Optimized Thermal
SYSTEMS

Optimize to Exceed

Cu Copper Alliance™

Advantages of Small Diameter Copper Tube-Fin Heat Exchangers

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Who are we?

- Pre-competitive
- Publishable R&D
- Shared by 30+ member companies
- Conducted by MS, PhD Students
- *General purpose software*

Exclusive License

- Proprietary R&D
- Confidential
- Contracts with specific clients
- Conducted by full-time professional staff
- *Customized Software*

- **Mission:** *Defend and grow markets for copper based on its superior technical performance and its contribution to a higher quality of life worldwide.*
- **Members:** Copper mining and copper fabricating companies worldwide
- **Main activities:**
    - Partner with governments, regulators, and NGOs to implement sustainable development initiatives
    - Conduct and disseminate scientific studies related to copper in health and the environment
    - Develop and transfer new technologies supporting copper applications

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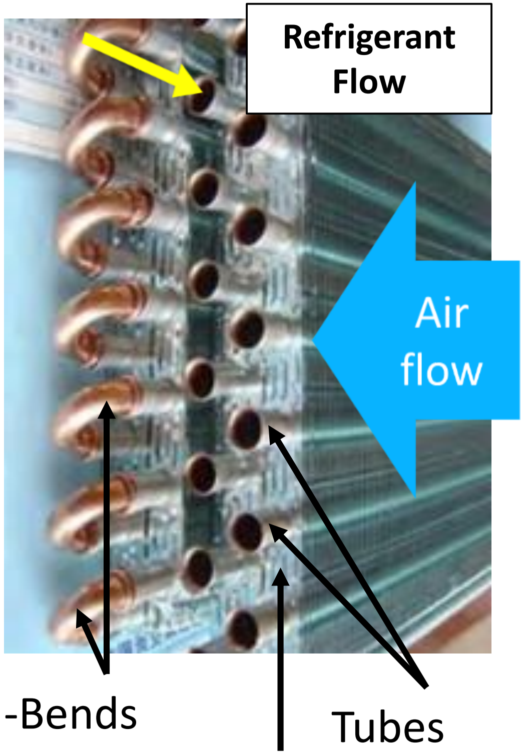
Introduction

Background

Tube-Fin Heat Exchanger

Parts and Working fluids



Refrigerant Flow

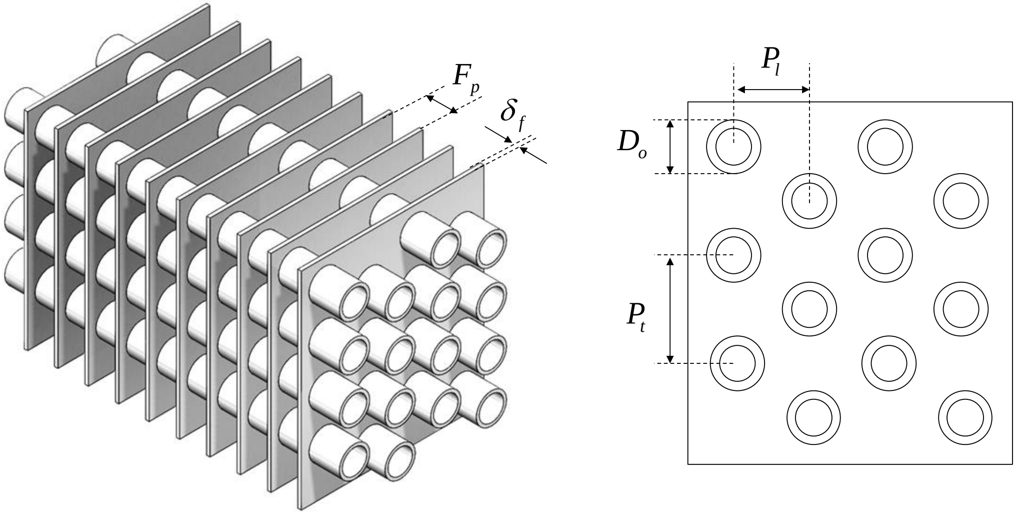
Air flow

U-Bends

Tubes

Fins (Extended Surfaces)

Nomenclature



D_o – Outer diameter

D_i – Inner Diameter

δ_w – Wall thickness

δ_f – Fin thickness

P_t – Transverse pitch (vertical spacing)

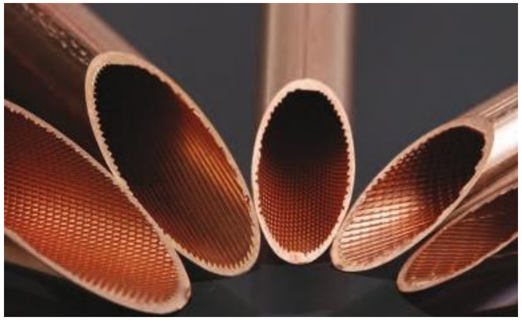
P_l – Longitudinal pitch (horizontal spacing)

F_p – Fin pitch

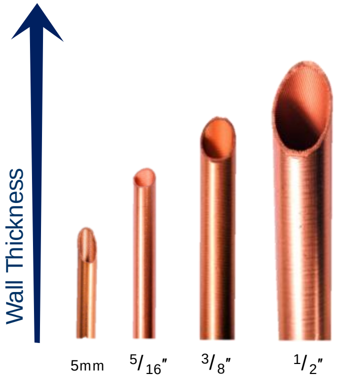
FPI – Fins Per Inch

Copper Tubes

- High thermal conductivity (~380W/m.K) → low wall thermal resistance



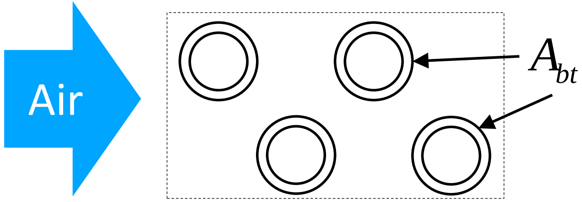
- Corrosion resistance
- Biofouling resistance
- Antimicrobial properties → reduce material build up, potentially fouling
- Soft metal; Pliable → ease of inner grooving
- Small diameter – thinner walls:
  - Lower thermal resistance
  - Withstands higher pressures with thinner walls (e.g. CO2)



Courtesy of MicroGroove^{TM}

Fins & Enhancements

Finless (Bare) Tubes - *bt*



$$\dot{Q}_{bt} = U \cdot A_{bt} \cdot \Delta T_{ml}$$

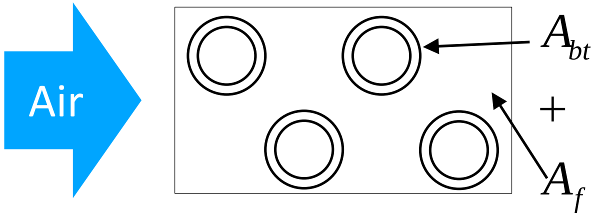
Heat Load

Heat Transfer Coefficient

Surface Area

Mean Log. Temp. Diff.

Finned Tubes - *ft*

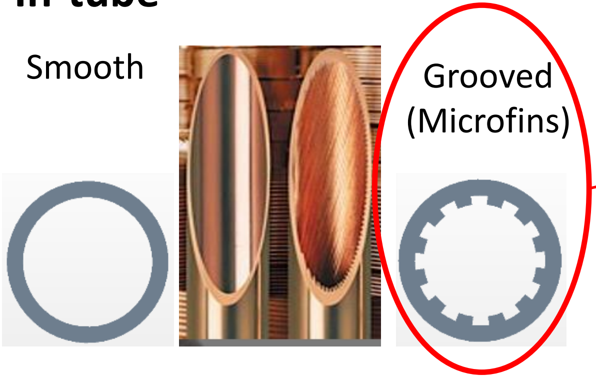


$$\dot{Q}_{ft} = U \cdot (A_{bt} + A_f) \cdot \Delta T_{ml}$$

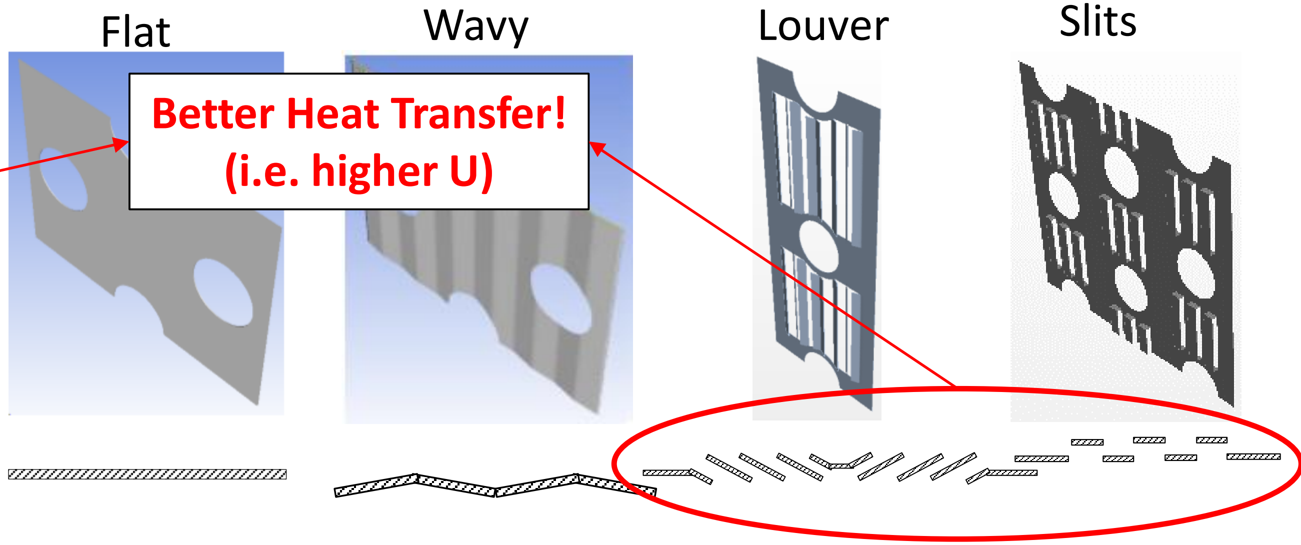
$$\dot{Q}_{ft} > \dot{Q}_{bt}$$

More compact!

In-tube



External Fins

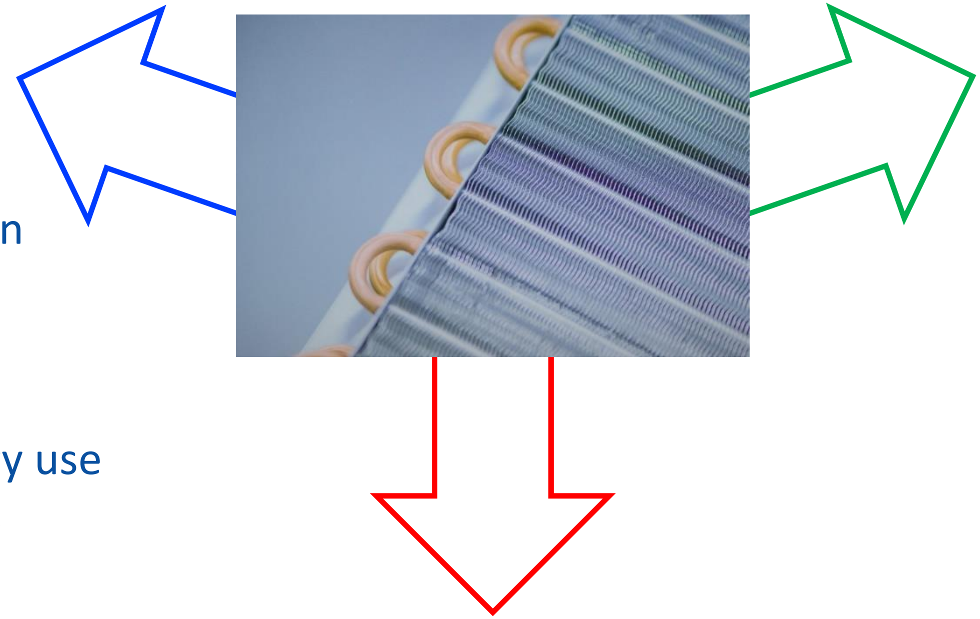


Better Heat Transfer!
(i.e. higher U)

Introduction

Motivation

Motivation - What drives heat exchanger design?



