicon wafer. Copyright 2011, IEEE. Reprinted with permission from Ref. 128. (c, d) Microscale posts covered with nanoscale features. Figures are reprinted from Refs. 15 and 129 with the permissions of AIP Publishing LLC (Copyright 2007) and the American Chemical Society (copyright 2011), respectively.

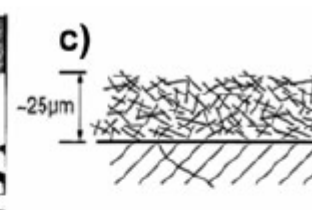
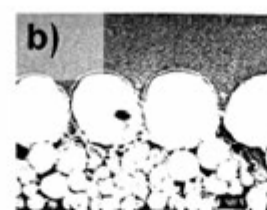
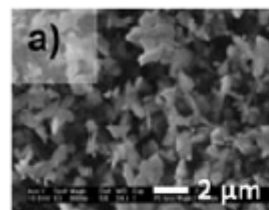


Figure 9. Examples of surface coatings: (a) SEM image of an electrospray boron nitride coating, reprinted from Ref. 122, copyright 2011, with permission from Elsevier. (b) Cross-sectional SEM picture of a controlled porous surface fabricated by sintering copper particles of increasing diameters on a copper substrate to study incipient boiling, reprinted from Ref. 123, permission requested from Hemisphere Publishing. The large particles on top are $\sim 250 \mu\text{m}$ in diameter. (c) Schematic of a porous surface obtained by gluing particles for electronics cooling, from US patent.¹²⁴



Table 2. Microfabrication techniques available for phase change heat transfer applications.

Category	Processing type	Fabrication method	Distribution of the patterns	Shapes (and size)	Suitable scale of the application	Material
Roughening or machining	Subtractive	Sandpapering, sandblasting	Random	Random (μm)	Large (mass production)	Copper ¹¹⁷
		Machining	Regular	Controlled (μm to mm)	Medium	Metal ¹¹⁸
		Chemical, electrochemical	Random	Controlled (10–200 nm)	Medium	Alumina (Anodisc™)
Coating	Additive	Sintering	Random	Complex (μm to mm)	Large	Oxide particles ¹²¹ ; Copper particles on copper ¹²³
		Electrospraying	Random	Random and complex	Medium	cBN particles on XC–Co ¹²²
		Gluing	Random	Controlled (1 μm to ~100 μm)	Large	Metal or ceramic particles ¹²⁴
Lithographic fabrication	Additive or subtractive	Photolithography, e-beam lithography	Regular	Controlled (nm to ~100 μm)	Small	Si pillars on Si ¹³³ ; metal pillars on Si ¹²⁸
		Interference lithography	Regular	Controlled (100 nm to ~1 μm)	Medium	Si pillars on Si ¹³⁴ ; Ti, Al and Au pillars on glass ¹³⁵



Figure 12. The Namib desert beetle features a network of hydrophobic and hydrophilic areas on its back, which have been hypothesized to facilitate collection and transport of water in scarce humidity conditions. Reproduced by permission from Macmillan Publishers Ltd,¹⁸⁸ copyright 2001.

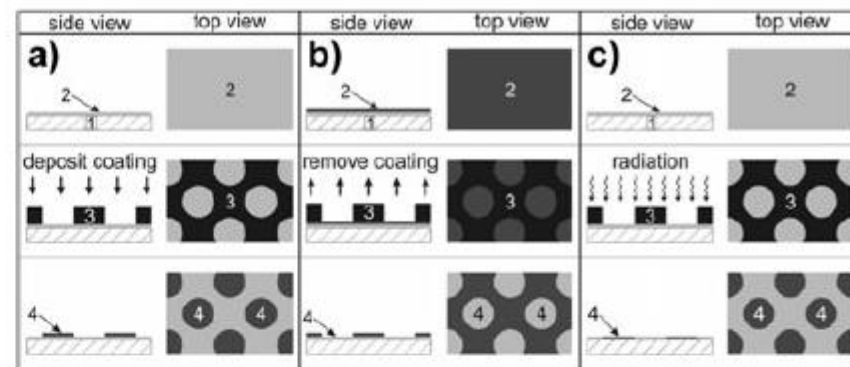


Figure 14. Schematics demonstrating three techniques for achieving biphilic surfaces. List of identification numbers: (1) substrate; (2) coating with homogeneous chemistry; (3) mask (e.g., lithography); (4) coating region with differential chemistry as compared with coating (2). In (a), biphilic surfaces are achieved by depositing (4) onto (2); *additive*. In (b), biphilic surfaces are achieved by removing (2) to reveal (4); *subtractive*. In (c), radiation or other reactive mechanisms are utilized to transform the surface chemistry of (2) into (4); *reactive*.



