COBEM 2025
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Flow Boiling Heat Transfer Under High Reduced Pressures: Applications, Experimental Data, and a New Mechanistic Model

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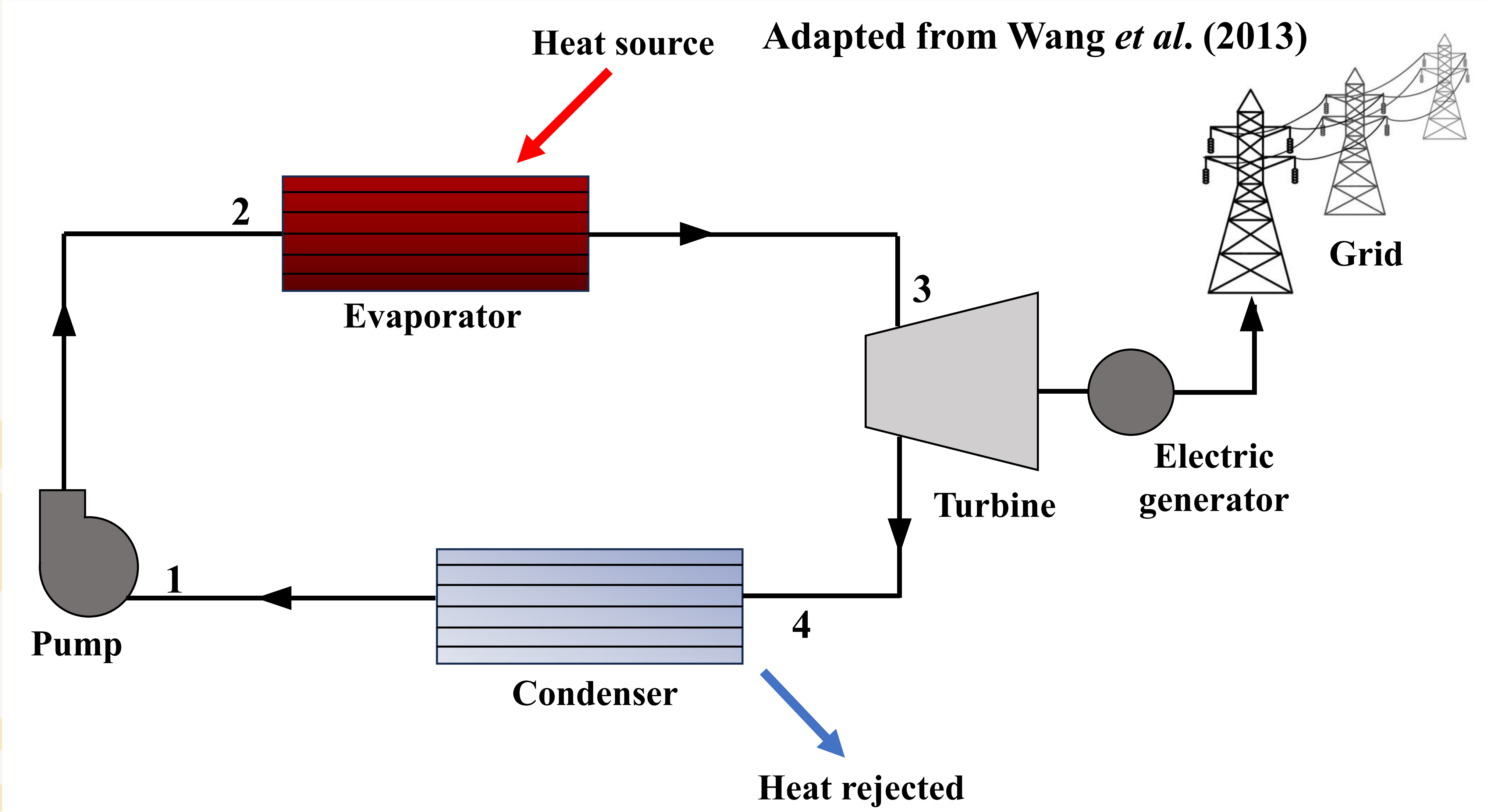
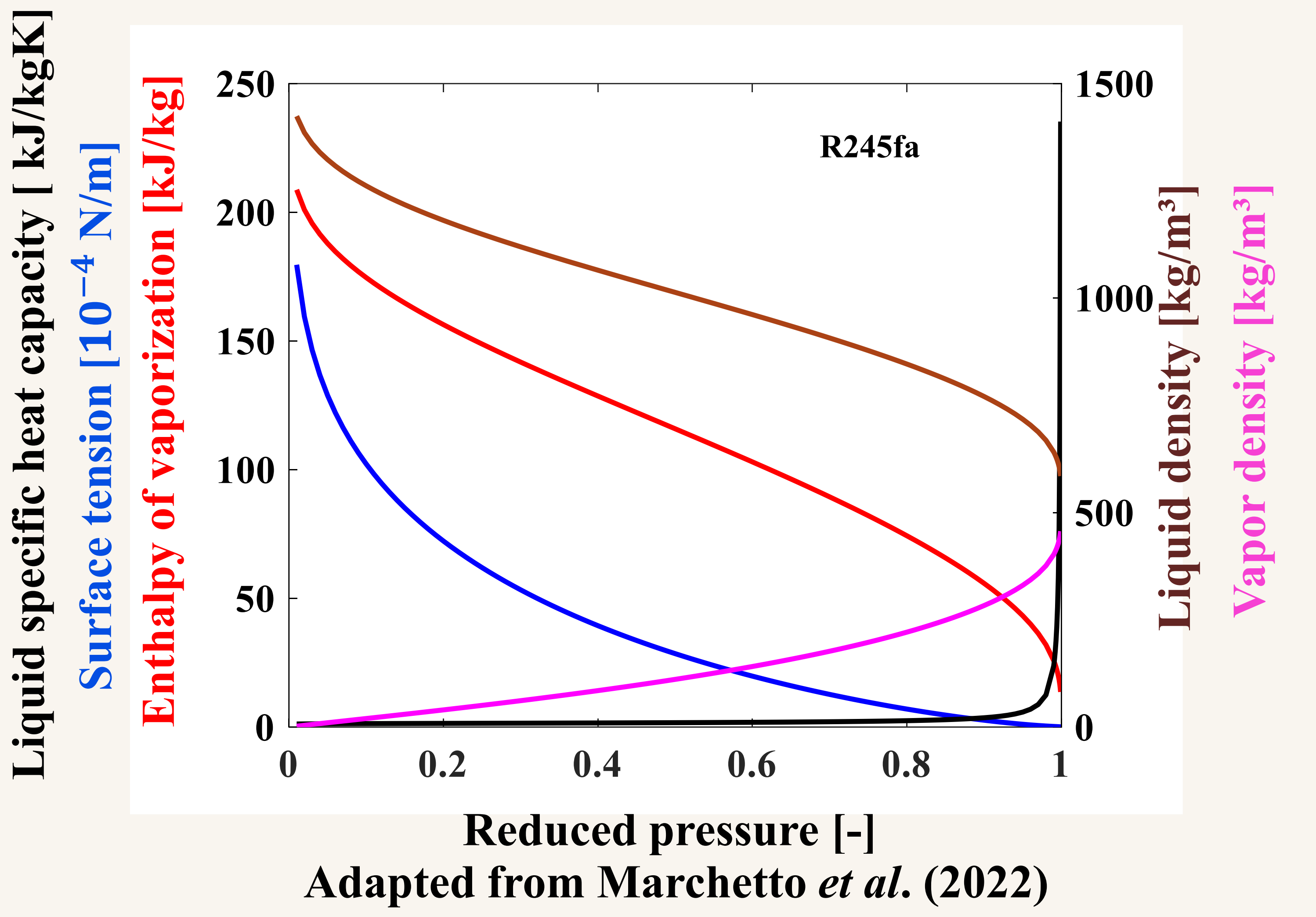
Agenda