Energy Efficiency

- Energy consumed in buildings
    - COP
    - Billing Cost
    - Primary energy use
    - CO_2 emissions
- Partial load

Environment and Safety

- Direct refrigerant emissions
- Footprints (e.g. CO_2, end-of-life equipment)
- Material (resources)

Cost

- Material
- Tooling
- Size / Weight

Air-to-Fluid Heat Exchangers (HX)

- Worldwide push for more efficient HXs
    - Enhanced fins and tubes (internally) → limited
    - Size increase → Increase surface area while also increasing volume

TIME

- Larger → shipping and space issues, more expensive
- Heavier → shipping
- More refrigerant charge → weight, GWP
- Partial load operations → oversized

Objectives

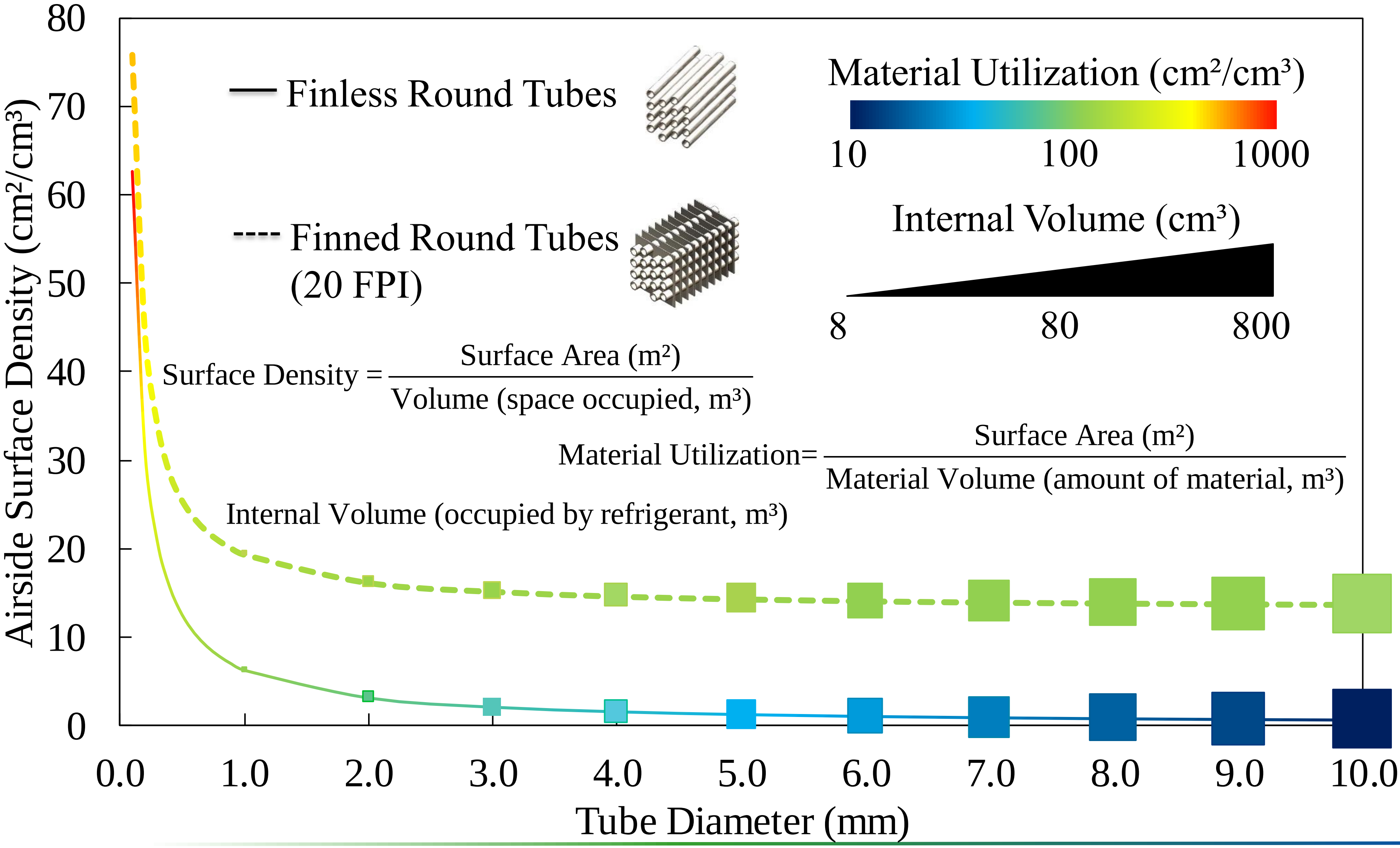
1. Small diameter tubes fundamentals
    - Performance → higher effectiveness, COPs
    - Overall size → weight, cost, space
    - Charge → environmental impact, weight
2. Heat exchanger design Challenges and Considerations
3. Accessible and inexpensive design tools
4. Present example cases where the three above objectives were successfully applied

70% material cost reduction, 50%+ charge reduction, 60% pressure drop reduction

Fundamentals

First Order Analysis

First Order Analysis

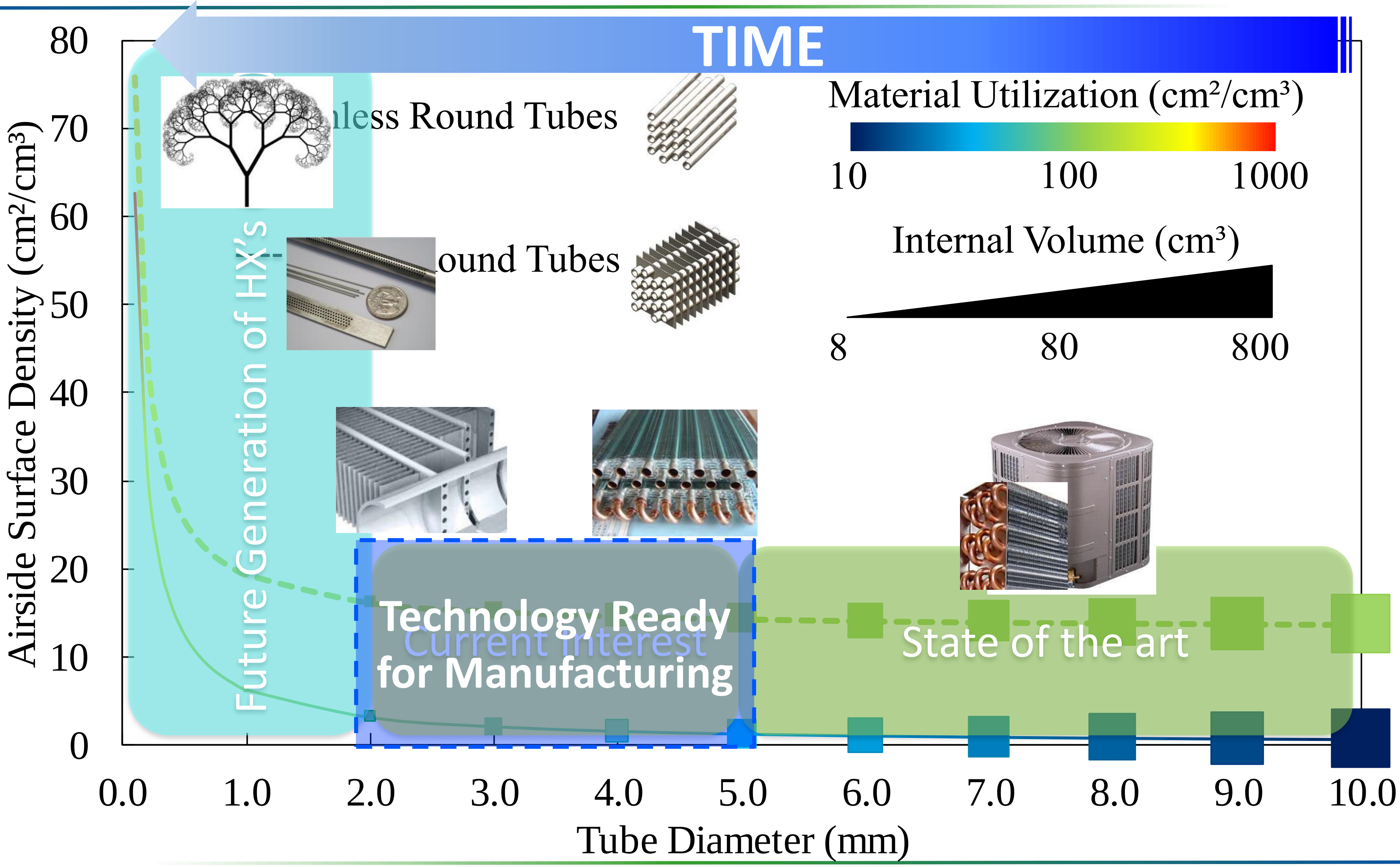


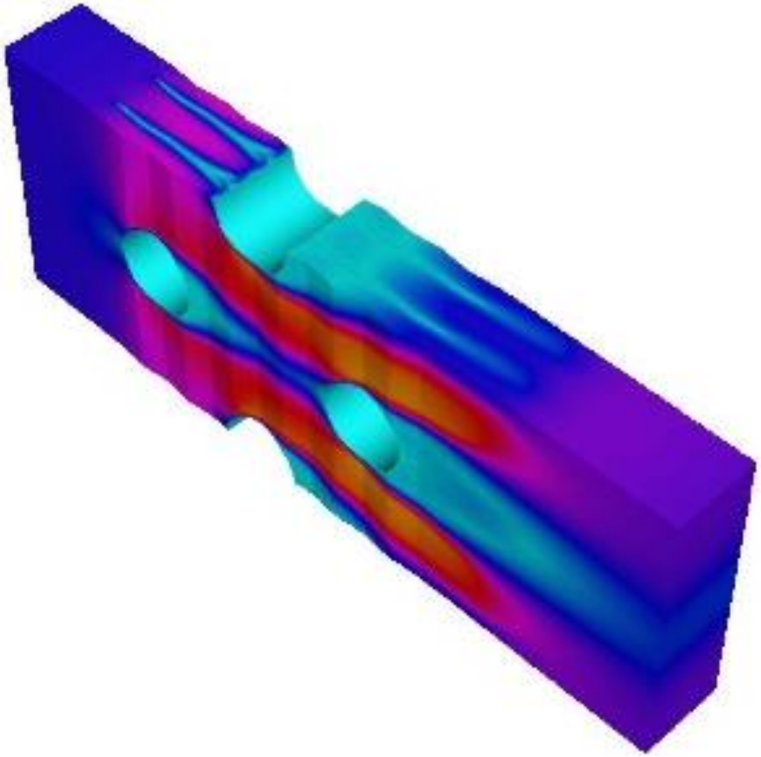
$$\text{Surface Density} = \frac{\text{Surface Area (m}^2\text{)}}{\text{Volume (space occupied, m}^3\text{)}}$$

$$\text{Material Utilization} = \frac{\text{Surface Area (m}^2\text{)}}{\text{Material Volume (amount of material, m}^3\text{)}}$$

Internal Volume (occupied by refrigerant, m³)

First Order Analysis (cont'd)





