# Response time characterization of ORC evaporators for dynamic regime analysis with fluctuating thermal power

**The 4th International Seminar on ORC Power Systems**

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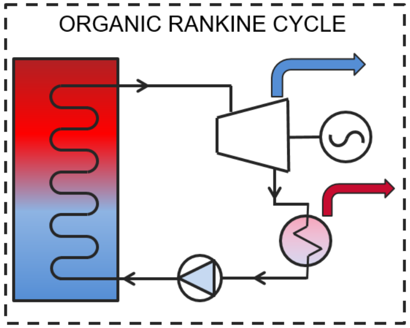
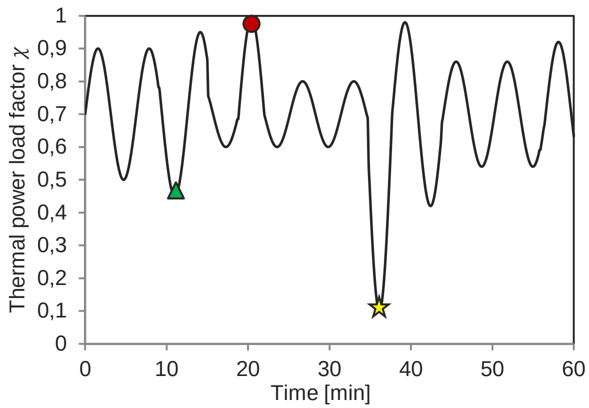
15 September 2017
Milan, Italy

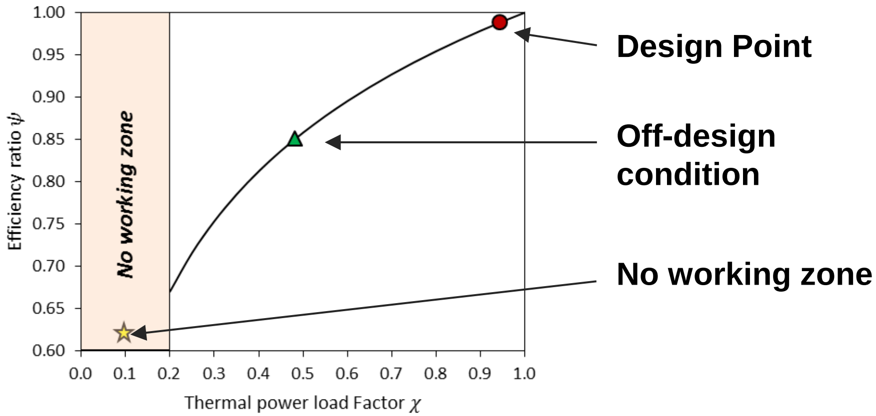
**Waste heat recovery**
- Stationary sources
- Mobile sources




**Exemplary profile of waste heat**




**Efficiency curve of ORC system**

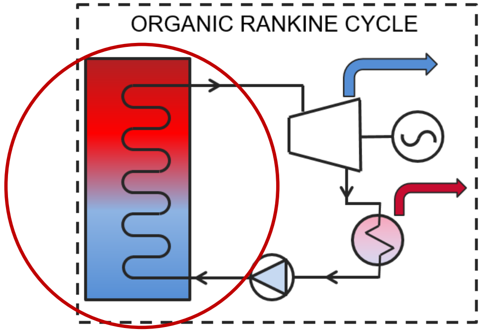



**Fluctuating source challenge**

**- Off-design conditions most of the time: poor efficiency**

**- Outside operating range: ORC downtimes**

**Waste heat recovery**
- Stationary sources
- Mobile sources

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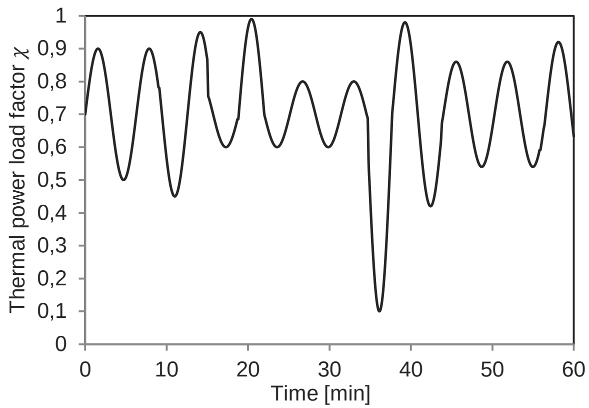


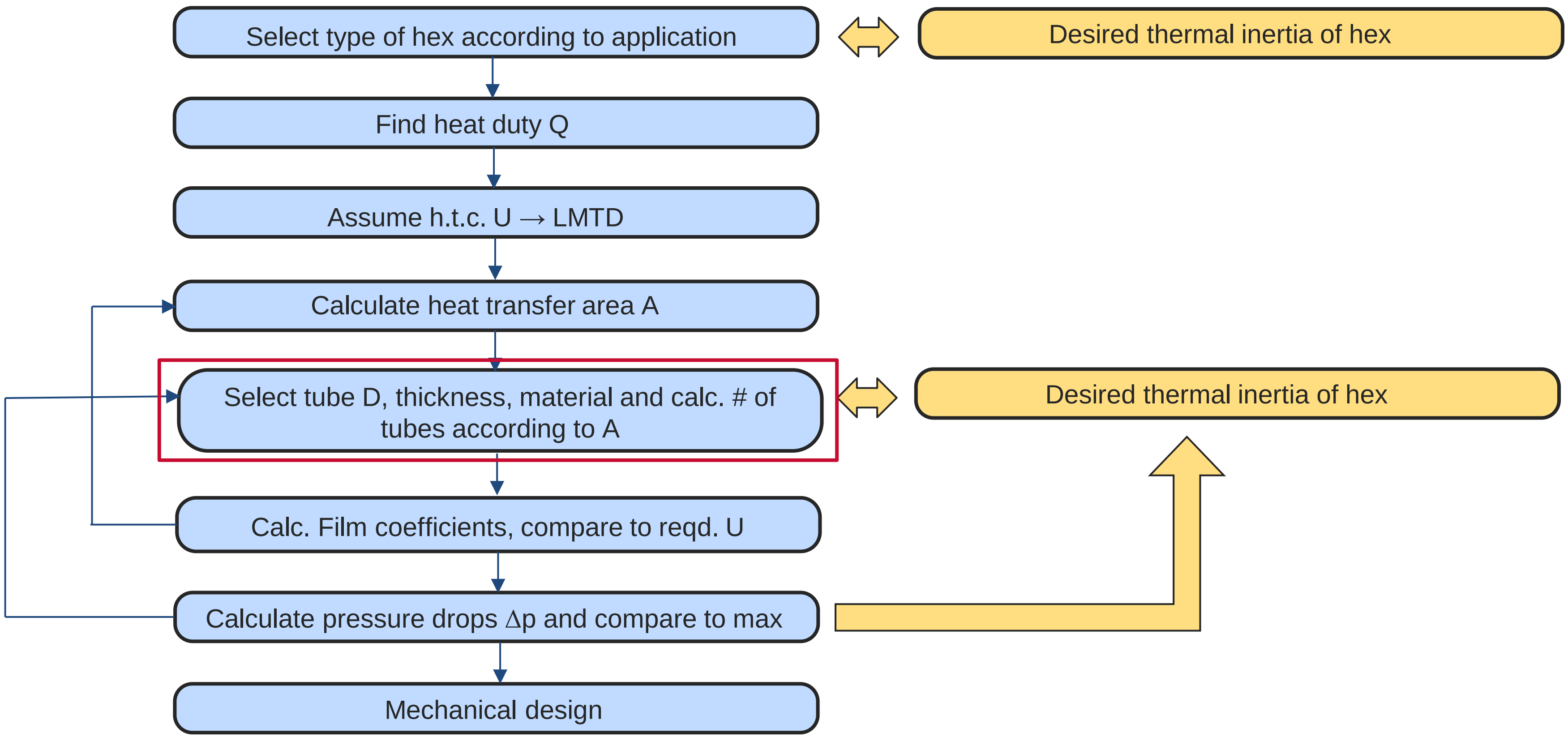

Fluctuating thermal power input means that the ORC system often experiences transients

- ORC evaporator: link between heat source and rest of components
- Dynamics of ORC dominated by heat exchanger transients
- <u>Evaporator intrinsic thermal inertia affects the dynamic behavior of the ORC under fluctuating thermal power</u>

# ORC evaporator design for dynamic behavior

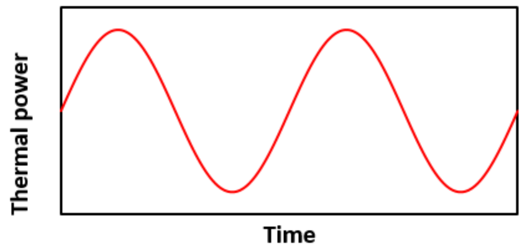
Standard methodology for heat exchanger design:

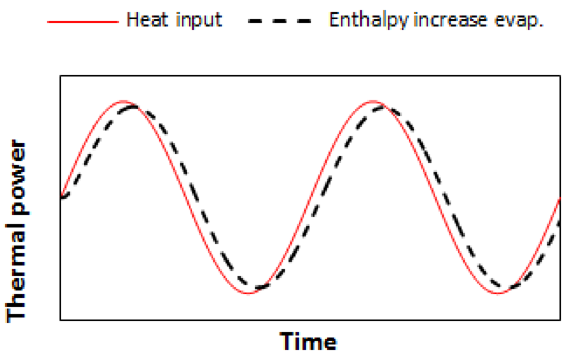
Include desired dynamic behavior of evaporator:



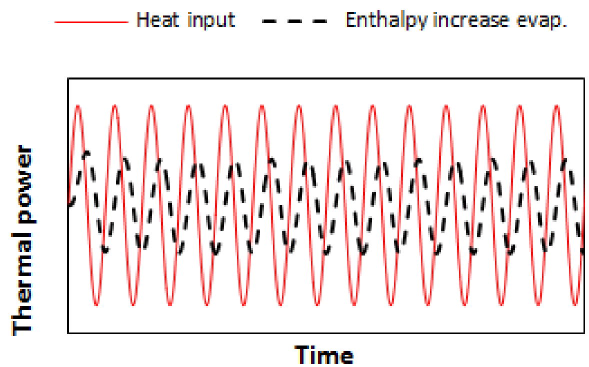
# Dynamic regimes and dynamic regime number

**Thermal power input**

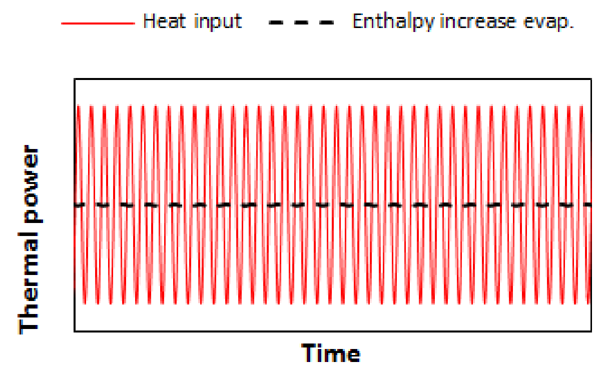



**Dynamic regime I**
*Quasi-steady*




**Dynamic regime II**
*Transient*




**Dynamic regime III**
*Quasi-constant*

# Dynamic regimes and dynamic regime number

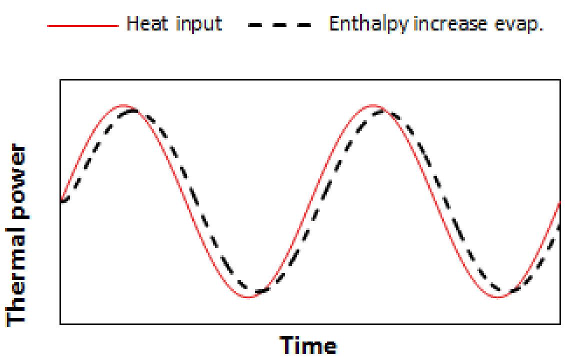
Define a characteristic response time of the evaporator $\tau_{ev}$

Period of fluctuation of the thermal power $T_{load}$

**Dynamic regime number *Γ*:** ratio of response time to period of load fluctuation

$$\Gamma = \frac{\tau_{ev}}{T_{load}}$$

How does the design parameters of the hex affect its response time?

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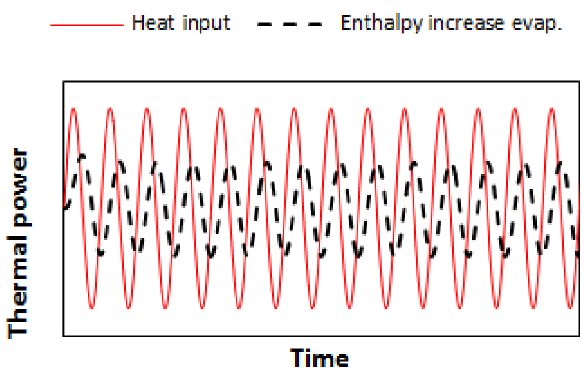
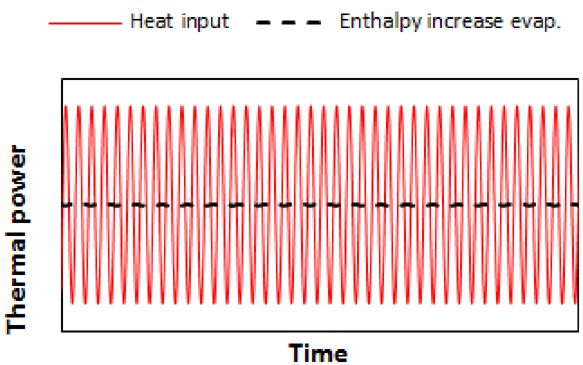


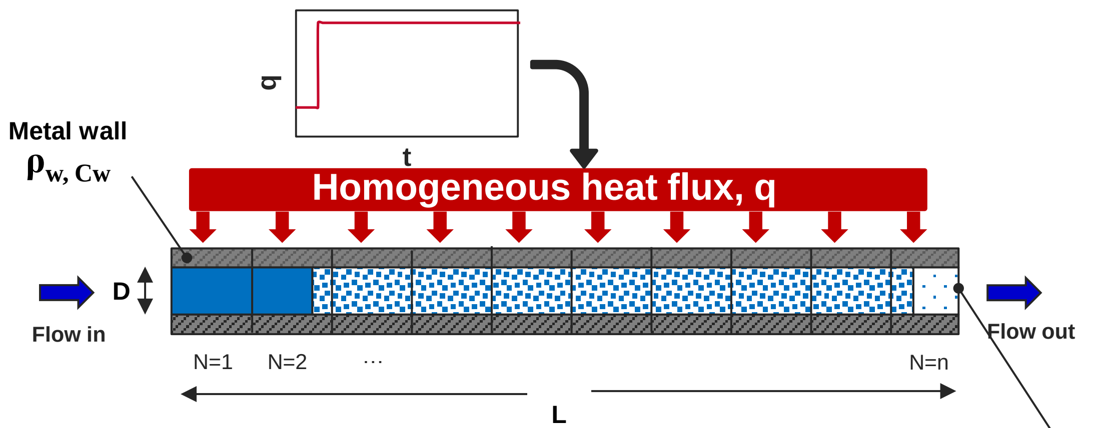

**Dynamic regime II**
*Transient*



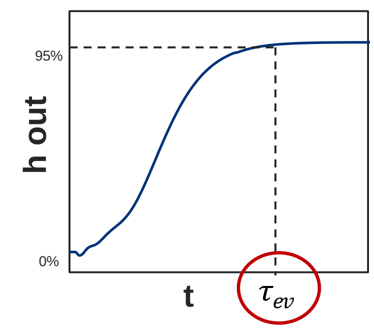

**Dynamic regime III**
*Quasi-constant*

# Methodology for dynamic characterization

**Finite volume 1-D model of evaporator:**




**Evaporator characteristic response time: $\tau_{ev}$**




**For each 1-D cell:**

**Continuity equation:** $$\frac{dM}{dt} = \frac{d(V\cdot\rho)}{dt} = V\cdot\left(\frac{\partial\rho}{\partial h}\frac{dh}{dt} + \frac{\partial\rho}{\partial p}\frac{dp}{dt}\right) = \dot{m}_{in} - \dot{m}_{out}$$

**Energy equation:** $$V\cdot\rho\cdot\frac{dh}{dt} - V\cdot\frac{dp}{dt} = \dot{m}_{in}\cdot h_{in} - \dot{m}_{out}\cdot h_{out} + \alpha_i\cdot A_{ht}\cdot(T_w - T)$$

**Heat balance in wall:** $$C_w M_w \frac{dT_w}{dt} = \dot{q}_o\cdot A_{ht} + \alpha_i\cdot A_{ht}\cdot(T - T_w)$$

**Heat transfer correlations:** 1p – Gnielinski, 2p - Shah

# Generalization of results: Dimensionless parameters

**Response time as function of dimensionless parameters:**

$$\frac{D}{L}$$

**Geometric ratio(s)**

$$Ja_{lv} = \frac{C_{p,v}(T_v - T_{sat}) + C_{p,l}(T_{sat} - T_l)}{\Delta H_{vap}}$$

**Jakob number:** relative ratio of sensible to latent heat transfer

$$CapR = \frac{\rho_w C_w}{\rho_{tp\ avg} \cdot (\frac{\Delta H_{vap}}{T_{sat}})}$$