Fluid Flow and Heat Transfer at Micro- and Meso-Scales With Application to Heat Exchanger Design, Appl. Mech. Rev., **53**, pp. 175–193

Various small and mini heat exchangers in terms of hydraulic diameter D_h :

- micro heat exchanger: $D_h = 1\text{--}100\ \mu\text{m}$
- meso heat exchanger: $D_h = 100\ \mu\text{m}\text{--}1\ \text{mm}$
- compact heat exchanger: $D_h = 1\text{--}6\ \text{mm}$
- conventional heat exchanger: $D_h > 6\ \text{mm}$



There are several important dimensionless numbers, which are used to represent the feature of fluid flow in microscale channels. According to these dimensionless numbers, the distinction between macro- and microscale channels may be classified as well. Triplett et al. [47] defined flow channels with hydraulic diameters D_h of the order of, or smaller than, the Laplace constant L ,

$$L = \sqrt{\frac{\sigma}{g(\rho_L - \rho_G)}} \quad (1)$$

as microscale channels, where σ is the surface tension, g is the gravitational acceleration, and ρ_L and ρ_G are, respectively, liquid and gas/vapor densities.

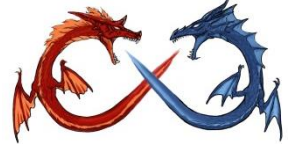
Kew and Cornwell [43] earlier proposed the confinement number Co for the distinction of macro- and microscale channels,

$$Co = \frac{1}{D_h} \sqrt{\frac{4\sigma}{g(\rho_L - \rho_G)}} \quad (2)$$

which is actually based on the definition of the Laplace constant.

Based on a linear stability analysis of stratified flow and the argument that neutral stability should consider a disturbance wavelength of the order of channel diameter, Brauner and Moalem-Maron [48] derived the Eotvös number $Eö$ criterion for the dominance of surface tension for microscale channels,

$$Eö = \frac{(2\pi)^2 \sigma}{(\rho_L - \rho_G) D_h^2 g} > 1 \quad (3)$$



Two-Phase Flow Patterns and Flow-Pattern Maps:

Fundamentals and Applications, Applied Mechanics Reviews

2008, Vol. 61 / 050802-1

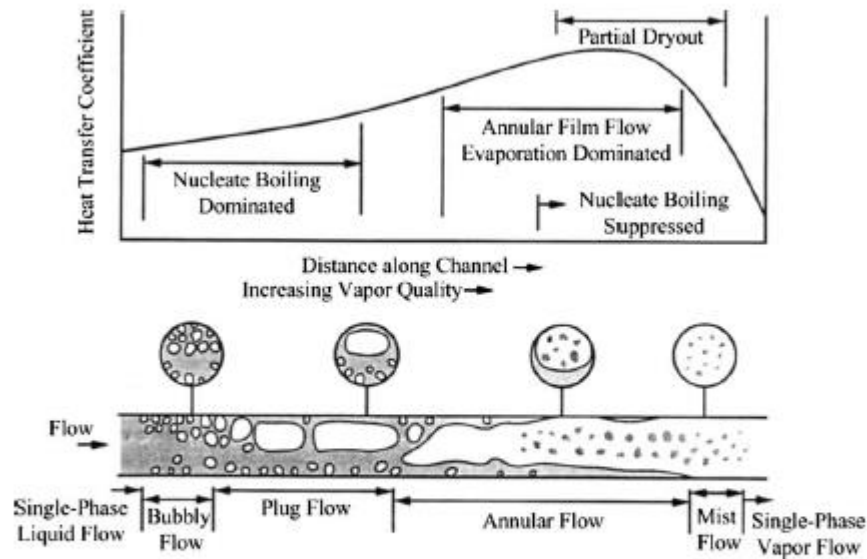


Fig. 4 Schematic of flow patterns and the corresponding heat transfer mechanisms and qualitative variation of the heat transfer coefficients for flow boiling in a horizontal tube [1,2]