Fundamentals

Air flow thermal-hydraulic performance

Air Heat Transfer Coefficient (h) vs. D_h

$$Nu \propto \mathrm{Re}^m \, \mathrm{Pr}^n$$

$$h \propto D_h^{m-1} \cdot u^m$$

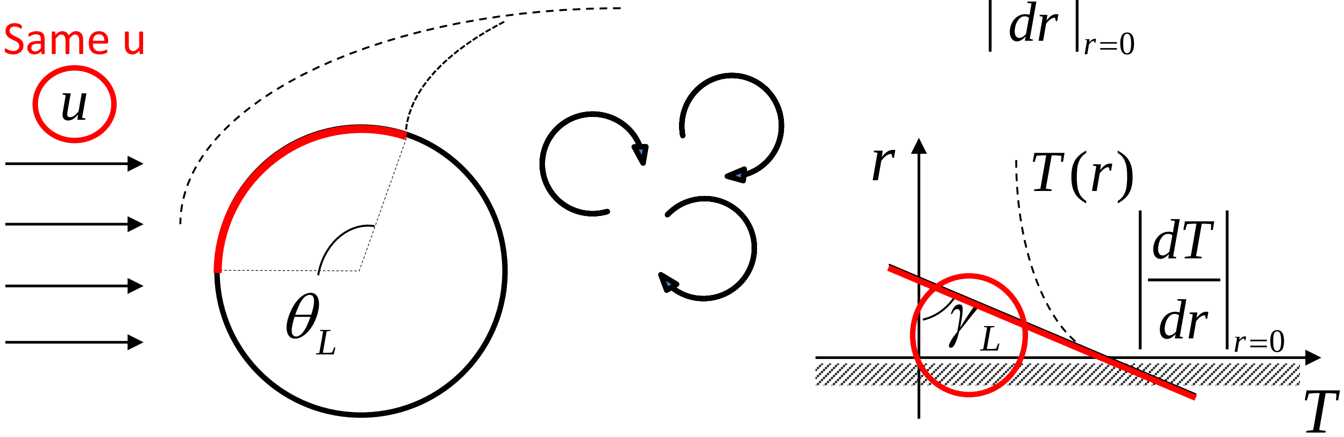
$$0 < m < 1$$

$$\boxed{D_h \downarrow \Rightarrow h \uparrow}$$

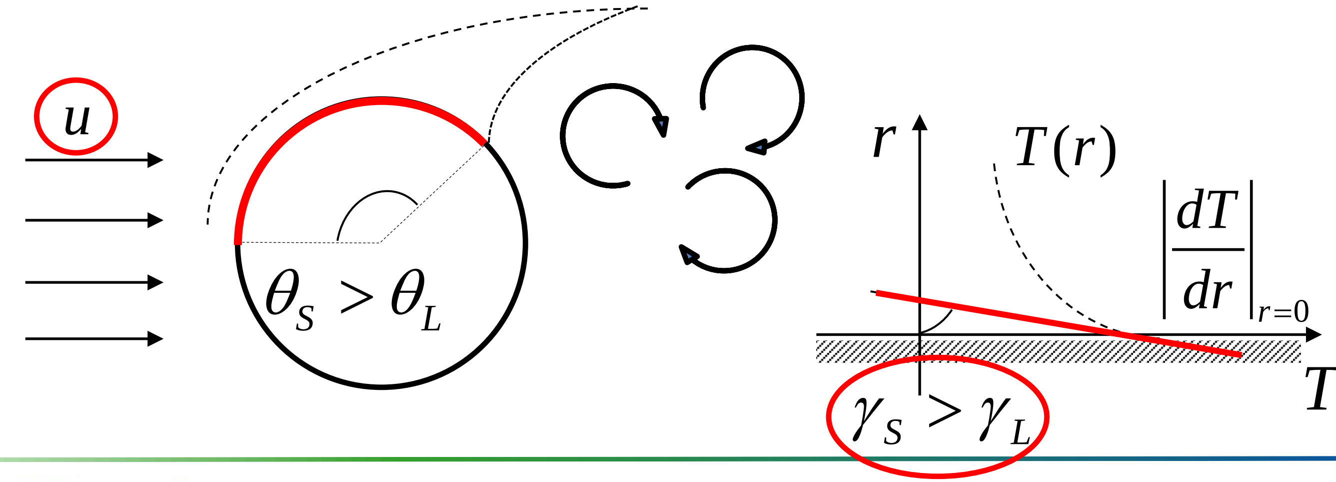
$$u \uparrow \Rightarrow h \uparrow$$

a) Large Diameter Tube

$$h \propto \left| \frac{dT}{dr} \right|_{r=0}$$



b) Small Diameter Tube



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Air Friction Factor (f) vs. D_h

$$f \propto \mathrm{Re}^p$$

$$f \propto D_h^p \cdot u^p$$

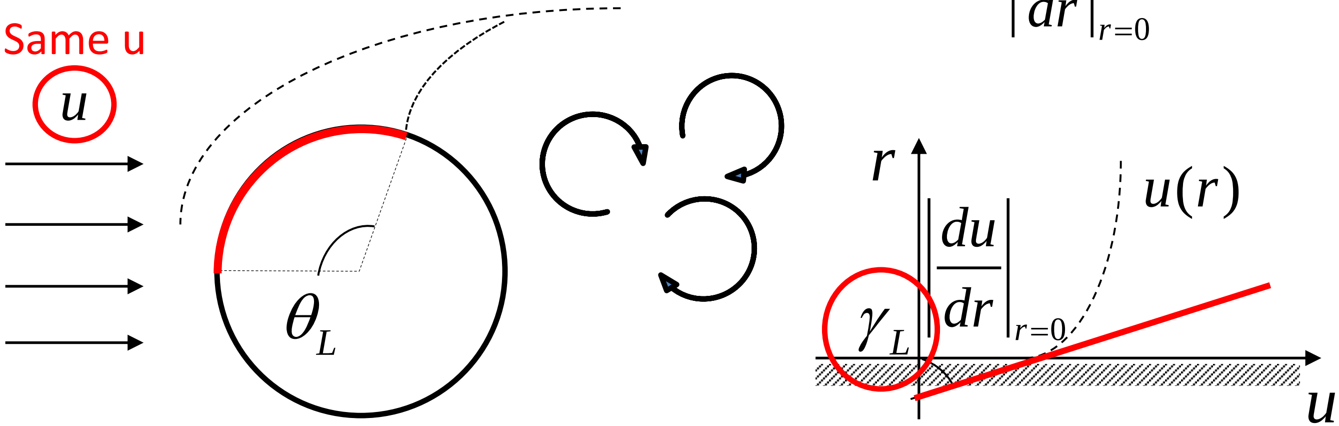
$$p < 0$$

$$\boxed{D_h \downarrow \Rightarrow f \uparrow}$$

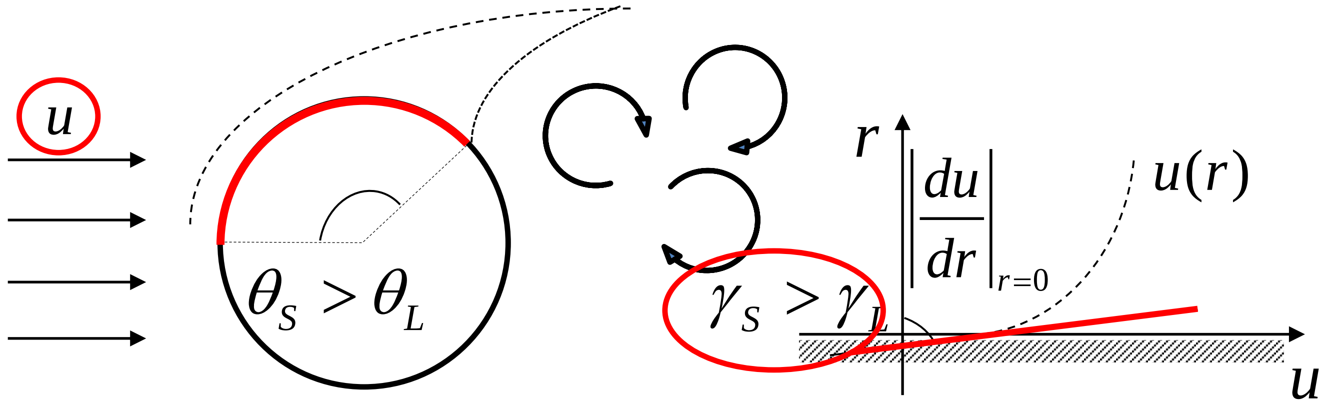
$$u \uparrow \Rightarrow f \downarrow \rightarrow \Delta P \uparrow \propto u^2 \uparrow$$

a) Large Diameter Tube

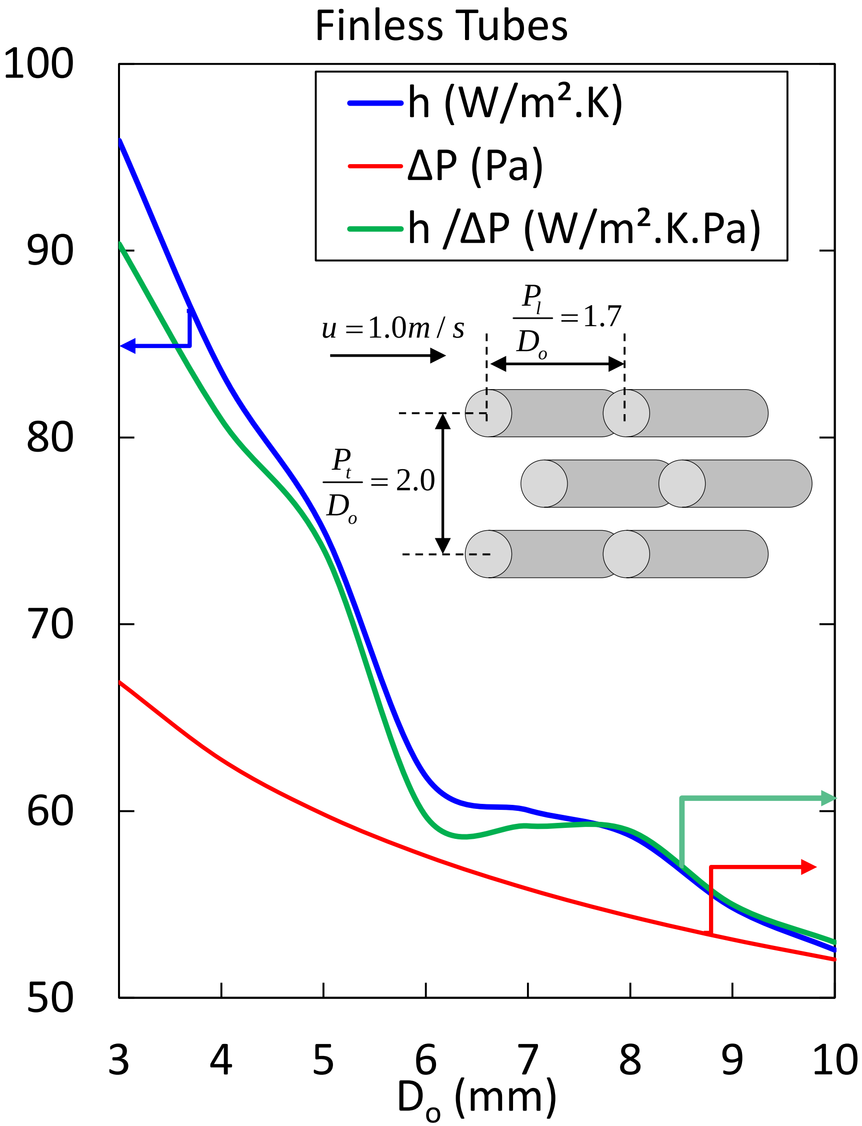
$$f \propto \left|\frac{du}{dr}\right|_{r=0}$$