**Cap Ratio:** ratio of wall heat capacity to a "relative heat capacity of fluid"

**1) Geometry**

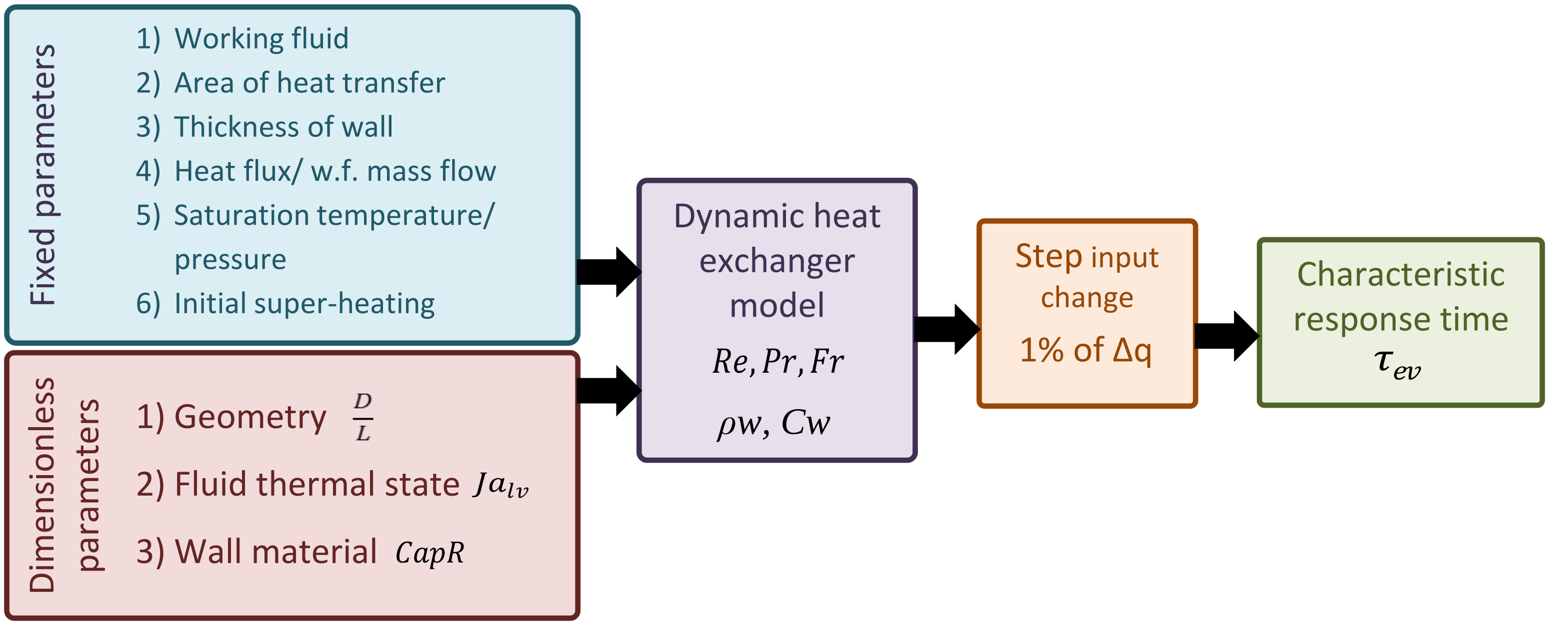
**+**

**2) Fluid thermal state**

**+**

**3) Wall material**

# Summary of methodology



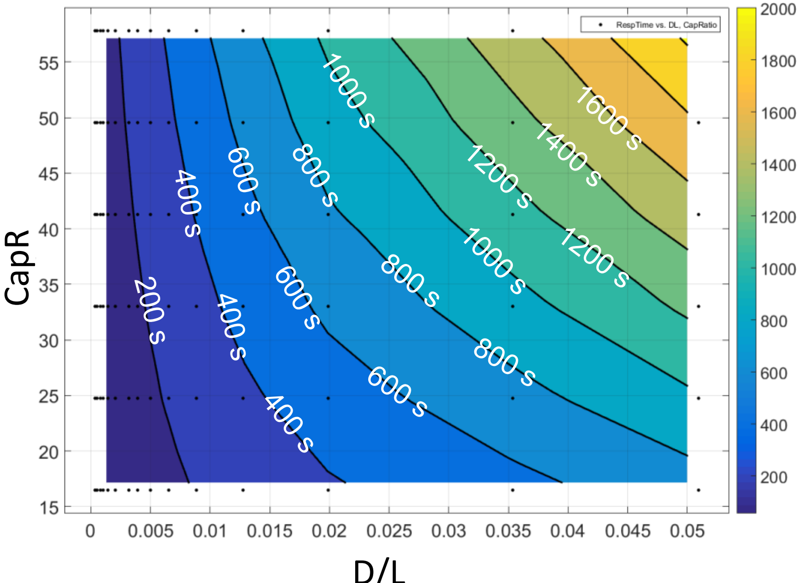

- Parametrization of $\tau_{ev}$ as function of dimensionless parameters
- From parametric points interpolate to build charts with “constant response” time curves

# ORC evaporator response time charts

**Working fluid thermal state**

**Ja = 0.400**

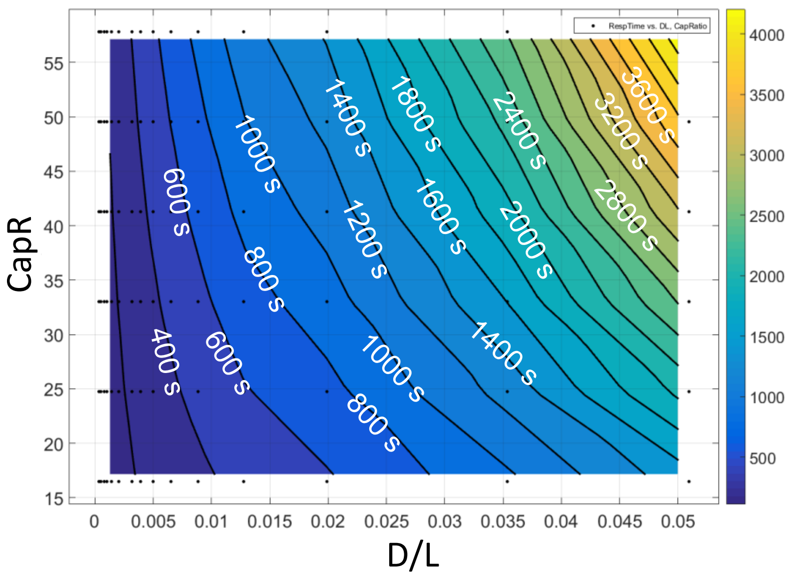
**HEX Wall material**




**HEX Geometry**

**Ja = 0.870**

**HEX Wall material**




**HEX Geometry**

**Fixed parameters:**

Area HT = 4m$^2$

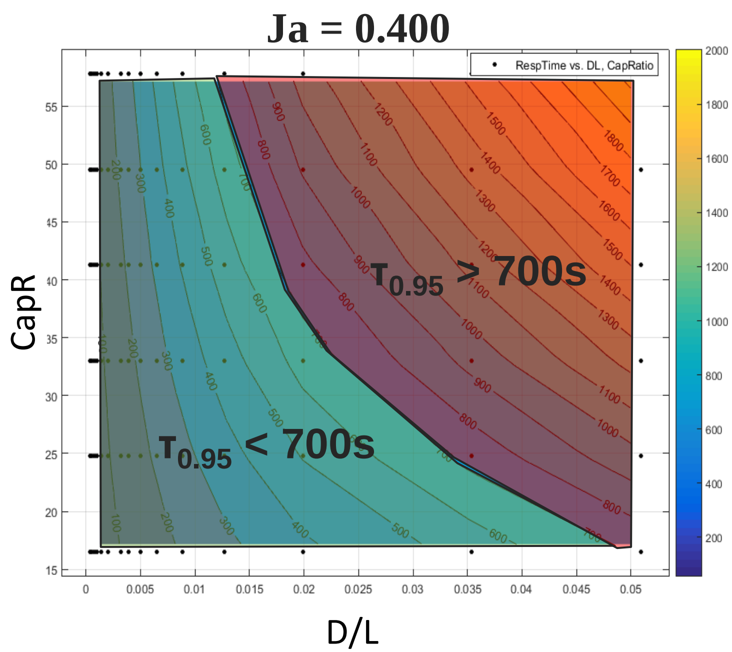
Wall-th = 2mm

q = 10 kW/m$^2$

Tsat = 450 K

Fluid: MM

# ORC evaporator response time charts




If we are interested in a response time slower or faster than 700 s, which area will that be?