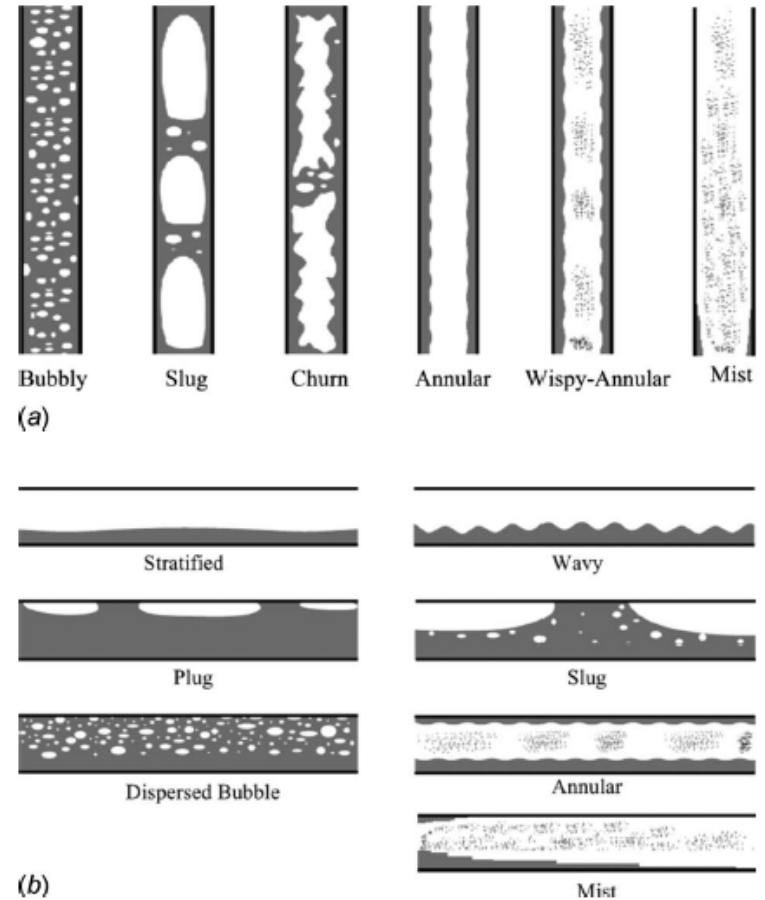
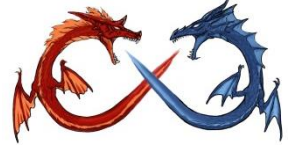


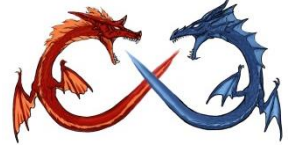
Fig. 2 (a) Schematic of flow patterns in vertical upward gas-liquid cocurrent flow; (b) schematic of flow patterns in horizontal gas-liquid cocurrent flow [1]



微鳍管 Microfin tube



Benefits: increase 100% more heat transfer coefficients with only 10~50% increase of pressure drops