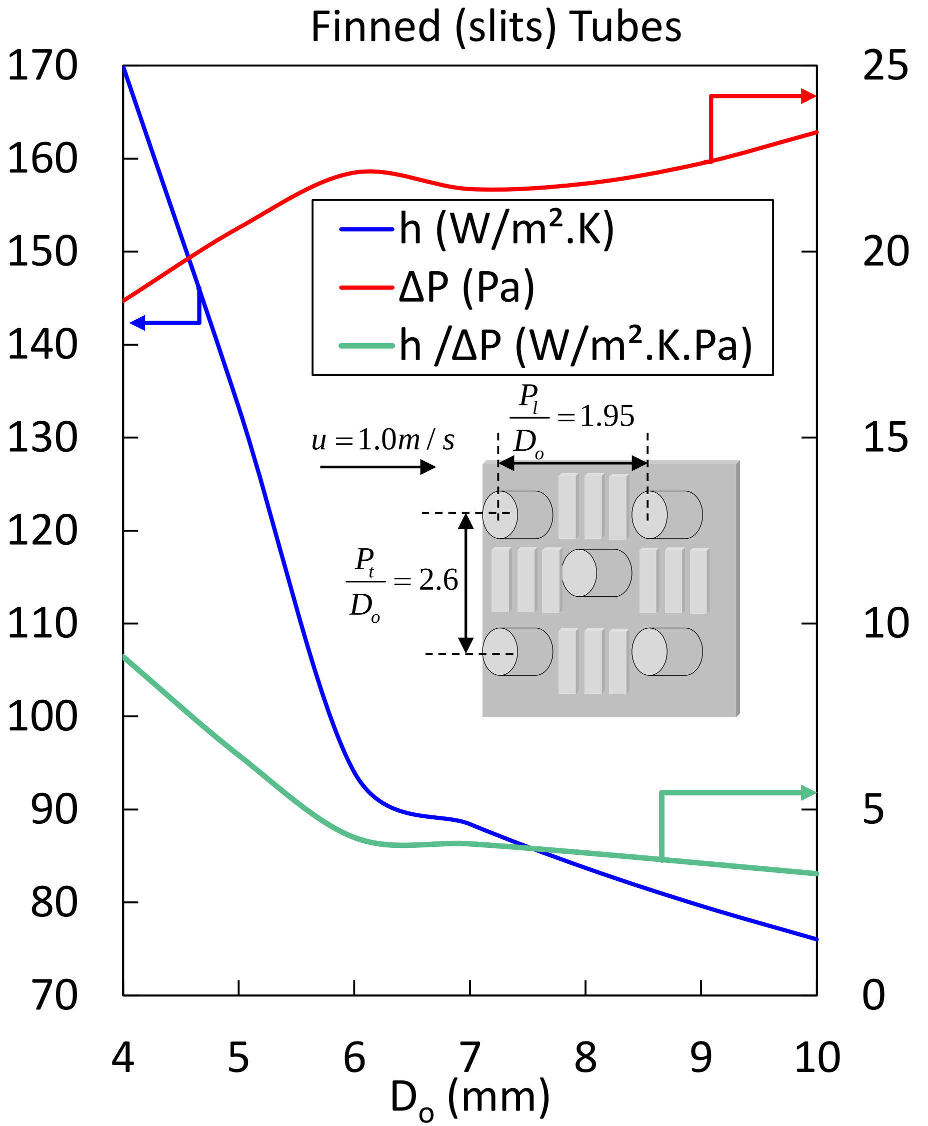


b) Small Diameter Tube



Example: Dry Air (1.0 atm, 300K)

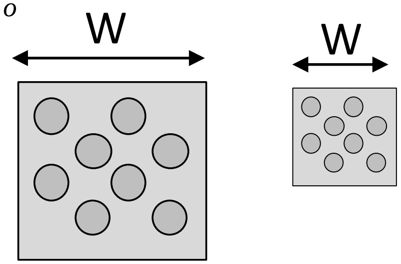


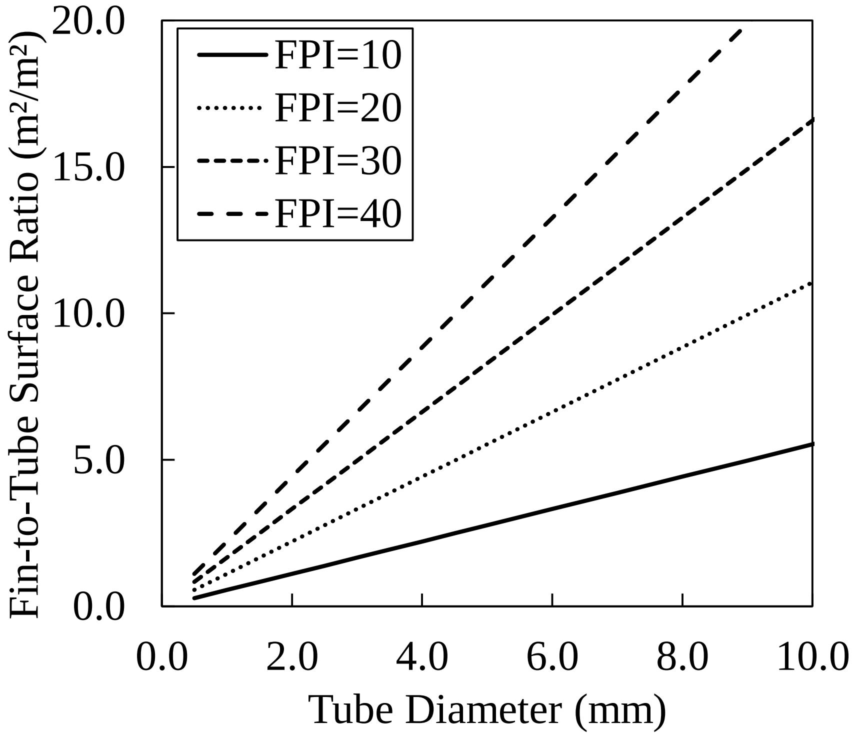


Discussion

- $D_o\downarrow$ → Relative fin surface area ↓
  - Impact (positive) on ΔP
  - *Less contribution to heat transfer*
- Airflow passage depth

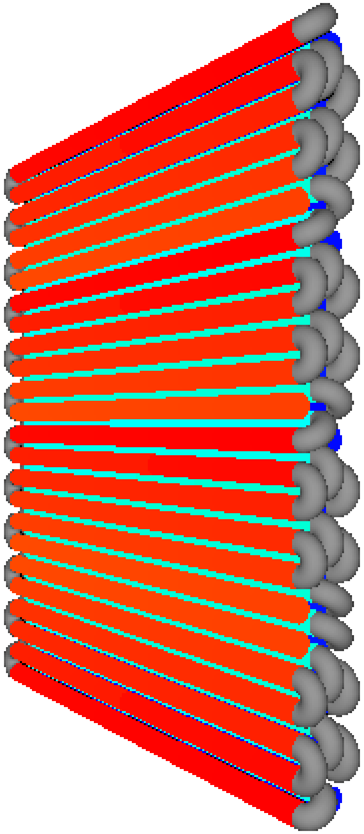
$$\frac{P_l}{D_o} = \text{fixed} \rightarrow D_o \downarrow \Rightarrow W \downarrow$$





- Friction resistance ≠ ΔP

(surface characteristic) vs. (HX characteristic)



Fundamentals

Refrigerant flow thermal-hydraulic performance

Internal Flow Momentum and Heat Transfer

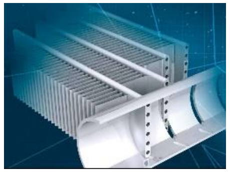
$$Nu \propto \mathrm{Re}^m \, \mathrm{Pr}^n \qquad f \propto \mathrm{Re}^p$$

$$h \propto D_h^{m-1} \cdot u^m \qquad f \propto D_h^p \cdot u^p$$

$$0 < m < 1 \qquad p < 0$$

$$D_h \downarrow \Rightarrow h \uparrow \qquad D_h \downarrow \Rightarrow f \uparrow$$

Microchannel with headers

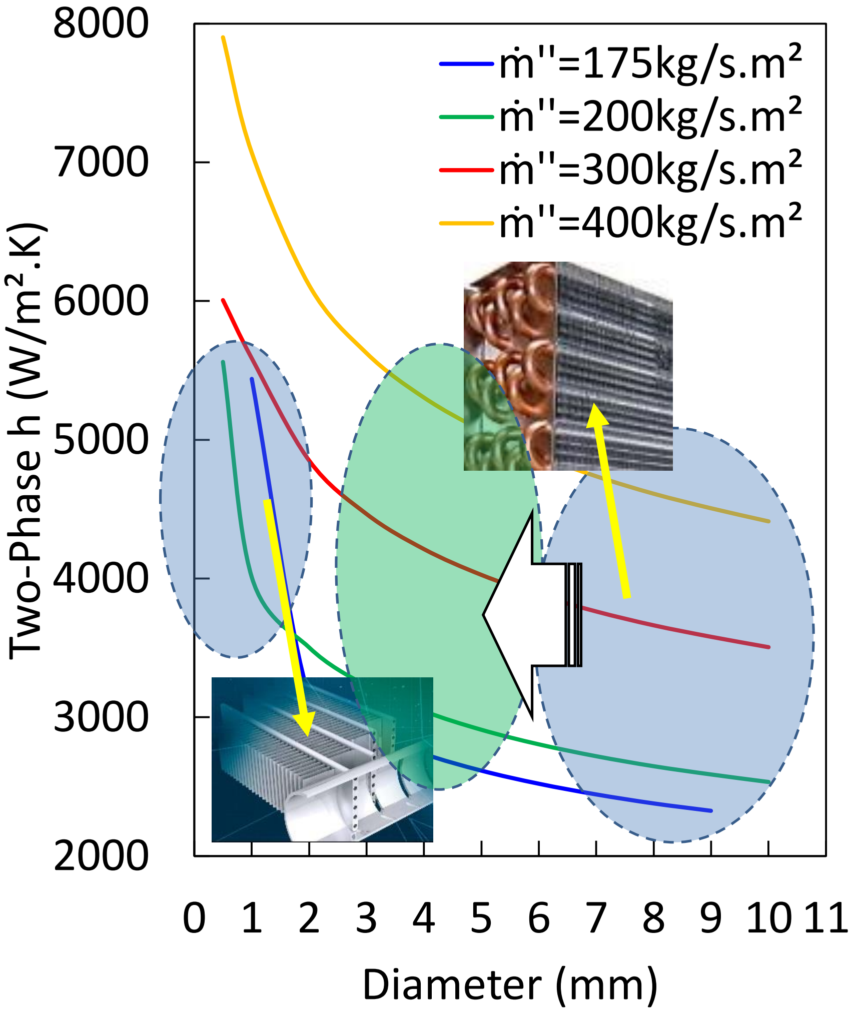


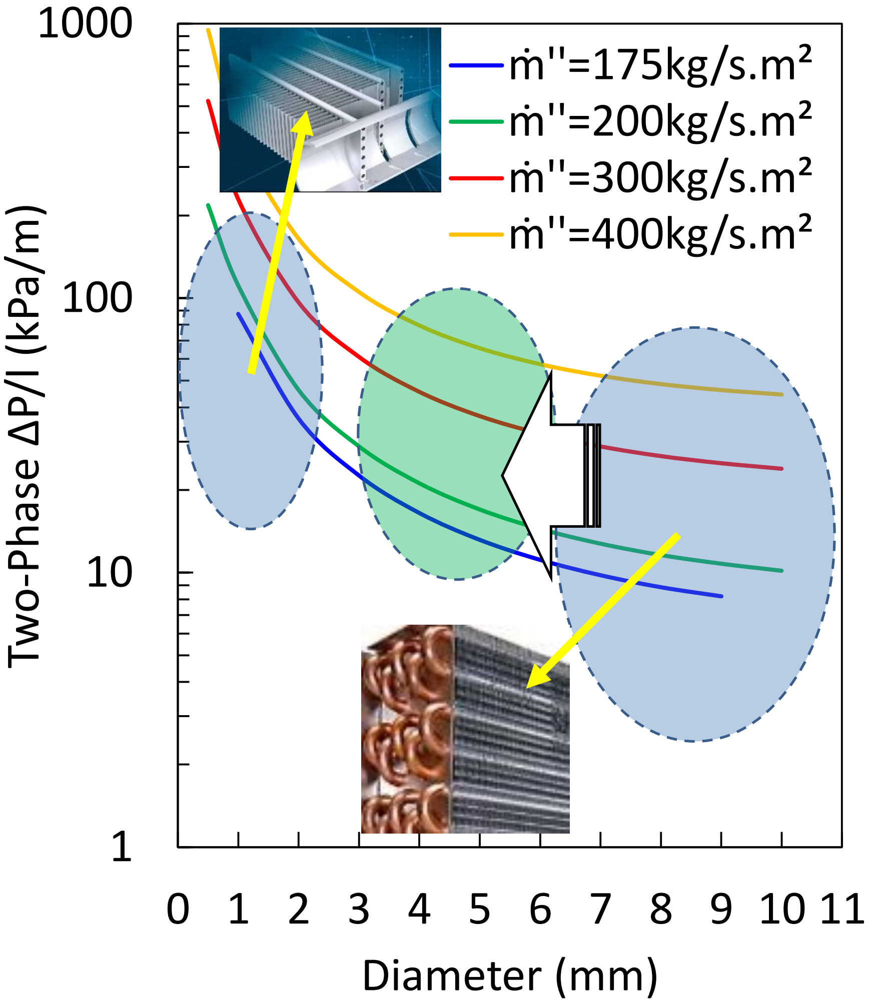
- Small D_h:
  - h↑
  - Compactness↑
- Multiple distribution tubes
  - ṁ”↓
  - Refrigerant path length↓
    - ΔP
  - Flow maldistribution
  - Header ΔP