**The charts show “what it takes” in terms of design to achieve a desired dynamic response**

# Dynamic regimes and dynamic regime number

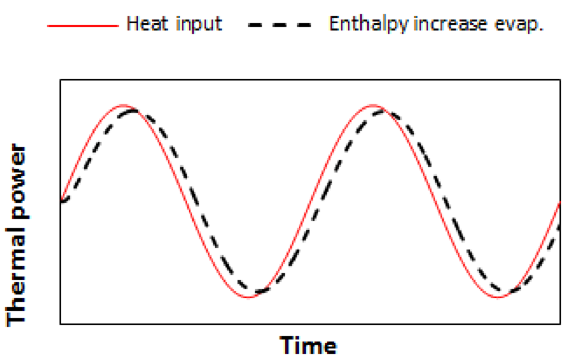
Define a characteristic response time of the evaporator $\tau_{ev}$

**Dynamic regime number** $\Gamma$**:** ratio of response time to period of load fluctuation

$$\Gamma = \frac{\tau_{ev}}{T_{load}}$$

We know what it takes to design the evaporator for a certain thermal inertia

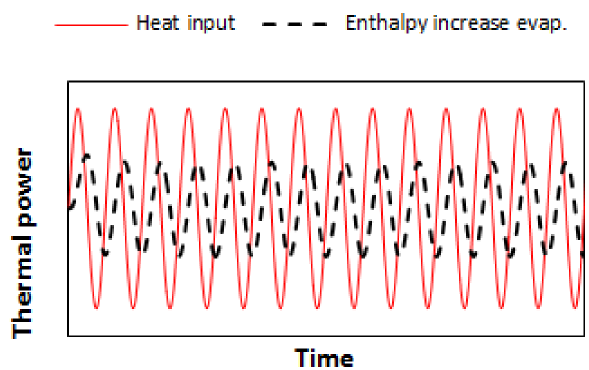
To have a desired dynamic regime behavior with a given fluctuating heat input




**Dynamic regime I**

*Quasi-steady*

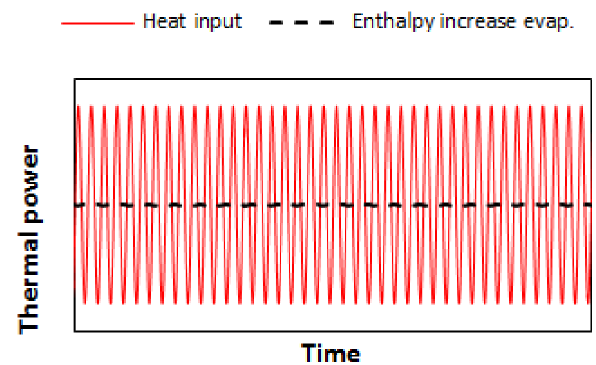
$\Gamma < 1$




**Dynamic regime II**

*Transient*

$1 < \Gamma < 10$

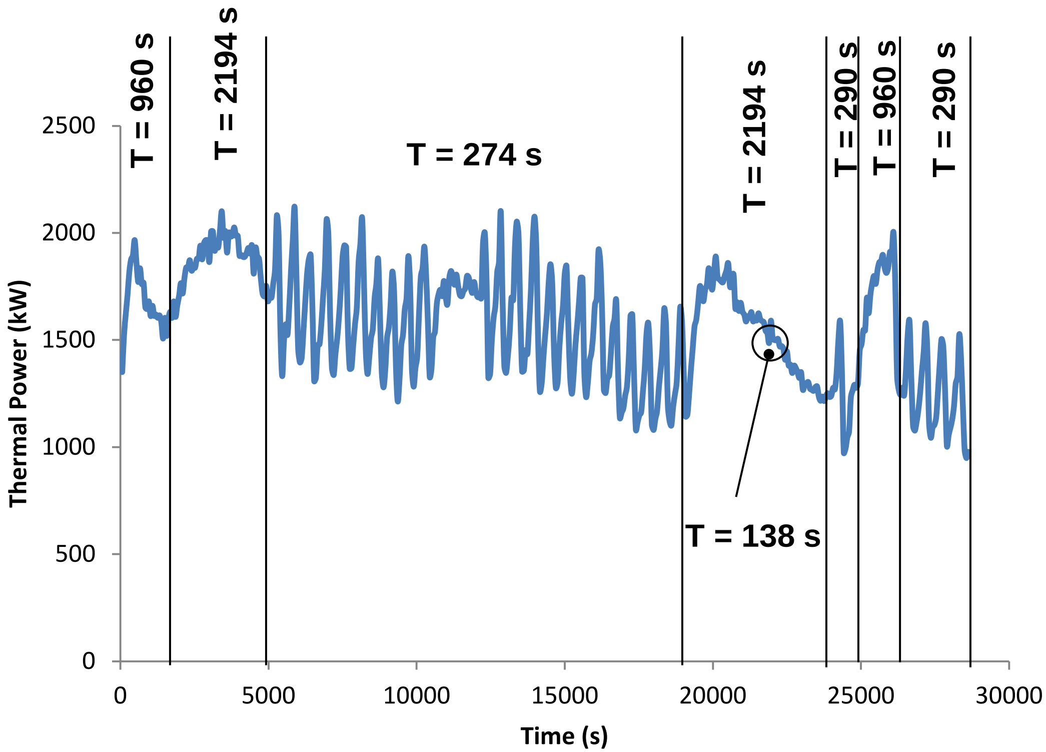



**Dynamic regime III**

*Quasi-constant*

$\Gamma > 10$

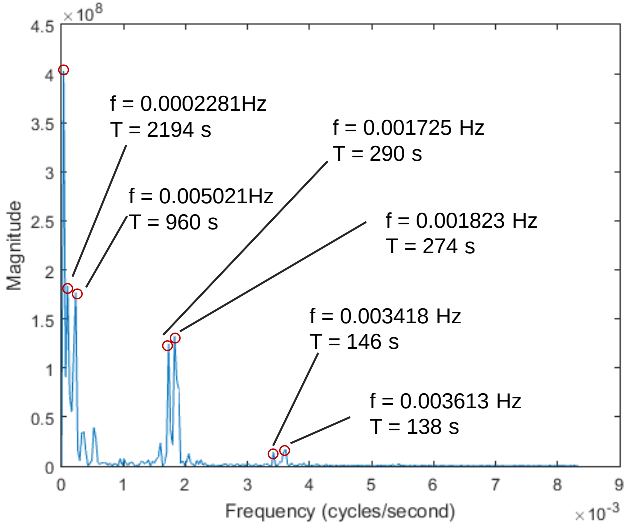
# Case study – Billet reheating furnace waste heat




**Discrete Fourier Analysis**

**Frequency components**

*T: characteristic period of load (half-wave)*
*f: frequency component (full-wave)*



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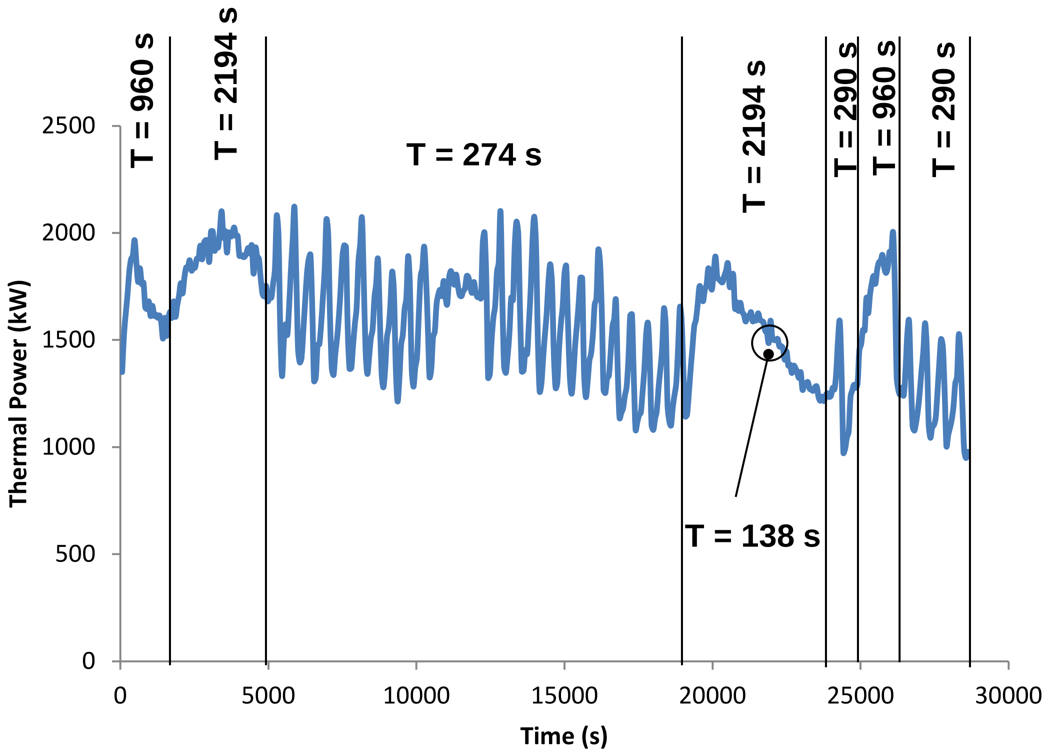
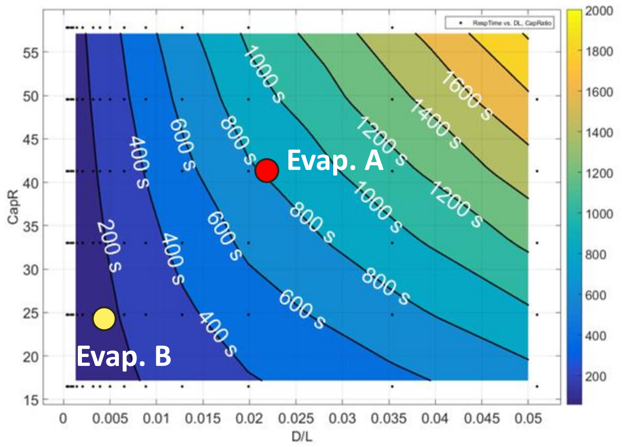