熱傳管幾何尺寸

空調機用熱傳管演進圖

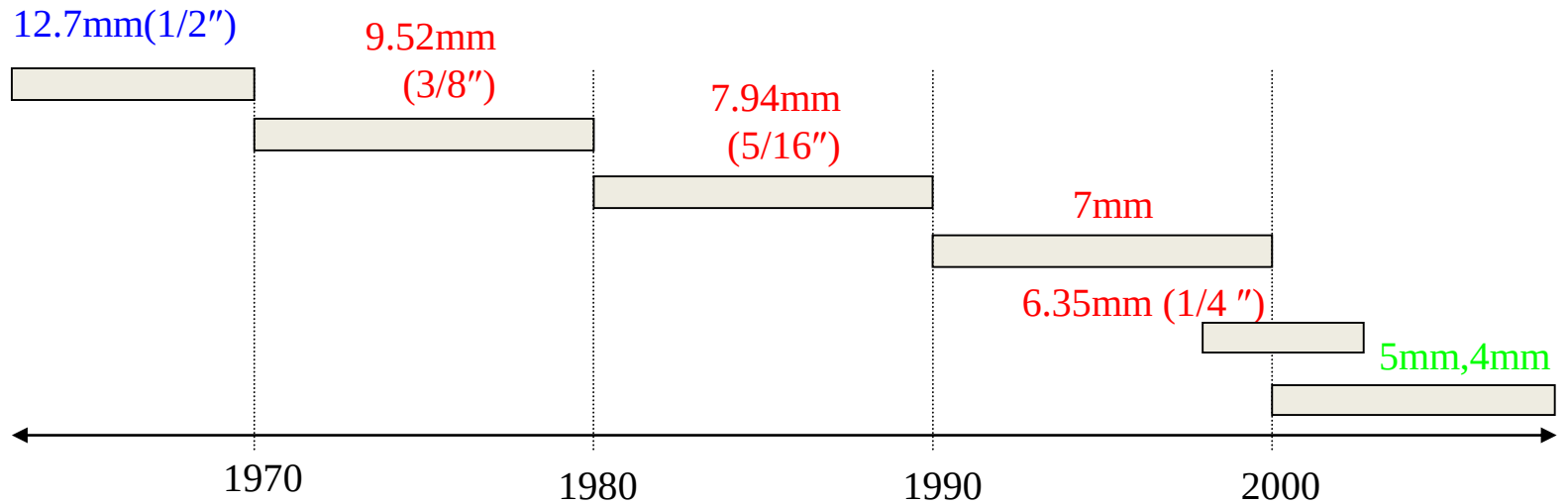


Fig. 各種型式的熱傳增強管

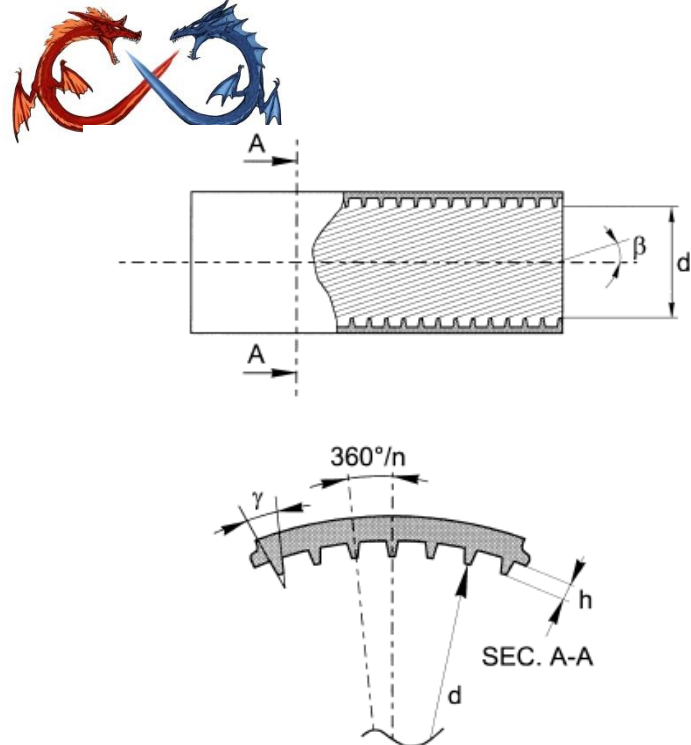


Fig. 1. Characteristic geometrical parameters of inside enhanced tubes.

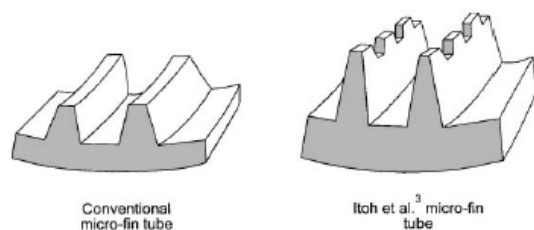


Fig. 3. Comparison between traditional and Itoh et al. [3] micro-fin tube.

Fig. 3. Comparaison entre les tubes à microailettes classiques et d'Itoh et al. [3].



Fig. 4. Shape of Muzzio et al. [4] micro-fin tube.