Tube-fin with serpentine circuits

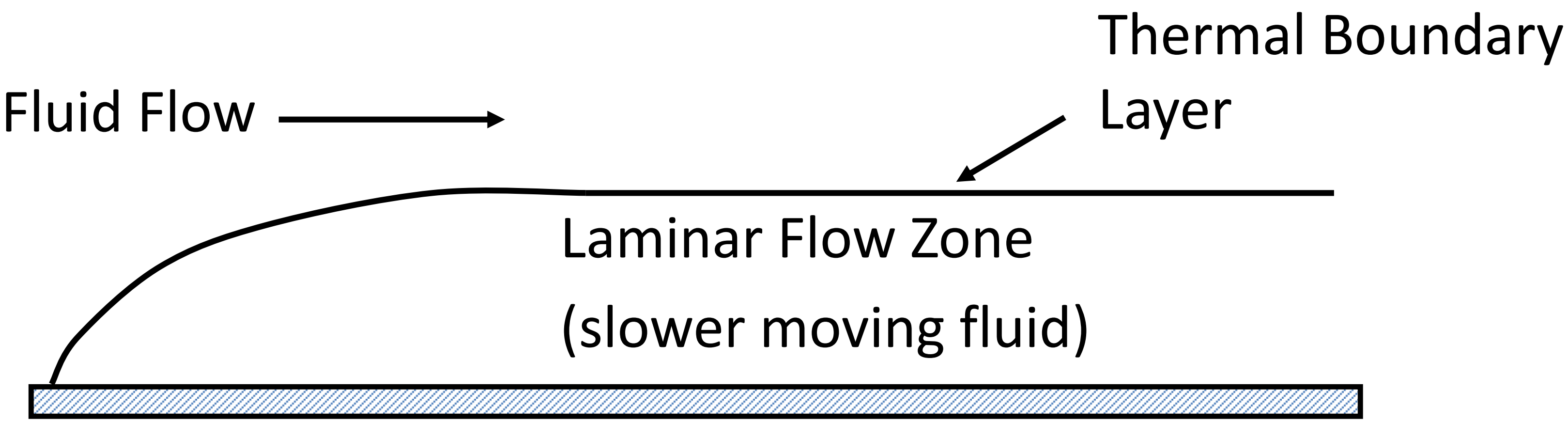
- Large D_h:
  - h ↓
  - Compactness ↓
- Few circuits
  - ṁ” ↑
  - Refrigerant path length ↑
    - ΔP

Example: Two-Phase R410A (2.7MPa)





Internally Enhanced Tubes



Fluid Flow

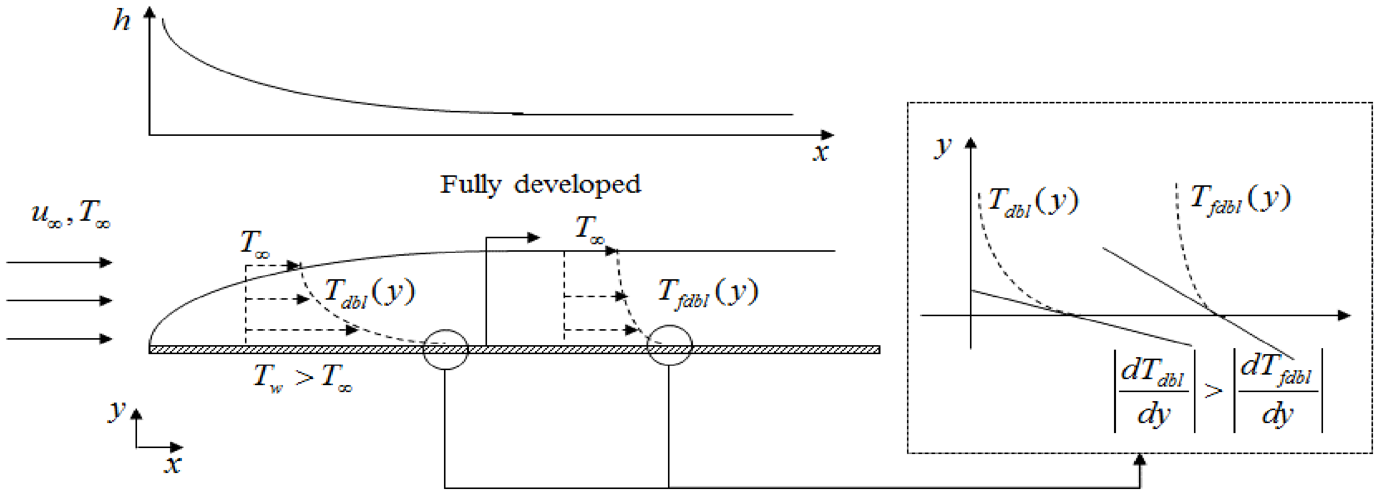
Thermal Boundary Layer

Laminar Flow Zone

(slower moving fluid)

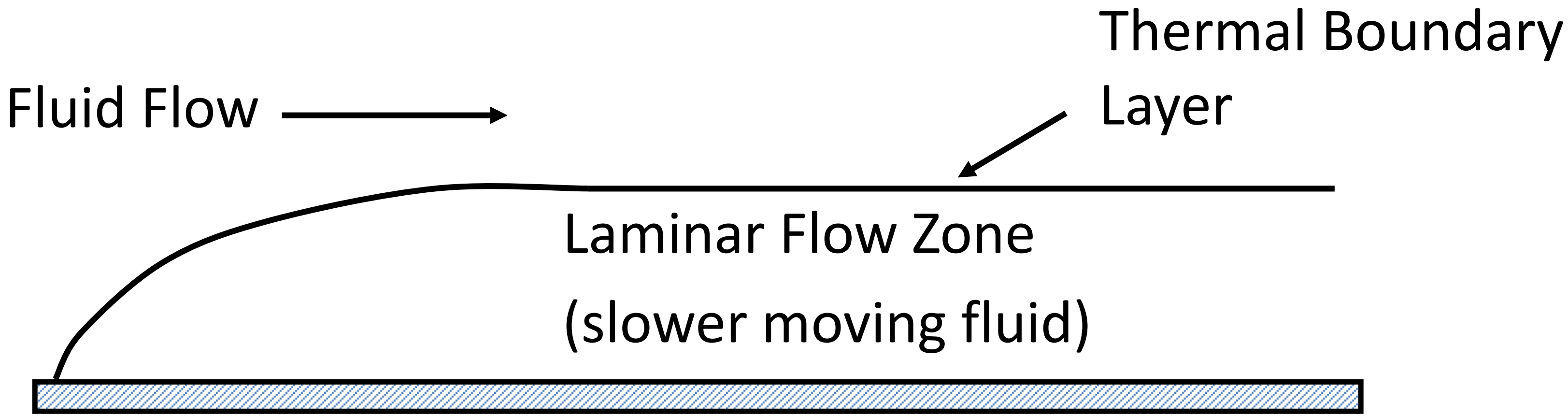
Smooth Internal Tube Wall

Courtesy of MicroGroove™



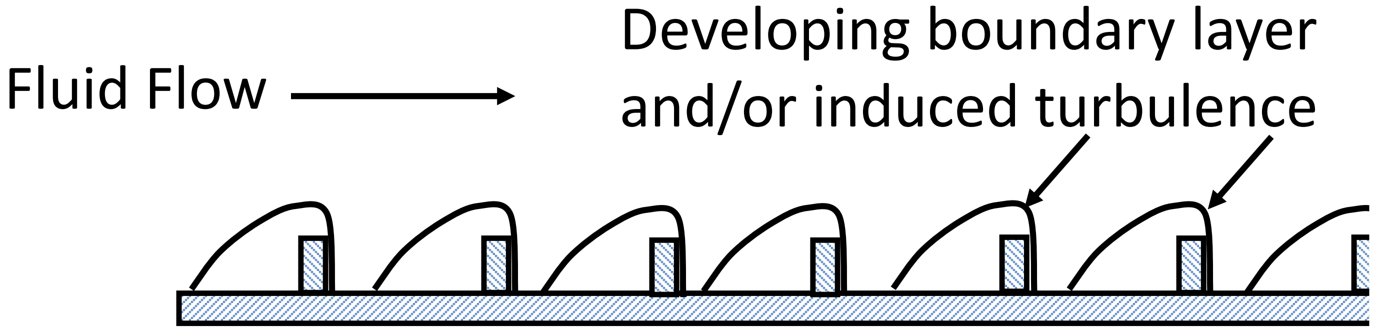
Courtesy of MicroGroove™

Internally Enhanced Tubes (cont'd)