Evaporator A – medium inertia

Evaporator B – low inertia



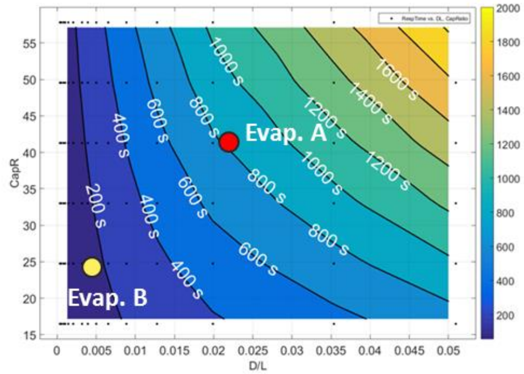

We can choose a combination of design variables for a desired dynamic regime depending on our thermal power profile

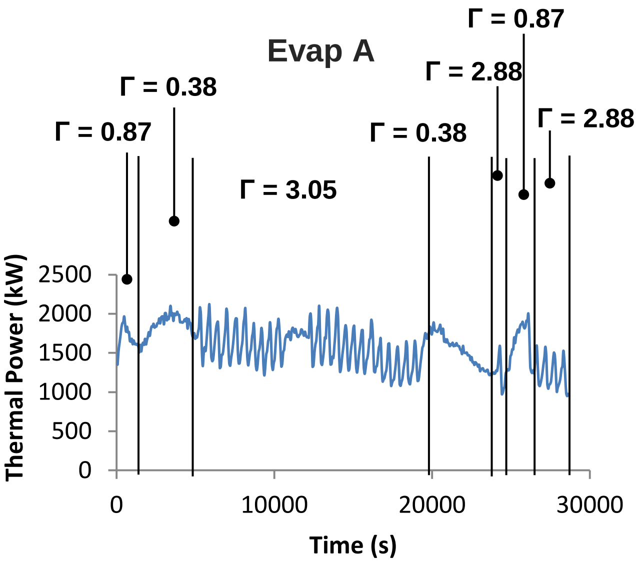
$$\Gamma = \frac{\tau_{0.95}}{T}$$

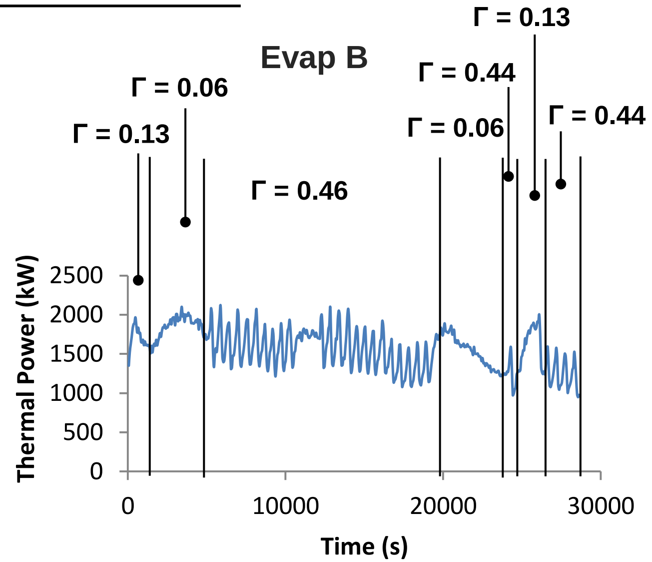
# Case study – Billet reheating furnace waste heat

$$\Gamma = \frac{\tau_{0.95}}{T}$$

| | Evaporator A | Evaporator B |
|---|---|---|
| **Charact. time $\tau_{0.95}$** | **834.9 s** | **126.4 s** |
| Fluid | MM | MM |
| Heat transfer area | 4 $m^2$ | 4 $m^2$ |
| **D/L** | **0.0199** | **0.0022** |
| $Ja_{lv}$ | 0.400 | 0.400 |
| CapR | 41.28 | 24.92 |
| **Wall material** | **Steel** | **Aluminum** |
| Wall thickness | 2 mm | 2 mm |
| Fluid mass flow | 0.25 kg/s | 0.25 kg/s |
| Sat temperature | 177 °C | 177 °C |
| # of parallel tubes | 25 | 25 |



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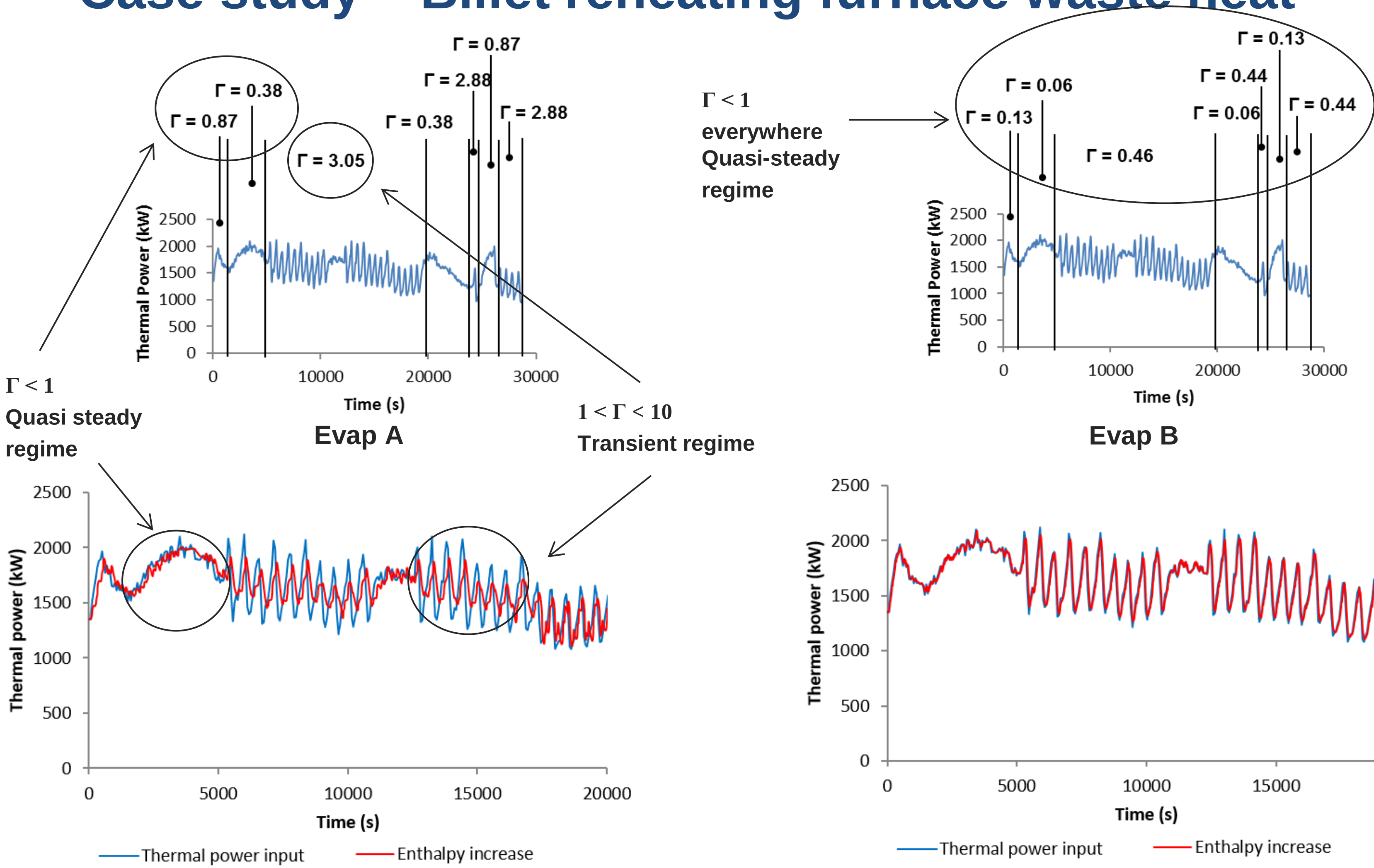
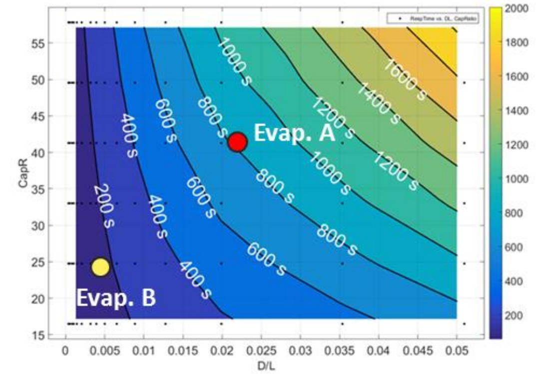
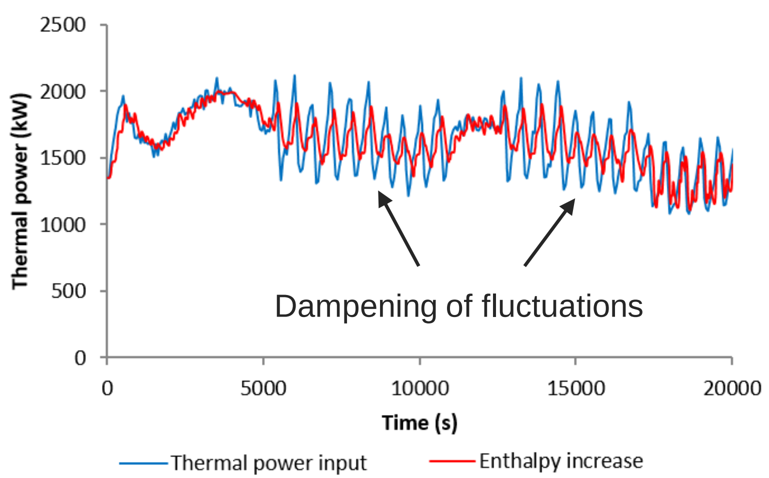