1. Applications
2. Summary of literature review
3. Objectives
4. Experimental apparatus
5. Data regression and validation
6. Results
7. Heat transfer modeling
8. Conclusions

Applications



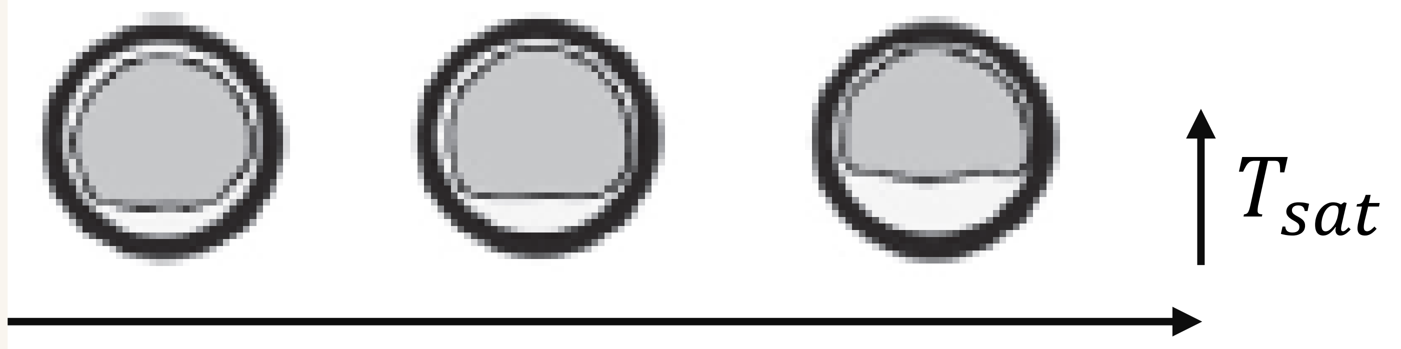


- Refrigeration and air-conditioning motivated most of the flow-boiling studies the last decades;
- Growing interest in high-temperature applications: Organic Rankine Cycles (ORCs) and high-temperature heat pumps (HTHPs);
- Thermophysical properties variation with increasing p_r and T_r;
- These variations in thermophysical properties have a pronounced impact on flow dynamics and heat transfer throughout the evaporation process;
- Prediction methods developed for low saturation temperatures are still valid at high T_{sat} ?

Status Quo

Two phase flow patterns

- Enhanced assymetry at high T_{sat}
- Smaller vapor bubble diameters at high T_{sat}
- Anticipation of I-A and A-D transitions
- Low accuracy of transition criteria



Adapted from Charnay *et al.* (2013)

Pressure drop

- Reduction of the frictional pressure gradient at high T_{sat}
- No clear influence of p_r on prediction methods accuracy was identified (1729 data from 15 studies analyzed)
- The methods evaluated fail to predict the vapor quality corresponding to maximum pressure gradient

Heat transfer coefficient

- Different HTC x vapor quality trends identified, depending on the dominant heat transfer mechanism
- The effect of operational parameters depends on dominant mechanism. In general, $\uparrow T_{sat}$ $\uparrow$ HTC
- Discrepancies under similar experimental conditions
- Higher deviations of prediction methods near to critical point (6257 data from 45 studies analyzed)

Status Quo

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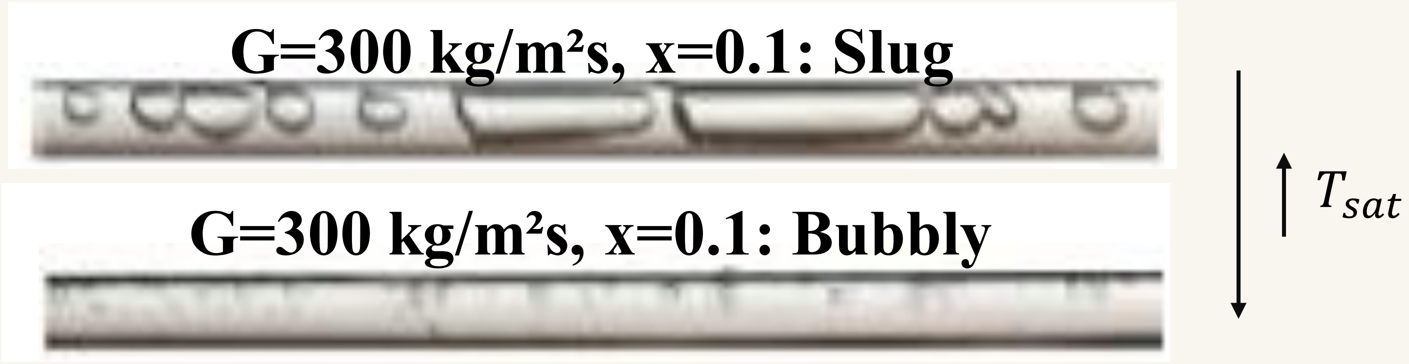