Smooth Internal Tube Wall

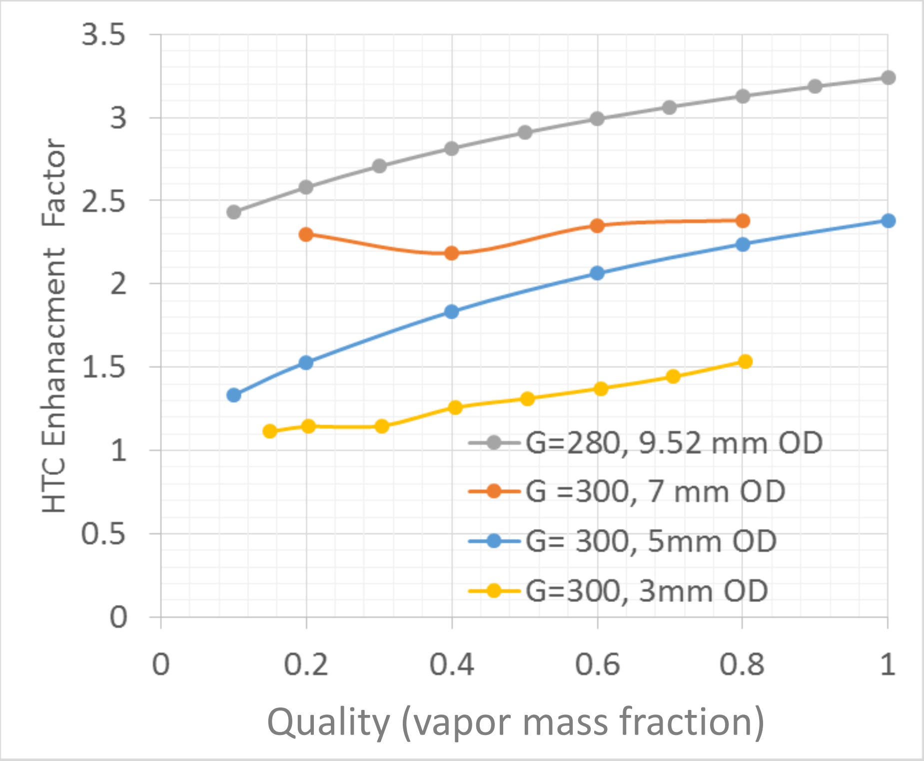
Courtesy of MicroGroove™

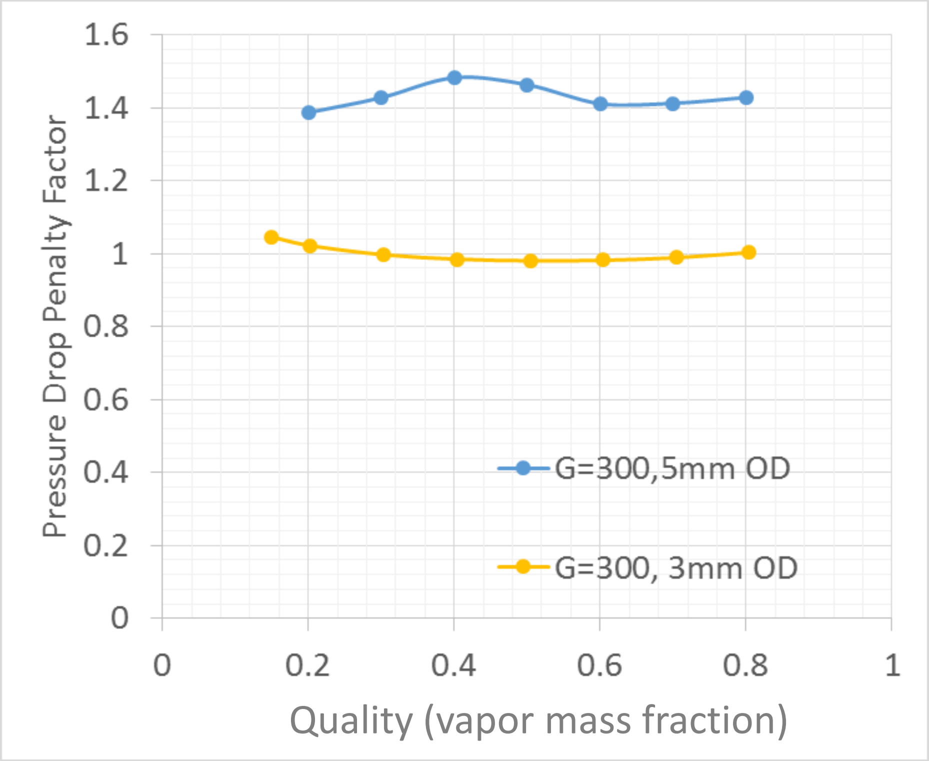


MicroGroove Tube

Courtesy of MicroGroove™

Internally Enhanced Tubes (cont'd)





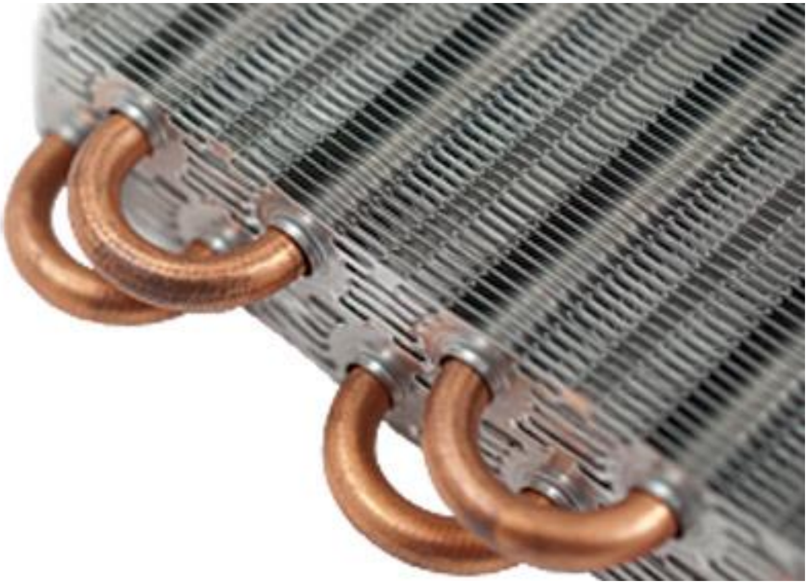
G= Mass flux (kg/s.m²)

- Quality↓ → HTC enhancement ↓
- D_o ↓ → HTC enhancement ↓

- ΔP Penalty Factor somewhat constant
- D_o ↓ → ΔP Penalty Factor ↓

Summary – Fundamentals

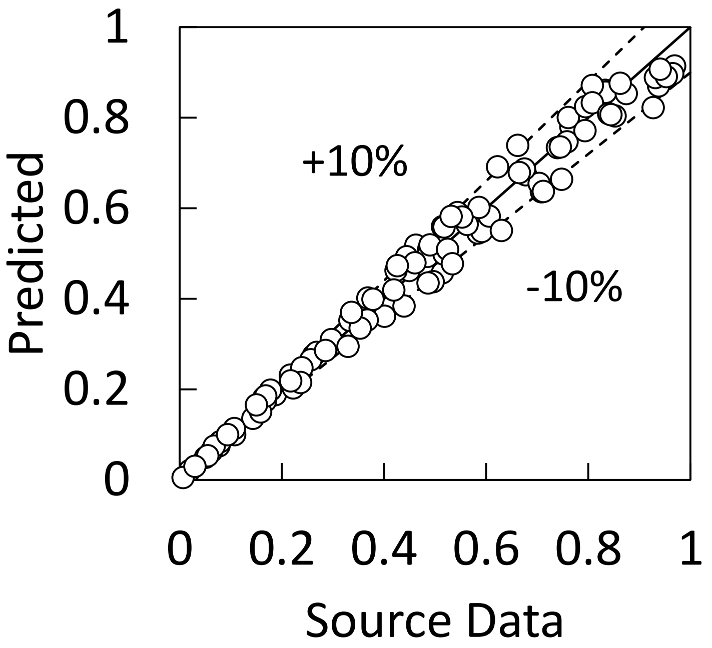
- First Order Analysis
  - More compact surfaces (more area per volume)
  - Less material consumption for same area
  - Smaller internal volume (less charge)
- Airside Thermal Hydraulic Characteristics
  - Higher Heat Transfer Coefficients at same velocities and Reynolds
- Refrigerant Side Thermal Hydraulic Characteristics
  - Higher Heat Transfer
  - Inner grooves enhance heat transfer regardless the tube diameter
- Challenges & Disadvantages
  - Higher friction factor → design changes to reduce pressure drop
  - Inner grooves enhancement reduce with tube diameter



Heat Exchanger Design

Design Considerations

- Airside friction resistance
  - Face area → reduce velocity (although typically undesirable)
  - Tube pitches → increase minimum free flow area
  - Fin density
- Refrigerant pressure drop
  - More circuits → reduce mass flux and flow length
  - Shorter tubes → reduce flow length
- Manufacturing (Webinar #02)
  - Fin collars → minimum fin density
  - Tube expansion
  - Fin dies, types, material, thickness
  - Number of tubes → Number of joints



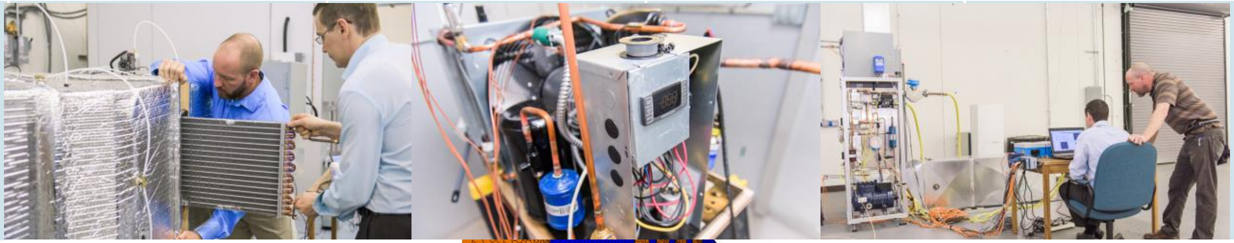
Heat Exchanger Design

Correlations

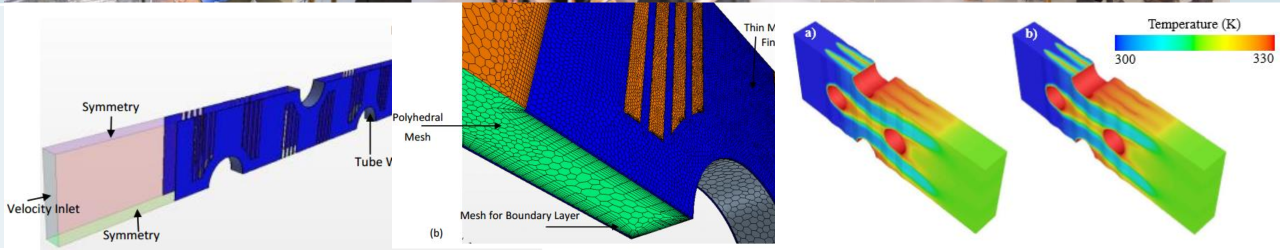
Importance of Correlations

Three methods to assess performance:

Experimental



Numerical (e.g. CFD)



Approximation (correlations)

$$j = c_1 \mathrm{Re}_{D_o}^{P_1} N_t^{P_2} \left(\frac{F_p}{D_o + 2\delta_f}\right)^{P_3} \left(\frac{P_t}{D_o}\right)^{P_4} \left(\frac{P_l}{D_o}\right)^{c_2}$$

$$f = c_1 \mathrm{Re}_{D_o}^{P_1} N_t^{P_2} \left(\frac{F_p}{D_o + 2\delta_f}\right)^{P_3} \left(\frac{F_p}{D_o}\right)^{P_4} \left(\frac{F_p}{P_t}\right)^{c_2}$$

Importance of Correlations (cont'd)

Method	Accuracy	Engineering Cost	Capital Cost	Computational Cost
Experimental	Very High	High	Very High	Low
Numerical (e.g. CFD)	High	Medium	High	Very High
Approximation (correlations)	Medium-High	Low	Low	Very Low

Most cost-effective in assessing performance!