# Case study – Billet reheating furnace waste heat

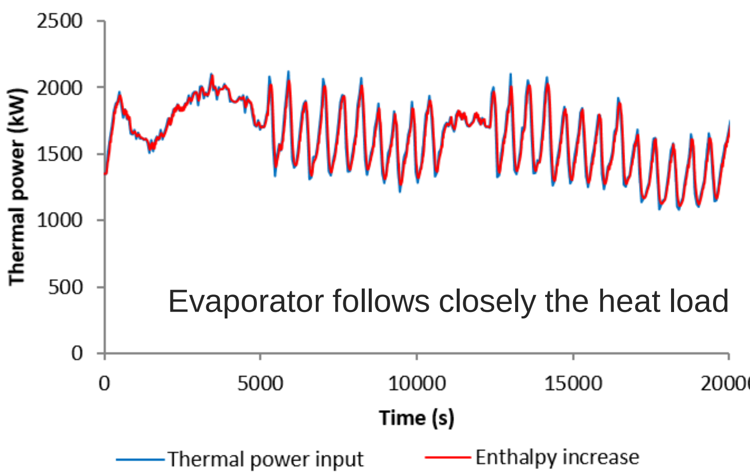
- Evaporator A can effectively filter out some of the variability of the heat
- Less deviation from a design point - "Thermal flywheel"
- Evaporator B reacts faster to changes



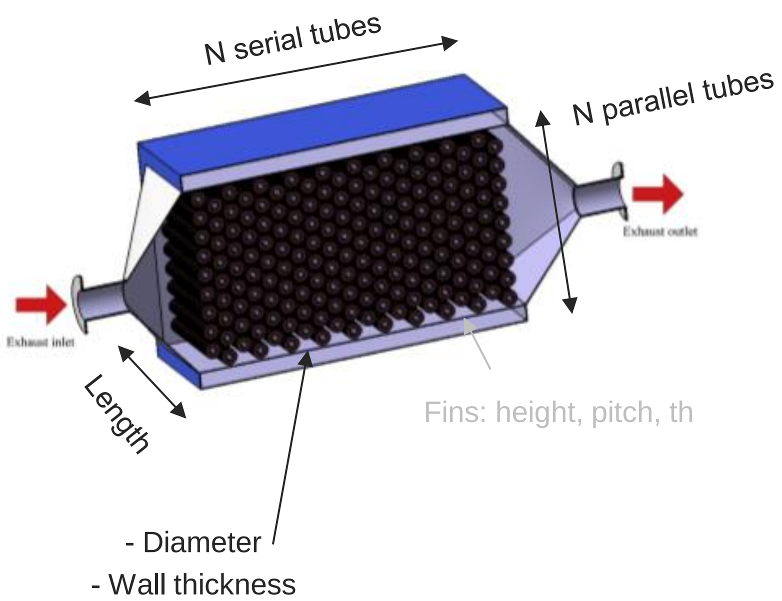

Evap A



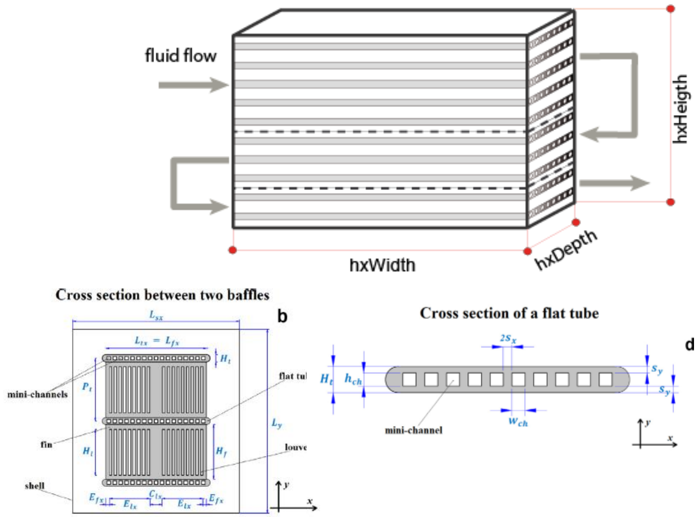

Evap B



# Current and future work




VS




Consideration of secondary fluid cooling (properties of air)

Step changes of mass flow/temperature

Fig. Finned Tubes (Yang, 2015)

Fig. Louvered Fins (Mastrullo, 2015)

# Concluding remarks

- Methodolgy to include dynamic behavior of ORC evaporator at design stage
- Response time charts as function of design decision variables: geometry, wall material, fluid
- Case study: evaporator selection that can reduce variability of heat
- Very simple geometry – method is to be extended to more realistic and complex geometries

Applications:

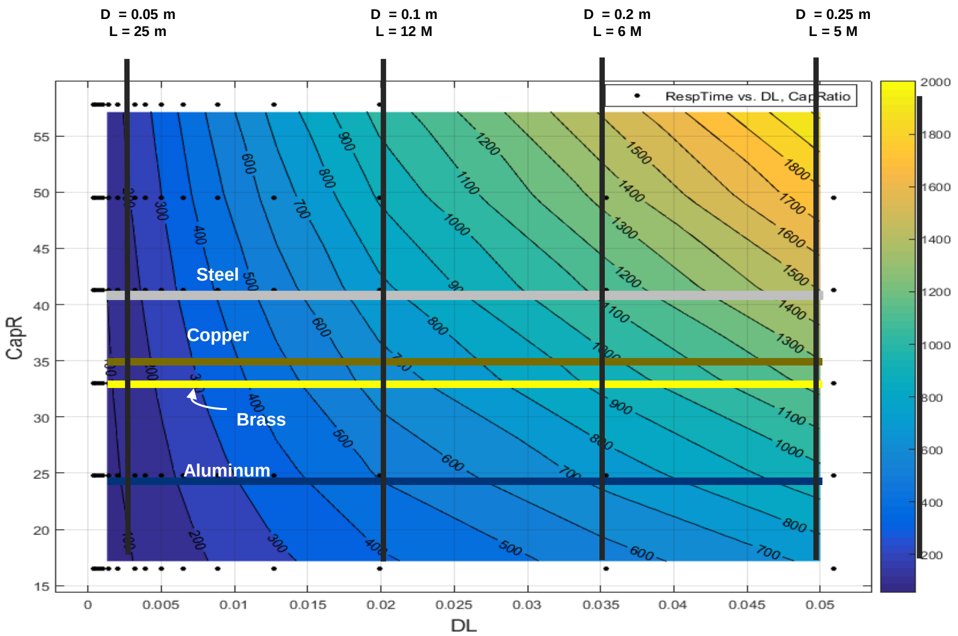
- Dampening to decrease inefficiencies of ORC related to off-design conditions
- Feasability of direct evaporation (no thermal oil loop) to reduce size of system on mobile applications
- Design “desired” dynamic behavior of ORC for control