Adapted from Ami *et al.* (2010)

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Status Quo

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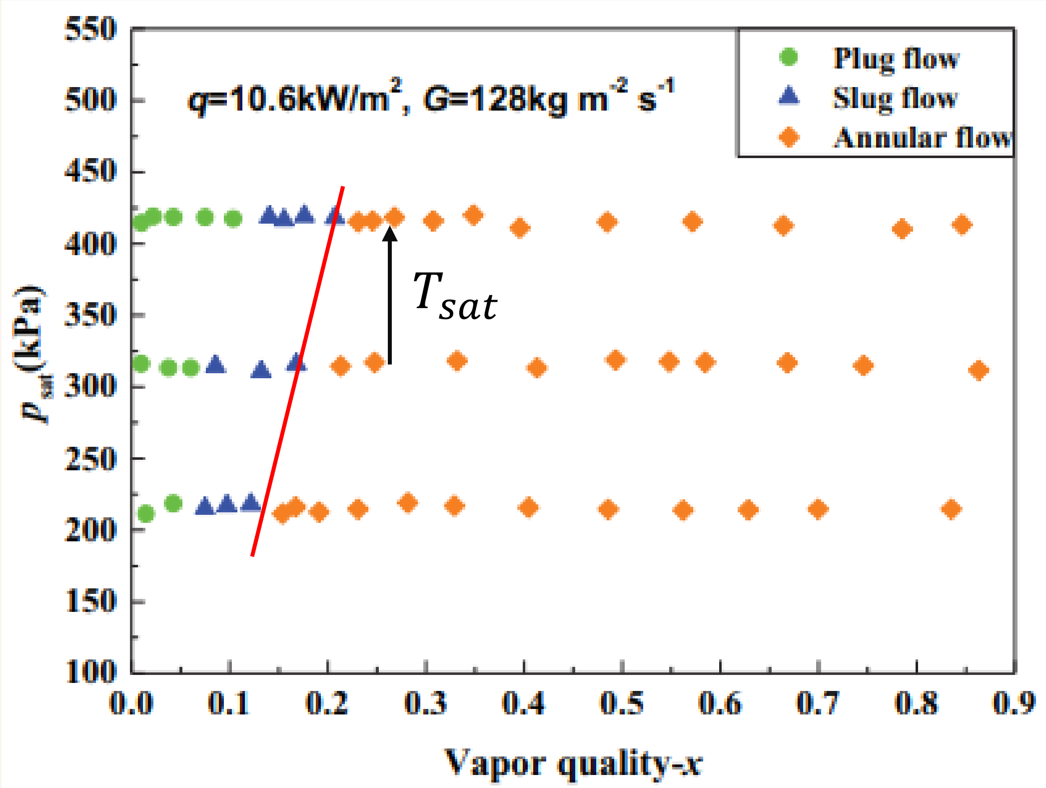
Adapted from
Yang *et al.* (2019)