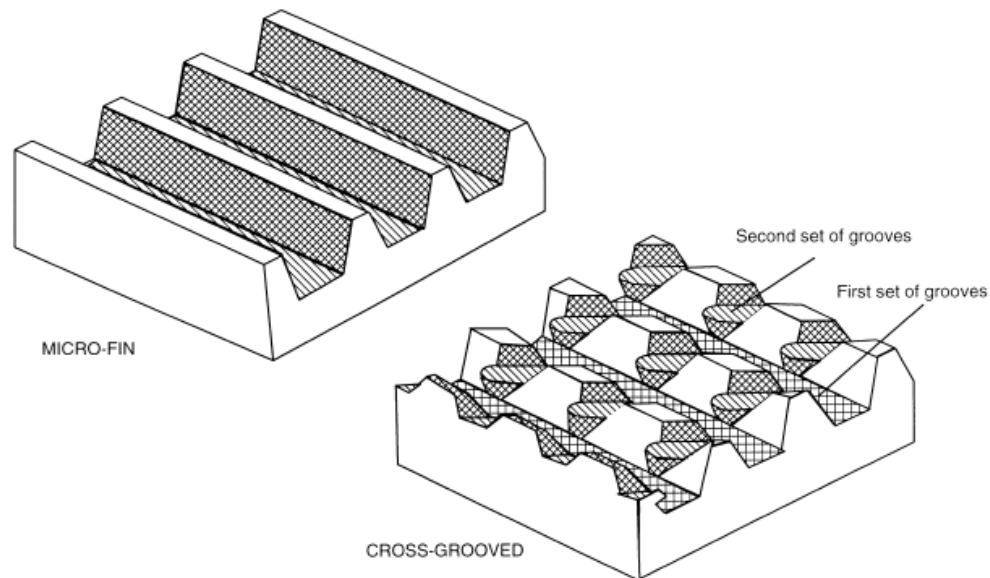


Fig. 2. Typical configuration of micro-fin and cross-grooved surfaces.

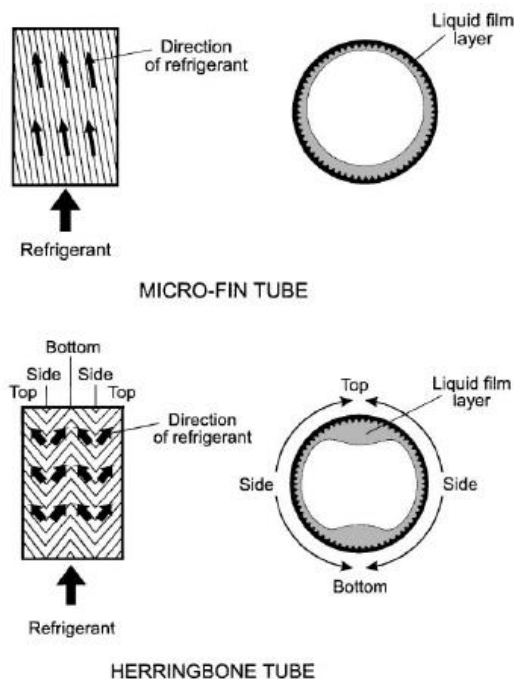
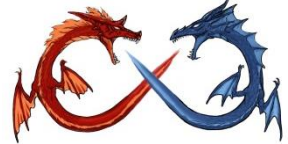


Fig. 5. Comparison between micro-fin and herringbone tubes in annular-type flow pattern.

Heat transfer and pressure drop during condensation of refrigerants inside horizontal enhanced tubes, Int. J. Refrig., 23 (2000) 4-25



Recent trend in studies for Microfin tube..

- Smaller..
- Low GWP refrigerant
- Refrigerant Mixtures

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Experimental investigation of R1234ze(E) flow boiling inside a 2.4 mm ID horizontal microfin tube

Andrea Diani^a, Simone Mancin^b, Alberto Cavallini^a, Luisa Rossetto^{a,*}

Experimental Thermal and Fluid Science 66 (2015) 127–136



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Flow boiling heat transfer of R1234yf inside a 3.4 mm ID microfin tube

A. Diani^a, S. Mancin^b, L. Rossetto^{a,*}

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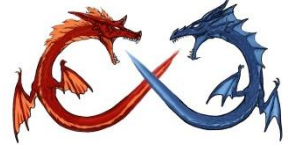
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Flow boiling heat transfer of carbon dioxide inside a small-sized microfin tube

Chaobin Dang*, Nobori Haraguchi, Eiji Hihara



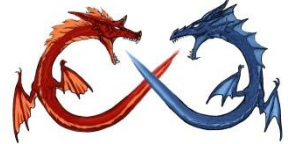
- Recommended correlations applicable for microfin tubes in single phase, condensation, and convective boiling is shown in the review article.