# Thank you for your attention!

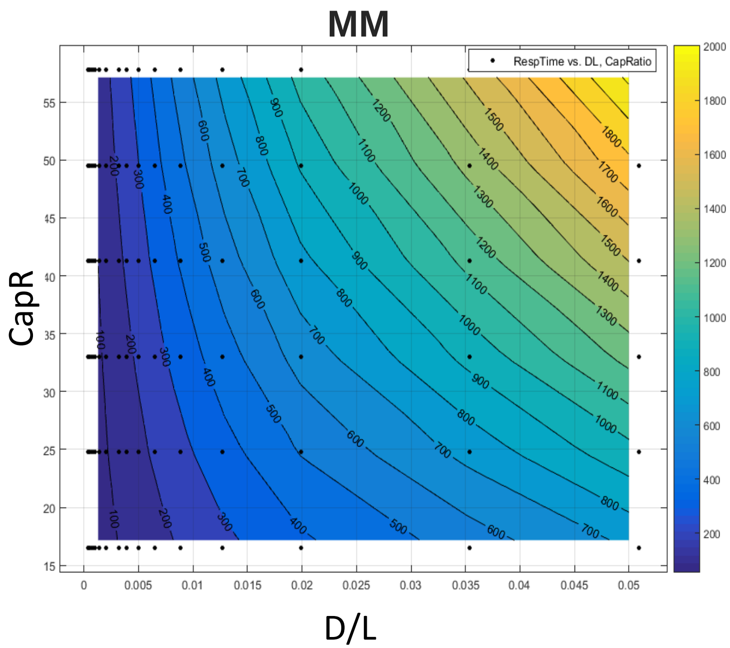
# Q & A

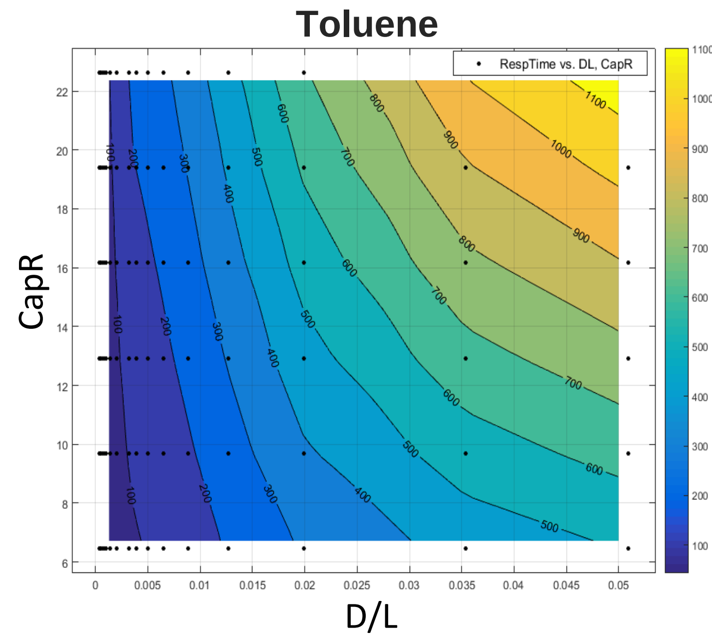
**Acknowledgements:** This research is part of the ICER collaborative project between NTU Singapore and TUM Germany

# Response Time charts



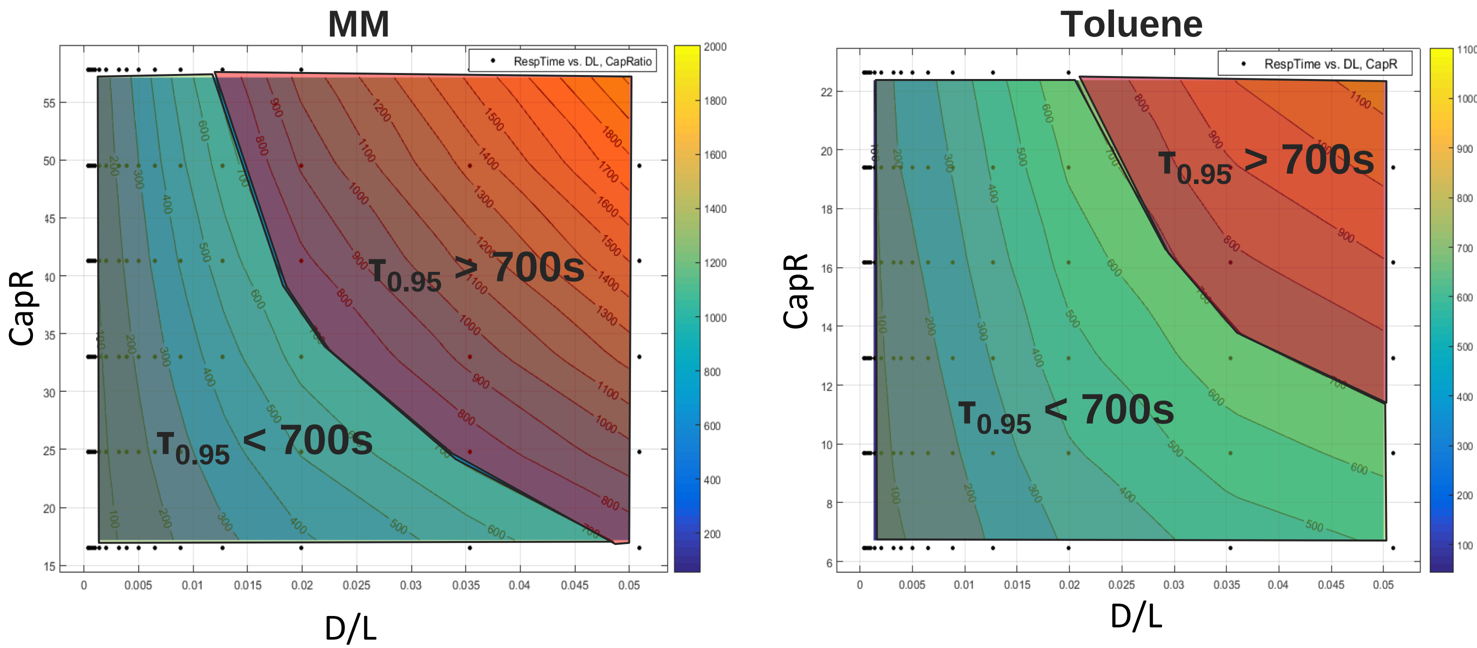
# ORC evaporator response time charts




Comparison of resp times for two dif. Fluids with the same fixed parameters. (area, heat flux) → Toluene shows relatively shorter response times

# ORC evaporator response time charts

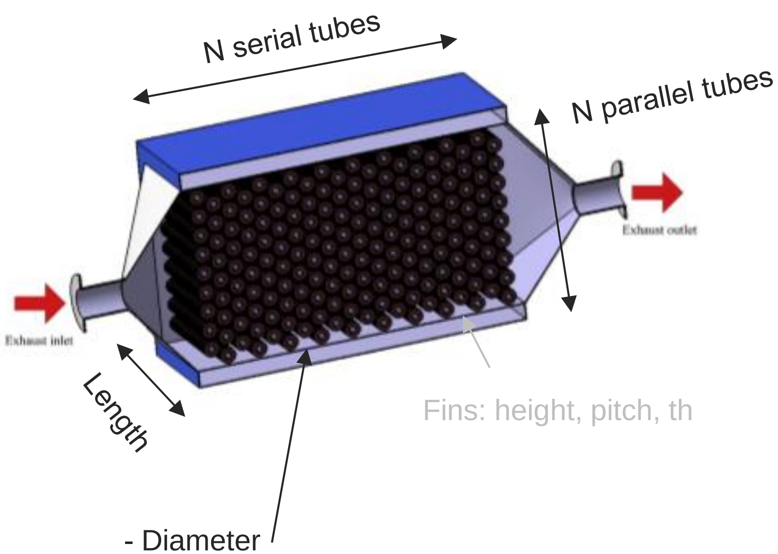



Comparison of resp times for two dif. Fluids with the same fixed parameters. (area, heat flux) → Toluene shows relatively shorter response times

If we are interested in a response time slower or faster than 700 s, which are will that be?

The charts show “what it takes” in terms of design to achieve a desired dynamic response

# Extension to more complex geometries: Finned tubes evaporator




**Key parameter to vary: diameter of tubes**

Investigation of response time, fixed UA, variable D (int)

Case 1: Variable diameter and N serial tubes

Case 2: Variable diameter and N parallel tubes

Case 3: Variable diameter and length

Case 4: "Fixed" int. diameter, variable thickness

Base geometry (Yang, 2015)

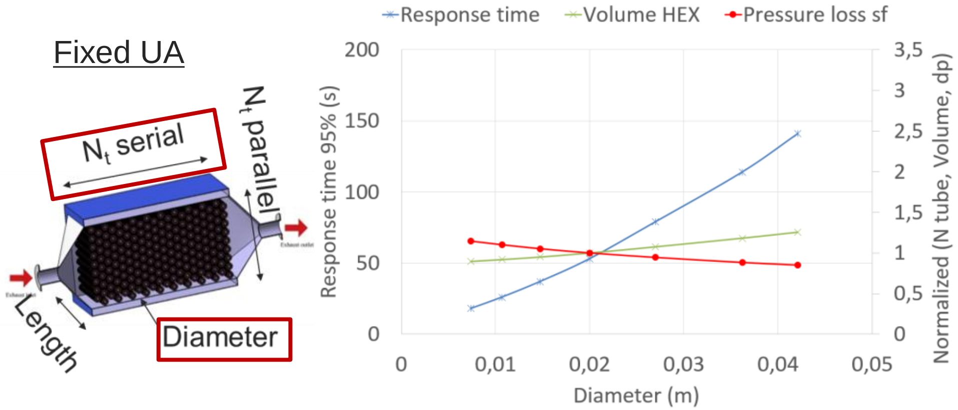
| | |
|---|---|
| N serial tubes | 9 |
| N parallel tubes | 20 |
| Tube length | 0.8 m |
| Diameter (int) | 20 mm |
| Wall thickness | 2.5 mm |