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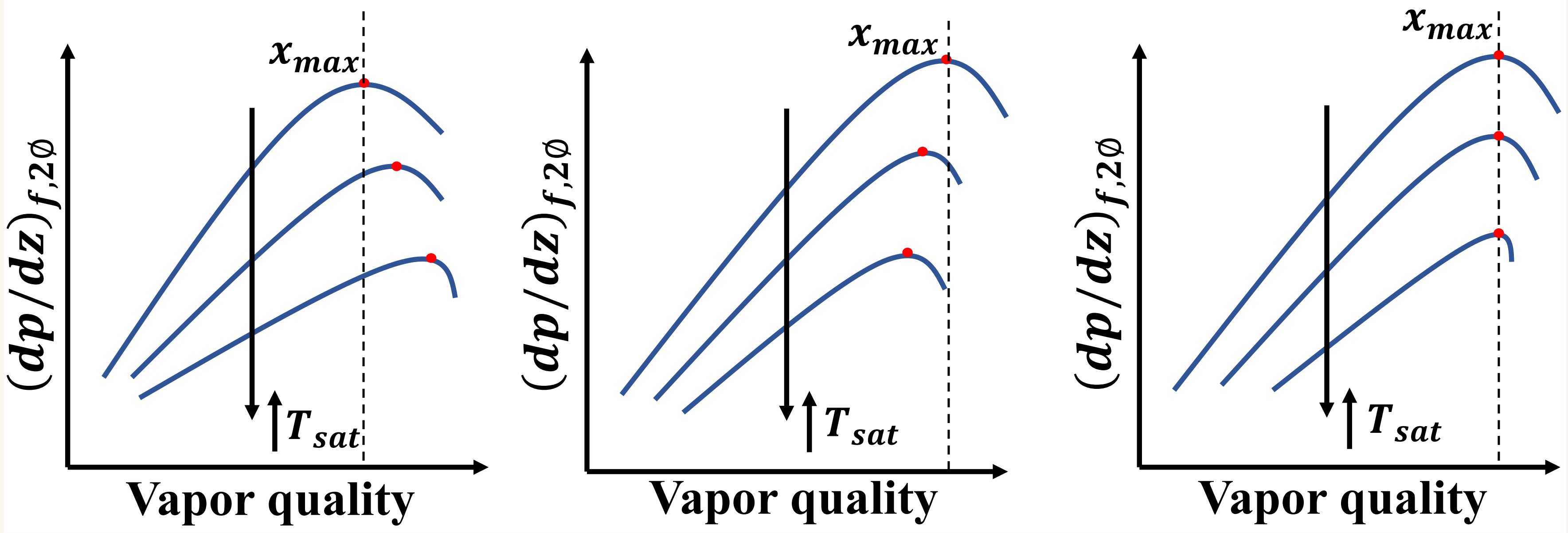

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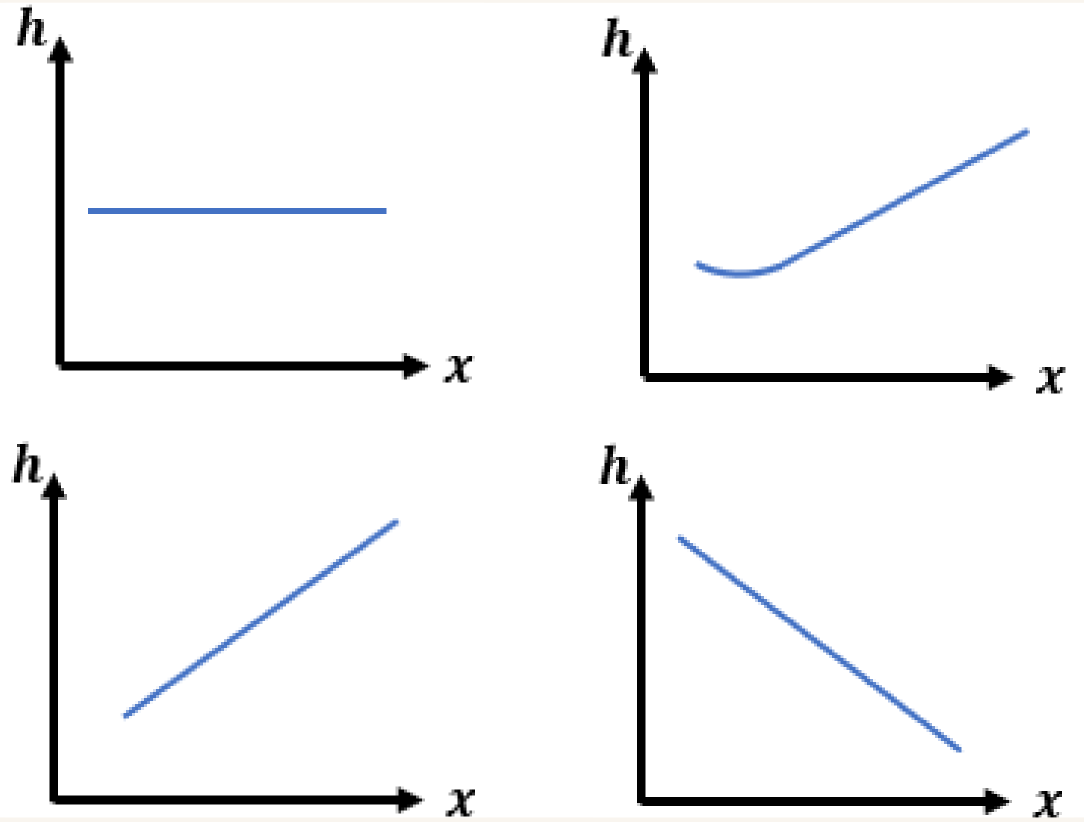