Literature Survey - Empirical

Author	Year	Fin type	Tube arrangement	Applicability	Regression Uncertainty
Grimison	1937	No fins	Inline/ Staggered	Re>2000 (D_h range unknown) 1.25 ≤ Pt ≤ 3.0 D_o 0.6 ≤ Pl ≤ 3.0 D_o, 3 ≤ N ≤ 9	N/A
Žukauskas	1972	No fins	Inline/ Staggered	1≤Re≤10^6 (D_h range unknown) 1.25 ≤ Pl, Pt ≤ 3.0 Do	N/A
McQuiston	1978	Plain	Staggered	9.675≤ D_o ≤ 16.13mm; 1.0 ≤ u ≤ 4m/s 25.4mm ≤ Pl, Pt ≤ 50.8mm	15%
Gray	1986	Plain	Staggered	D_o>9.96mm, 500≤Re≤24000 N ≤ 4, 1.7 ≤ Pl, Pt ≤ 2.6 D_o	7.3% (HT), 13% (F)
Webb	1990	Plain	Staggered	D_c ≥ 7.95mm, 2 ≤ N ≤ 6 1.15 ≤ Pt/Pl ≤ 1.67	5%
Kim et al.	1999	Plain	Staggered	7.3 ≤ D_o ≤ 19.3mm	20%
Wang et al.	2001	Plain	Staggered	6.35 ≤ D_c ≤ 12.7mm, 1 ≤ N ≤ 6 17.7 ≤ Pt ≤ 31.8, 12.4 ≤ Pl ≤ 27.5mm	15%
Webb	1990	Wavy	Staggered	D_c ≥ 7.95mm, N = 3 Pt/Pl = 1.15	5%
Kim et al.	1997	Wavy	Inline/ Staggered	D_o = 10.46, 500≤Re≤9000 1 ≤ N ≤ 8, 1.16 ≤ Pt/Pl ≤ 1.33	10% (HT), 15% (F)
Wang et al.	2002	Wavy	Staggered	7.66 ≤ D_c ≤ 16.85mm, 12 ≤ Pl ≤ 33mm 1 ≤ N ≤ 6; 21 ≤ Pt ≤ 38mm	15%
Wang et al.	1999	Louver	Staggered	6.93 ≤ D_c ≤ 10.4, 17.7 ≤ Pt ≤ 25.4mm 1 ≤ N ≤ 6, 12.7 ≤ Pl ≤ 19.05mm	15%
Wang et al.	1999	Slit	Staggered	D_c = 10.34mm, 1 ≤ N ≤ 6	10%

D_o > 6.0mm

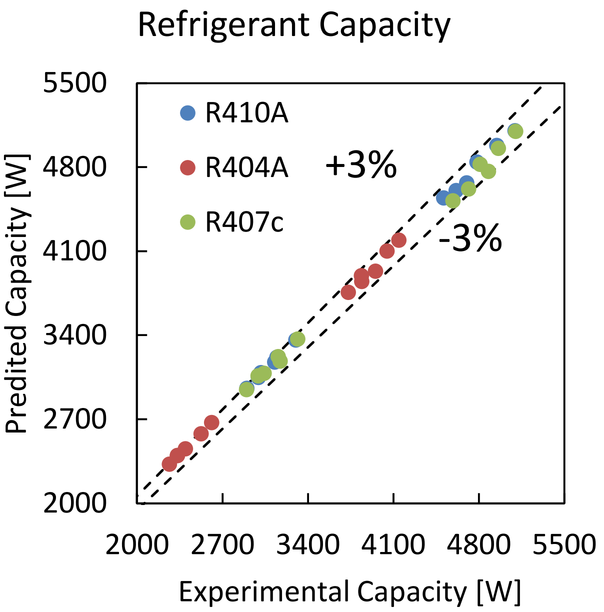
Literature Survey – CFD-based

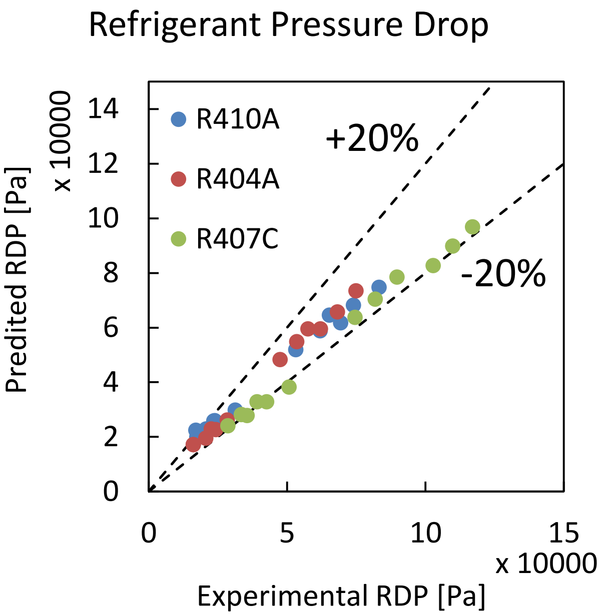
Author	Fin type	Tube arrangement	Application range	Accuracy (staggered only)
Bacellar et al. (2015)	Finless	Staggered	0.5mm≤Do≤2.0mm 2≤N≤40	j : 15%(92.9% of data) f: 15% (88.1% of data)
Bacellar et al. (2016)	Finless	In-line	0.5mm≤Do≤2.0mm 2≤N≤40	j : 20%(80% of data) f: 20%(80% of data)
Bacellar et al. (2014)	Finless	Staggered	2.0mm≤Do≤5.0mm 2≤N≤20	j : 10%(98.5% of data) f: 10%(91.9% of data)
Bacellar et al. (2014)	Flat	Staggered	2.0mm≤Do≤5.0mm 2≤N≤20	j : 15%(82.1% of data) f: 15% (82.3% of data)
Bacellar et al. (2016)	Wavy Herringbone	Staggered	2.0mm≤Do≤5.0mm 2≤N≤20	Nu: 15%(96% of data) Cf: 15%(94% of data)
Bacellar et al. (2015)	Wavy Smooth	Staggered	2.0mm≤Dc≤5.0mm 2≤N≤10	j : 20%(64% of data) f: 20% (66% of data)
Bacellar et al. (2016)	Wavy Smooth	Staggered	2.0mm≤Do≤5.0mm 2≤N≤20	Nu: 15%(94% of data) Cf: 15%(93% of data)
Sarpotdar et al. (2016)	Slit	Staggered	3.0mm≤Do≤5.0mm 2≤N≤6	h: 15%(99% of data) ΔP: 15%(93% of data)
Sarpotdar et al. (2016)	Louver	Staggered	3.0mm≤Do≤5.0mm 2≤N≤8	h: 15%(99% of data) ΔP: 15%(94% of data)

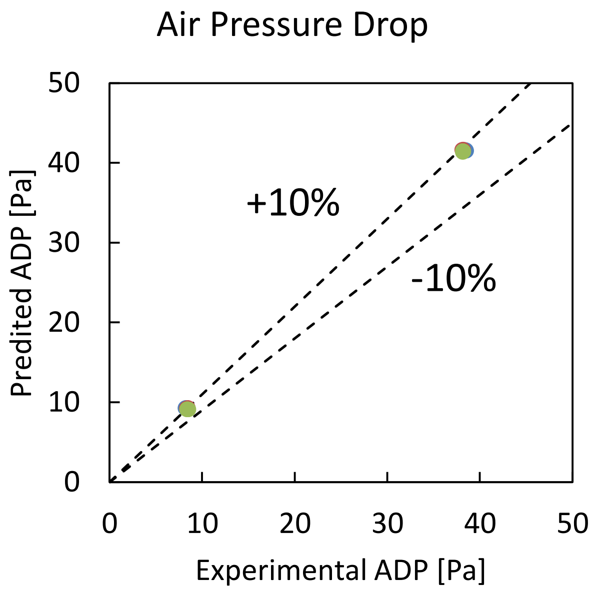
Refrigerant Correlations – Internally Enhanced Tubes

Author	Phase	Refrigerant	Application
Shlager et al. (1989)	Two-Phase (horiz.)	Any	-
Ravigururajan & Bergles (1996)	Single Phase	Any	Re = 5000-60,000
Koyama & Yonemoto (2006)	Condensation (horiz.)	Any	Mass Flux: 100-500 kg/m2s
Koyama & Yonemoto (2006)	Condensation (horiz.)	CO2	Mass Flux: 100-500 kg/m2s
Shah (2016)	Condensation (horiz.)	Water, R-11, R-12, R-22, R-32, R-113, R-123, R-125, R-134a, R-142b, R-404A, R-410A, R-502, R-507, isobutane, propylene, propane, benzene, ethanol, methanol, toluene, Dowtherm 209, DME, CO2	Tube diameter, mm (in.) 2 to 49 (0.079 to 1.93)
Wu et al. (2013)	Evaporation	Any	Nominal diameter: 2.1-14.8mm
Son & Oh (2012)	Condensation (horiz.)	CO2	ID = 4.6 - 4.95mm;

Correlation Validation


Refrigerant Capacity


Refrigerant Pressure Drop


Air Pressure Drop

- **Using new, tuned, louver fin correlation Sarpotdar et al. (2016), CoilDesigner® predictions match experimental results very closely**
- **Koyama & Yonemoto (2006) correlation shows acceptable refrigerant-side performance**

Tuning Correlations

- Experimental data for correlation Verification & Validation

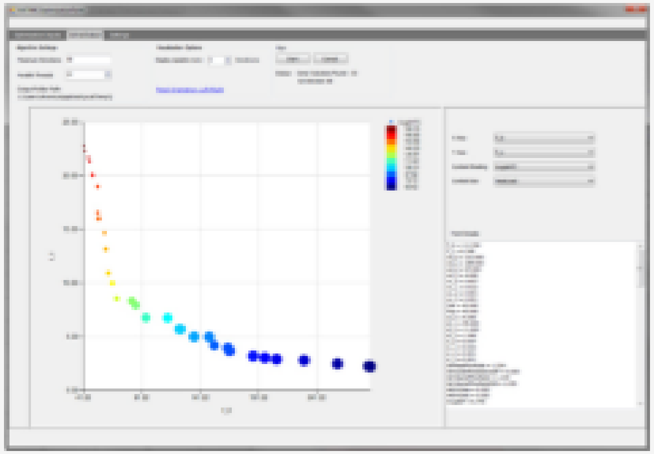
- Opportunity for collaboration
  - Cross-validation with data from multiple sources
  - Large data base
  - Publications
  - Access

Heat Exchanger Design

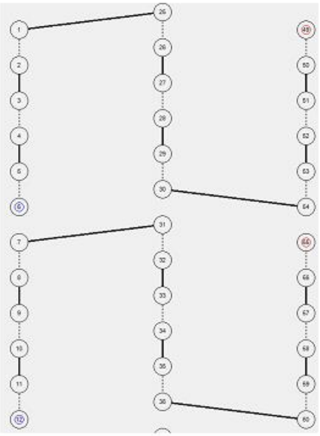
Tools

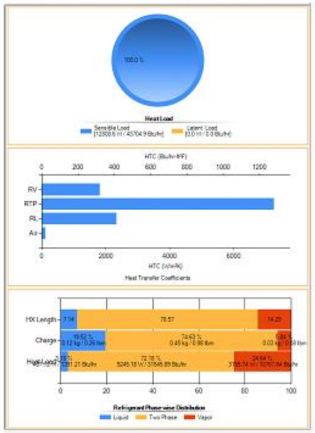
CoilDesigner® - Webinar #03

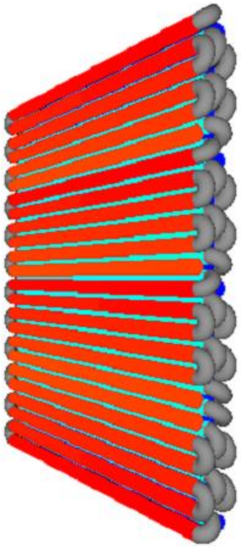
CoilDesigner® is a highly customizable software tool that designs, simulates and optimizes the performance of a variety of heat exchangers. This unique tool helps to shorten product development time frames and associated costs. With one integrated tool, you can design your product, simulate its performance, and optimize it for multiple objectives (e.g. cost, efficiency, and power consumption).

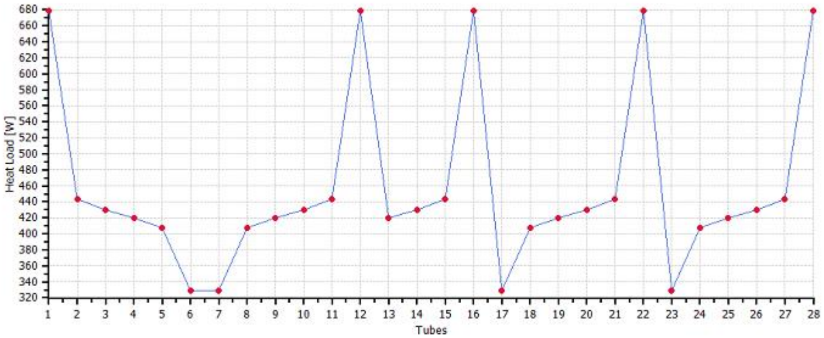


Optimization and Parametric Analysis!