Base thermal boundaries

| | |
|---|---|
| Mass flow flue gas | 3 kg/s |
| Temperature flue gas | 350 ° C |
| Inlet sub-cooling | 10 ° C |
| Outlet superheating | 1 ° C |
| Working fluid | R245fa |
| Evap pressure | 30 bar |

Same UA means same amount of heat is being transferred for the same inlet conditions of both fluids

# Extension to more complex geometries: Finned tubes evaporator

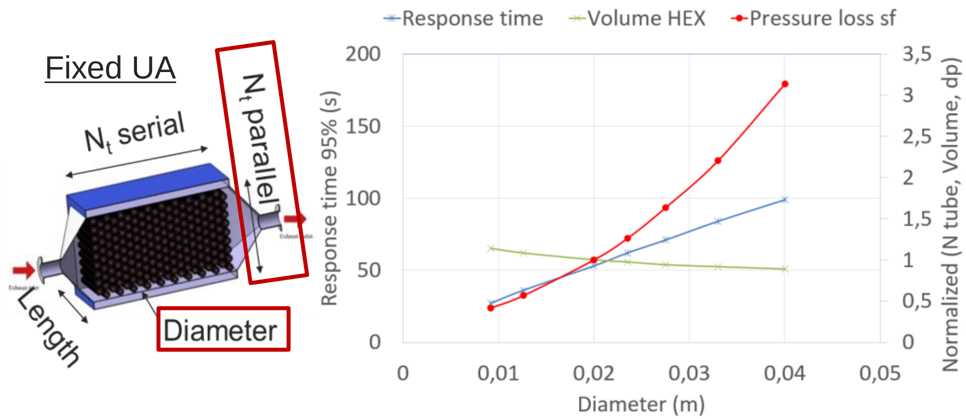


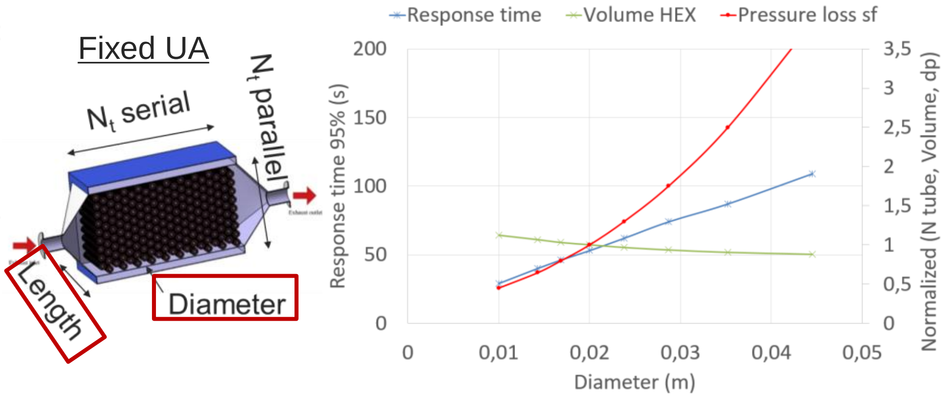

UA determined by thermal req.

Important quantities for design:

- Response time
- Pressure losses
- Volume and weight

From this data non-dimensional charts including more geometrical parameters can be built as before




*response time to step changes in mass flow of hot fluid

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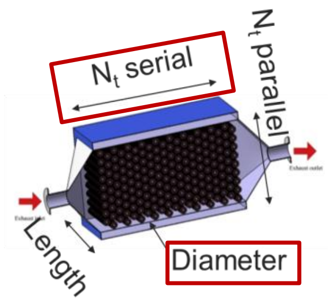
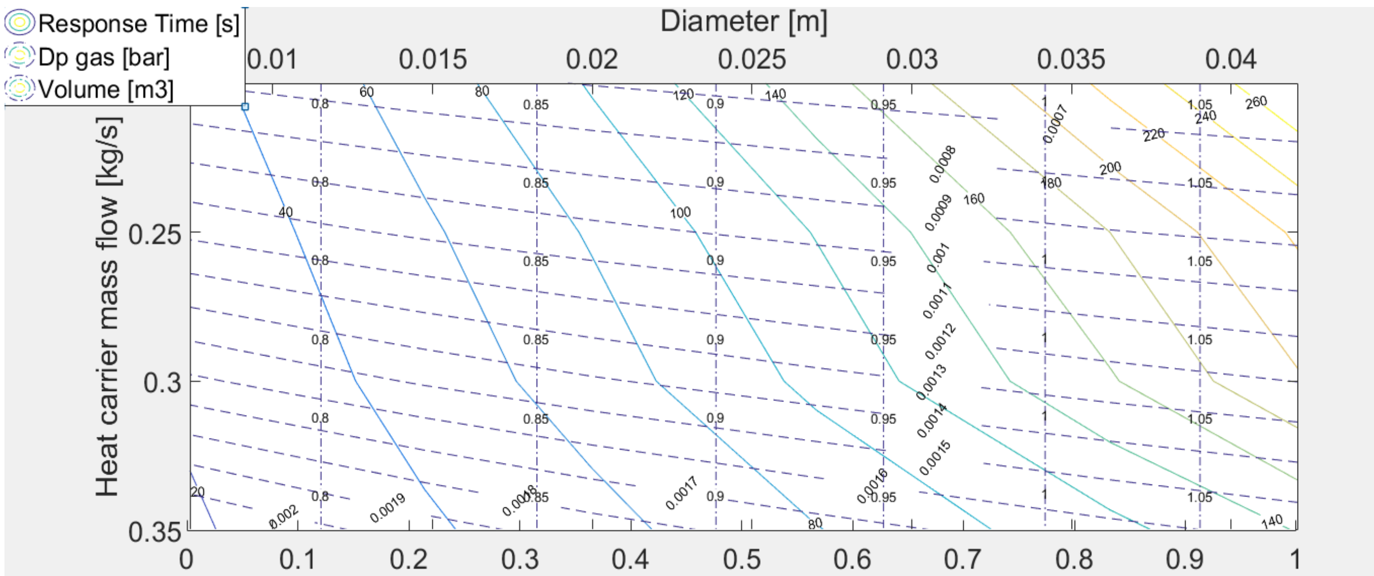
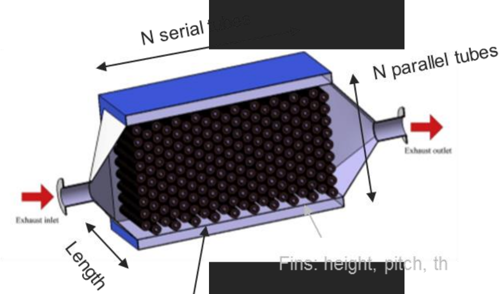


# Next: con[redacted]on of termal inertia with louvered fins heat [redacted]nger

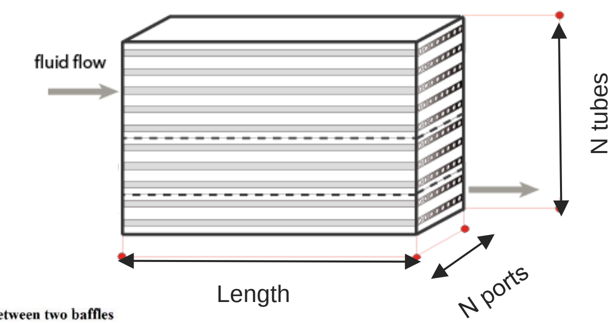
Finned tu[redacted]
Base geo[redacted]15

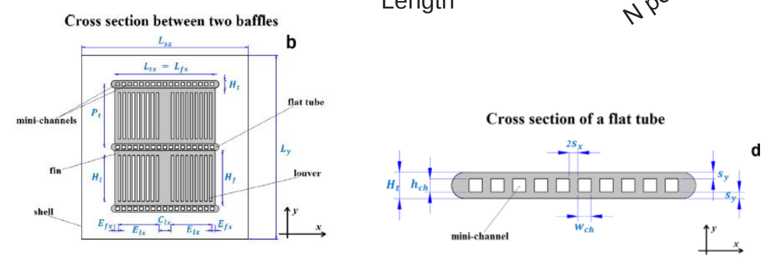



VS

Comparison variables:
N parallel tubes – N tubes
N serial tubes – Length
Length – N ports
Thickness - Thickness

Louvered fins ORC evap
Base geometry - Mastrullo, 2015




Compare to th[redacted]y conditions

- Conven[redacted]s take practically the same values of mass flow and temperature of the heat source
- How to [redacted]ries?
  - Hyd. Diameter - very different ranges, characteristics
  - Equivalent “macro” dimensions: <u>Aspect ratio of gas side path</u>

# Louvered fins evaporator – dynamic characterization with changes in geometry and step change in gas mass flow