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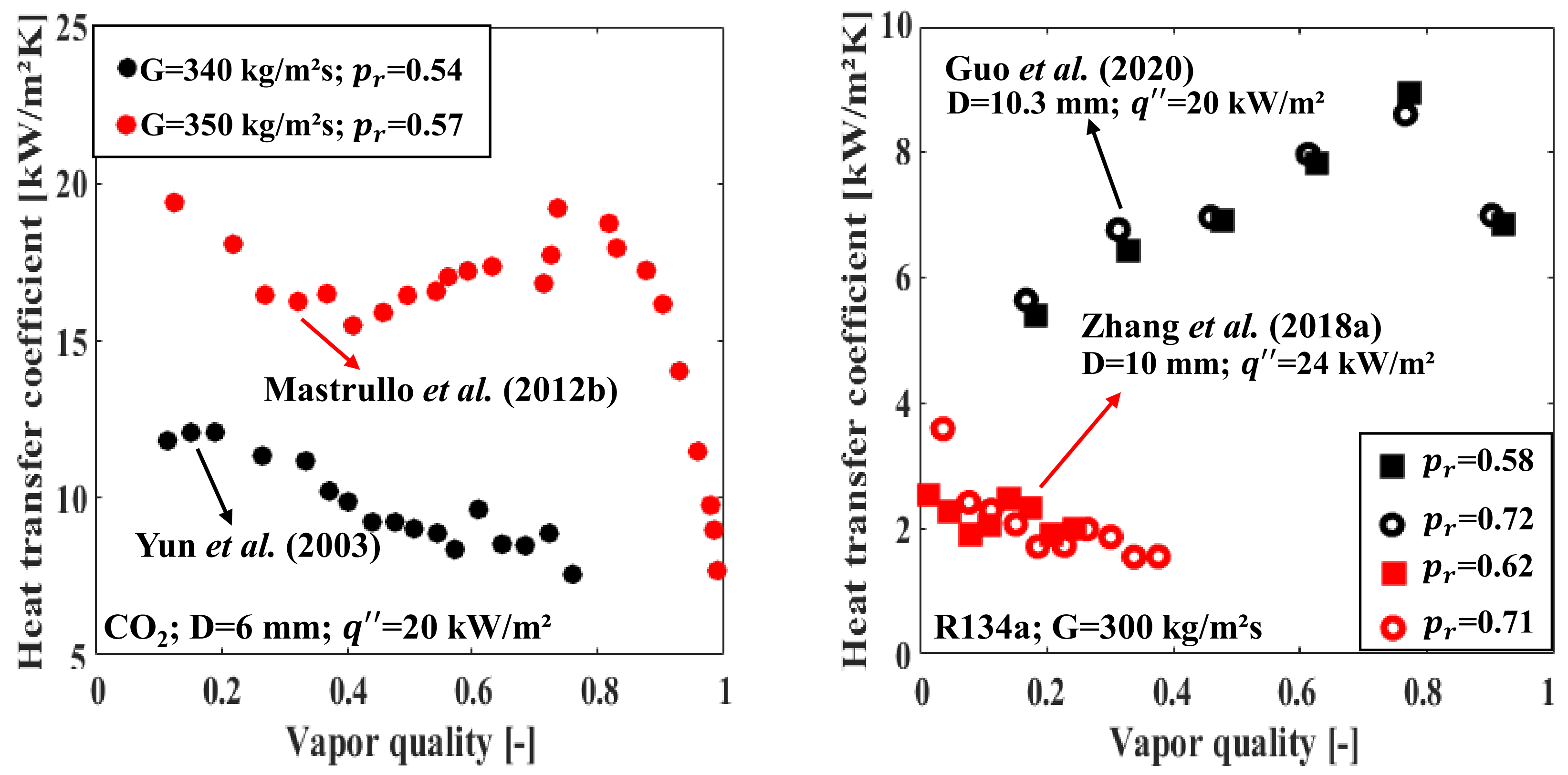