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Heat Transfer Correlations for Single-Phase Flow, Condensation, and Boiling in Microfin Tubes



Horizontal convective boiling of R448A, R449A, and R452B within a micro-fin tube

- R410A (GWP=1725) and R404A (GWP=3300)
- Three new refrigerants, R448A (GWP=1274), R449A (GWP=1283), and R452B (GWP=676) , are potential low GWP replacements for R404A (R125/143a/134a [44/52/4]).
- R452B is also a proposed replacement for R410A (R32/125 [50/50])
- R452B (R32/125/1234yf [67/7/26]) are essential for the evaluation of their use for unitary application.



Table 2
Characteristics of refrigerants selected [20].

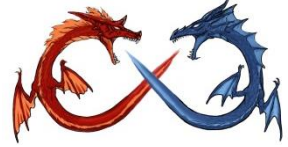
	R404A	R448A	Rel. dev.
ODP/GWP _{100-yr}	0/3943	0/1390	
ASHRAE Safety Class	A1	A1	
Glide ^a (K)	0.75	6.27	
Critical Pressure (kPa)	345.20	356.81	3%
Critical Temperature (K)	3728.85	4674.93	25%
NBP (K)	227.41	233.05	2%
Liquid density ^a (kg m ⁻³)	1150.59	1192.39	4%
Vapour density ^a (kg m ⁻³)	30.32	22.09	-27%
Liquid c_p ^a (kJ kg ⁻¹ K ⁻¹)	1.39	1.42	2%
Vapour c_p ^a (kJ kg ⁻¹ K ⁻¹)	1.00	0.98	-2%
Liquid therm. cond. ^a (W m ⁻¹ K)	$73.15 \cdot 10^{-3}$	$92.41 \cdot 10^{-3}$	26%
Vapour therm. cond. ^a (W m ⁻¹ K)	$12.82 \cdot 10^{-3}$	$12.01 \cdot 10^{-3}$	-6%
Liquid viscosity ^a (Pa s ⁻¹)	$179.70 \cdot 10^{-6}$	$188.35 \cdot 10^{-6}$	5%
Vapour viscosity ^a (Pa s ⁻¹)	$11.00 \cdot 10^{-6}$	$11.42 \cdot 10^{-6}$	4%

^a Temperature = 273 K.

ASHRAE Number	R-452B
Composition	HFO/HFC blend
Boiling Point	-60°F (-51°C)
Critical Temperature	169°F (76°C)
Ozone Depletion Potential	0
AR5 Global Warming Potential	676
ASHRAE Safety Classification	A2L
Temperature Glide	~1K

Table 3. Representative properties from REFPROP (Lemmon et al. 2013).

Refrigerant	Evaluated at average test conditions for each fluid			Evaluated at $T_s = 277.6$ K			
	$T_d - T_b$ (K)	k_l (W m ⁻¹ K ⁻¹)	Pr	σ (mN m ⁻¹)	ρ_l (kg m ⁻³)	ρ_v (kg m ⁻³)	$[P_s]_{xq=0}$ (kPa)
R404A	0.50	0.074	3.5	7.05	1132.8	35.8	700.6
R410A	0.11	0.100	2.3	8.38	1151.9	34.6	920.5
R448A	5.74	0.088	2.9	9.73	1180.7	28.8	717.1
R449A	5.39	0.090	2.9	9.53	1180.2	26.0	708.2
R452B	1.24	0.110	2.1	10.1	1072.5	28.1	889.7



- The heat transfer coefficient for R449A was approximately 8% larger than that for R448A, while the heat transfer coefficient for R452B was more than 59% larger than either R448A or R449A.
- The heat transfer coefficients for R448A and R449A were roughly between 26 and 48% less than that of R404A for the example case.
- In contrast, the model predicts that the R452B heat transfer coefficient was approximately 13% larger than that of R404A for the same conditions.