Applications

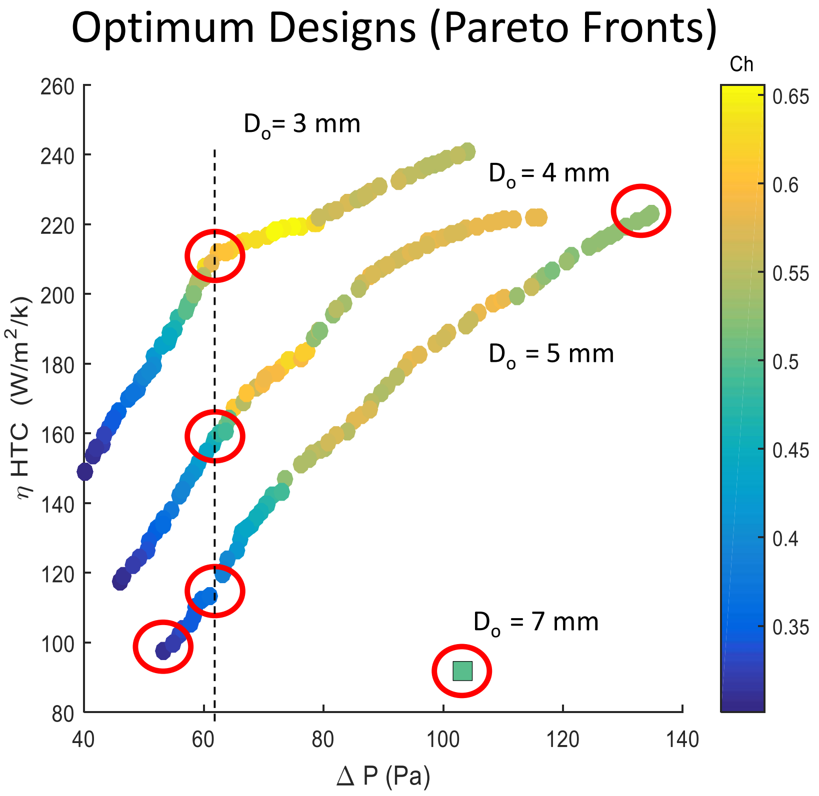
Case Studies

- Study I: Airside Performance Optimization
  - New correlations
  - Maximize heat transfer coefficient and minimize pressure drop
- Study II: Split Condenser Optimization
  - New correlations + CoilDesigner®
  - Minimize air pressure drop, raw material cost and refrigerant charge
- Study III: Window AC Condenser
  - New Correlations + CoilDesigner®
  - Improve system’s performance, reduce cost & refrigerant charge

Study I: Airside Performance Optimization

- **New correlations reveal performance potential of smaller diameter tube heat exchangers**
- **Figure: 3, 4, 5 mm slit fin performance (shaded by slit height)**
- **Sample fin designs:**

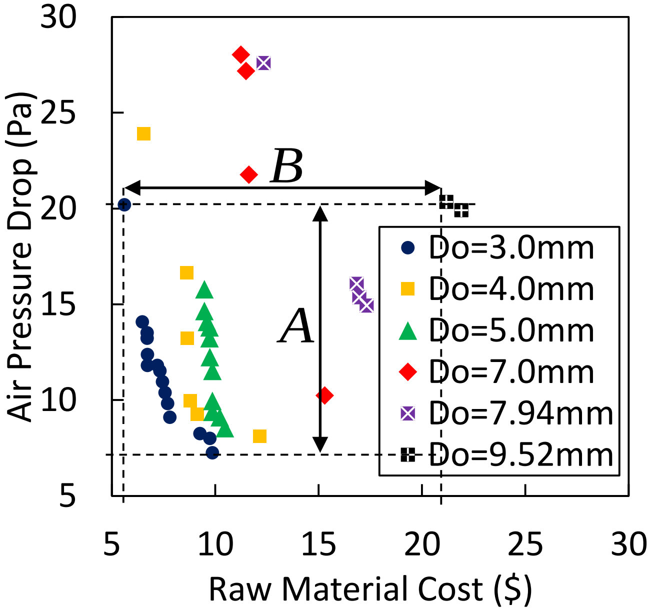
Metric	7 mm	5 mm (max HTC)	5 mm (min ΔP)	5 mm sample	4 mm sample	3 mm sample
D_o (mm)	7	5	5	5	4	3
u (m/s)	2.875	2.875	2.875	2.875	2.875	2.875
HTC (W/m^2K)	**92.4**	**222.8**	**97.3**	**126.5**	**160.5**	**214.3**
ΔP (Pa)	**113.4**	**134.8**	**53.2**	**65.4**	**63.53**	**65.5**

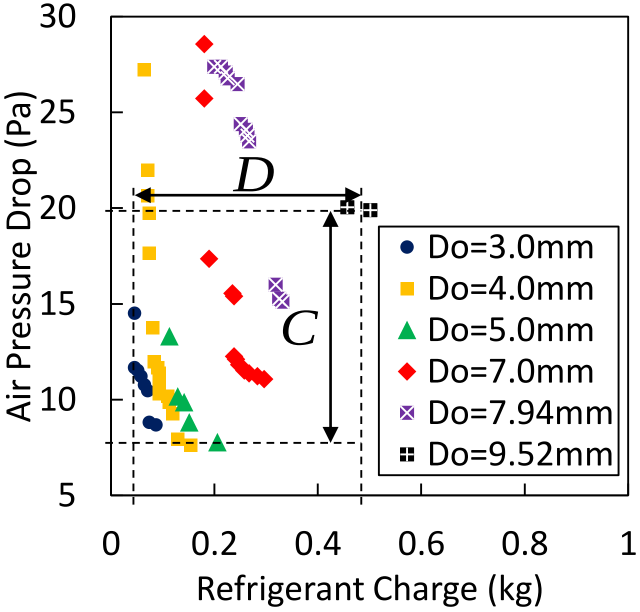


Sarpotdar et al. , CFD Based Comparison of Slit Fin and Louver Fin Performance for Small Diameter (3mm to 5 mm) Heat Exchangers, 16th International Refrigeration and Air Conditioning Conference at Purdue, July 11-14, 2016

Study II: Split Condenser Optimization

- Identify optimal drop-in replacement condensers for room AC
- Constraints: performance should be equivalent to the baseline
- Objectives: minimize air-side pressure drop (improve efficiency), material consumption, and refrigerant charge





$A = 64\% \downarrow$

$B = 74\% \downarrow$

$C = 62\% \downarrow$

$D = 91\% \downarrow$

Study III: Window AC Condenser

- Improve system performance
- Reduce cost
- Reduce refrigerant charge
- Slit and louver fin designs considered, working with a manufacturing partner

FRIEDRICH

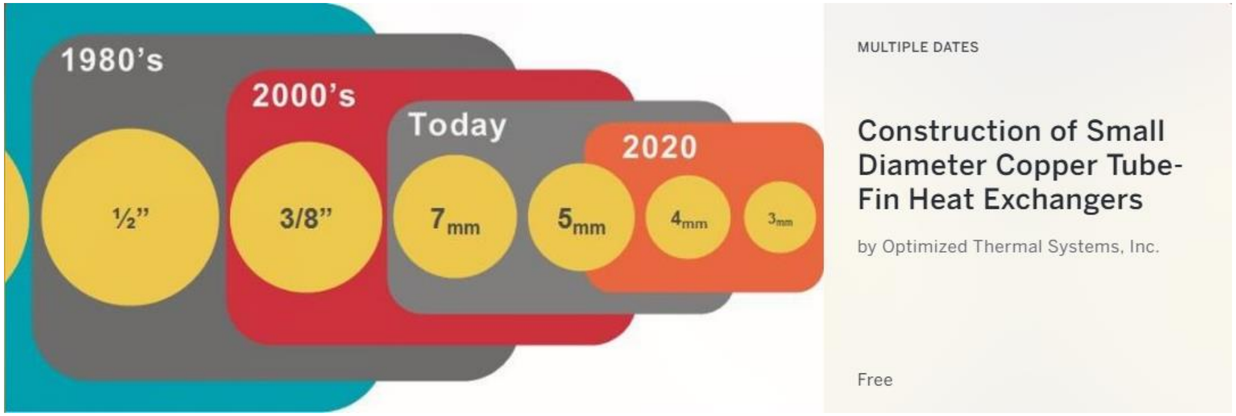
Design	Tube material [kg]	Fin material [kg]	Material Reduction [%]	Simulated Cond. Charge Reduction [%]	Simulated (Measured) COP	COP Improvement [%]
Baseline	1.8	2.5	-	-	2.60 (2.86)	-
Slit fin 17 FPI	1.4	2.0	**21%**	**24%**	2.76	**6.1%**
Louver fin 15 FPI 3 row	1.0	1.3	**47%**	**46%**	2.64	**1.5%**
Louver fin 13 FPI 4 row	1.3	1.4	**37%**	**10%**	2.77	**6.5%**

Experiment: Louver fin coil achieves 10% *system* charge reduction and 4% COP increase while reducing cost by approximately 40%

Take Home Messages

- Small diameters
  - Compact, less material & less charge
  - Better heat transfer, higher friction
- Copper tubes
  - Low thermal resistance
  - Withstands high pressures with less material (small diameter)
  - Corrosion & biofouling resistance
  - Ease of inner grooving
- Design tools
  - Correlations are the most cost-effective way of assessing performance
  - Need for experimental data for tuning/modifying these tools
- Case studies
  - 3.0mm ~70% higher heat transfer than 5.0mm at same air pressure drop
  - ~60% air pressure drop reduction, ~75% material reduction, ~90% less charge
  - Real system resulted in ~10% less charge (system not heat exchanger), ~4% more COP

Coming Next



MULTIPLE DATES

Construction of Small Diameter Copper Tube-Fin Heat Exchangers

by Optimized Thermal Systems, Inc.

Free

Webinar #02

When:

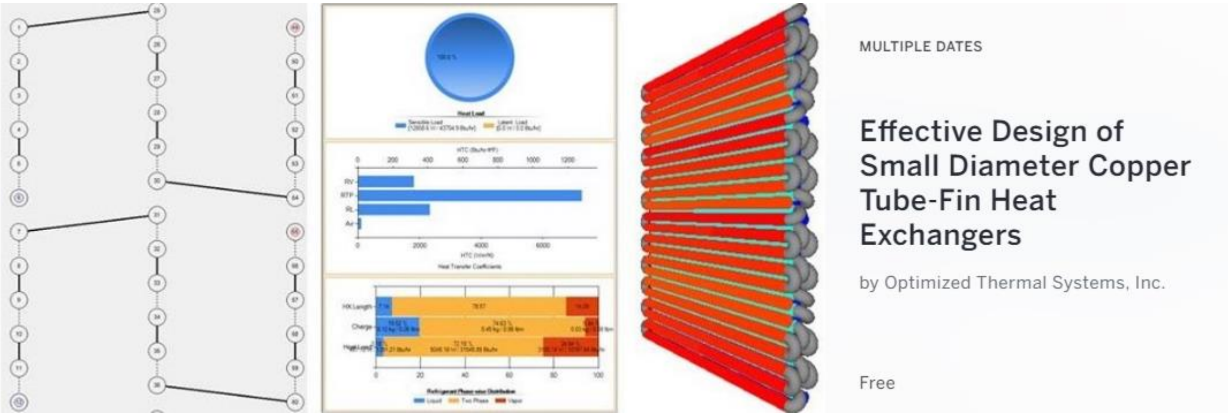
Wednesday April 26th, 2017
Morning Session: 8:00AM EST
Afternoon Session: 4:00PM EST

Registration: http://www.microgroove.net/ots-ica-educational-outreach

Webinar #03

When:

Wednesday May 24th, 2017
Morning Session: 8:00AM EST
Afternoon Session: 4:00PM EST



MULTIPLE DATES

Effective Design of Small Diameter Copper Tube-Fin Heat Exchangers

by Optimized Thermal Systems, Inc.

Free

THANK YOU!

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