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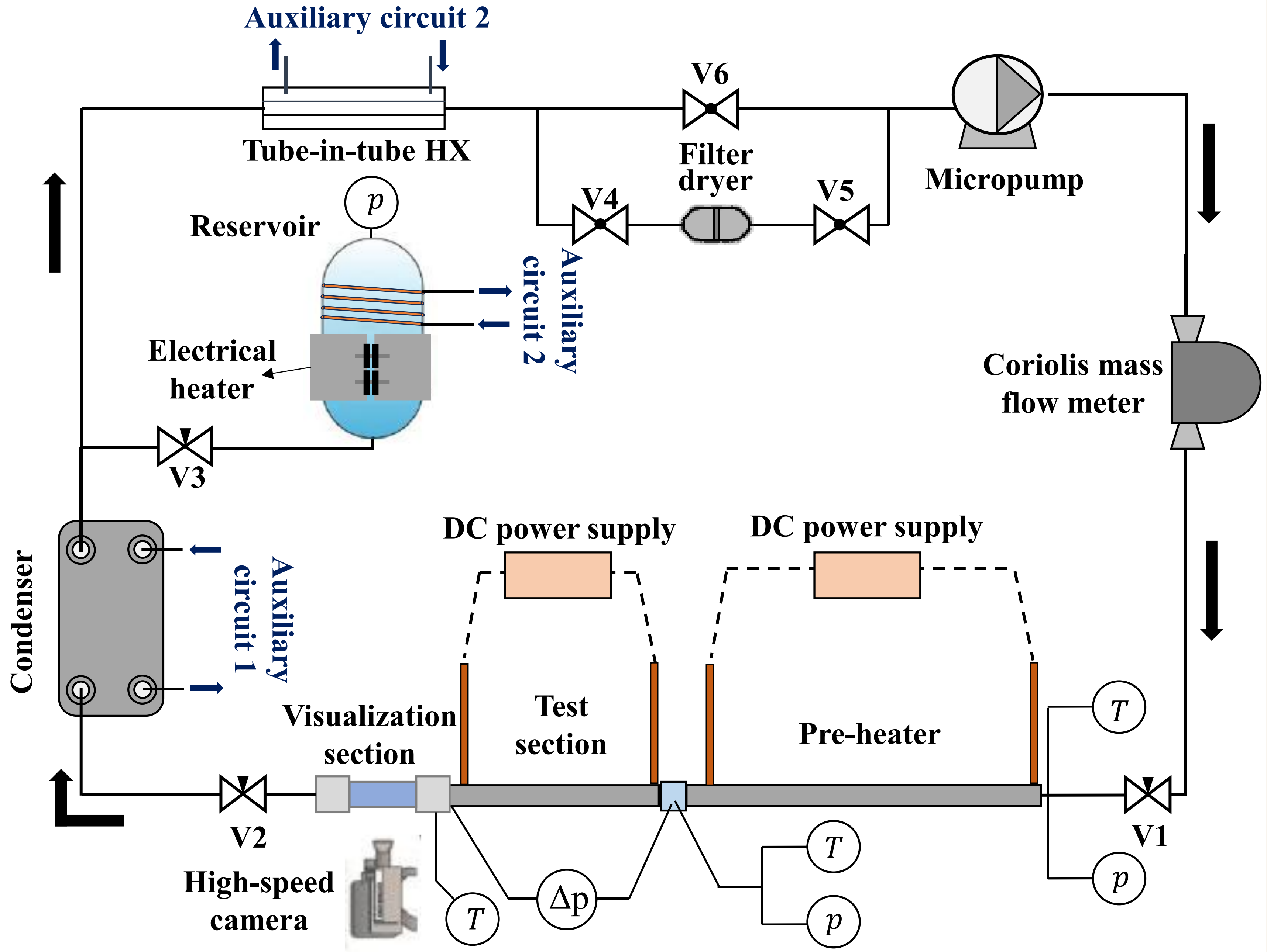
Status Quo

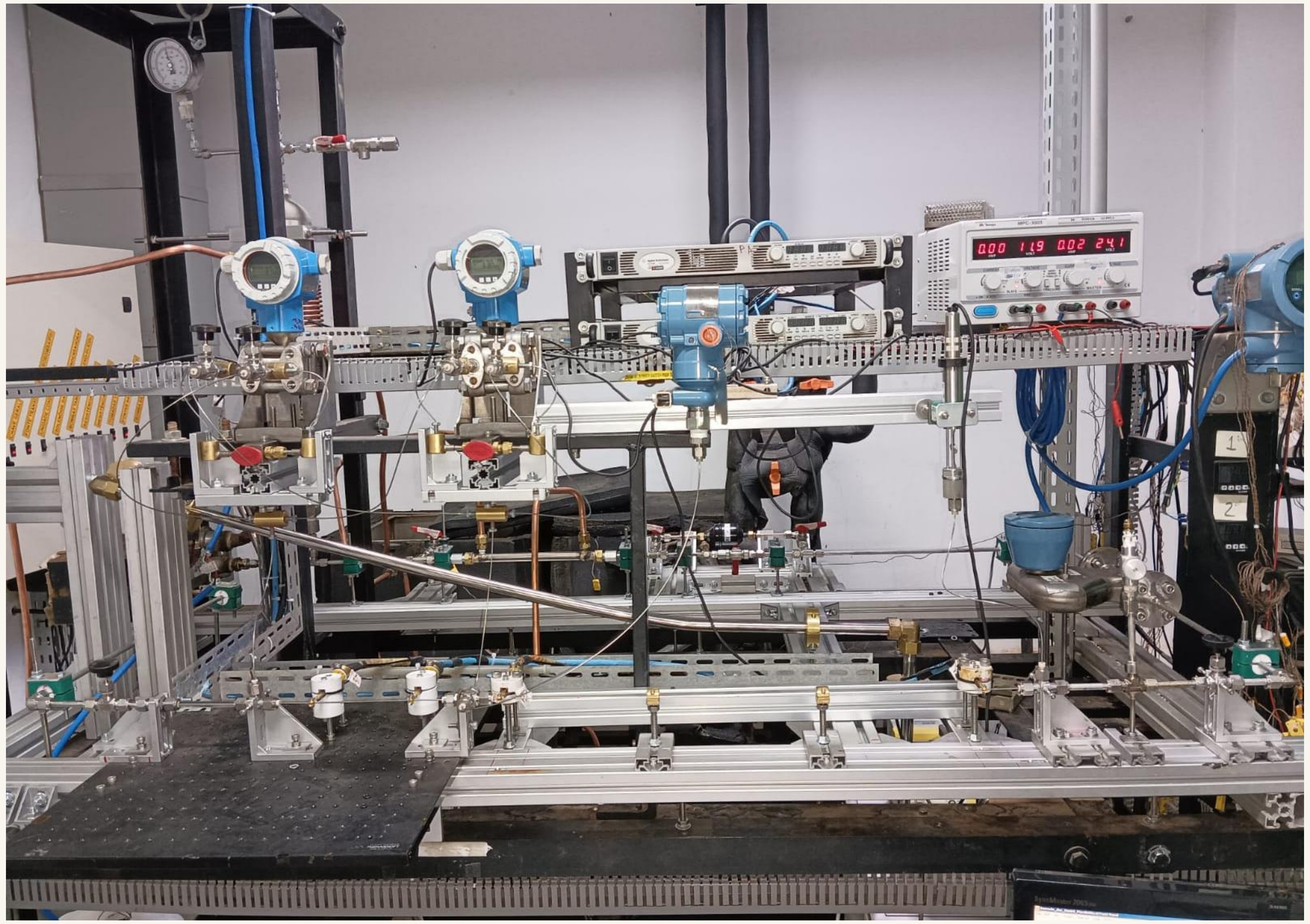
General Overview

- Despite the growing interest in ORCs and HTHPs, experimental data at high T_{sat} (50-150°C) are still scarce
  - → Most studies performed in conventional diameter channels
  - → Limited experimental conditions
  - → Shortage of experimental data for low-GWP refrigerants
- R245fa: High-performance working fluid for ORCs (He *et al*., 2012; Su *et al*., 2017 Babatunde *et al*., 2018)
- Low-GWP alternatives: R1233zd(E), R1336mzz(Z)

Experimental Apparatus

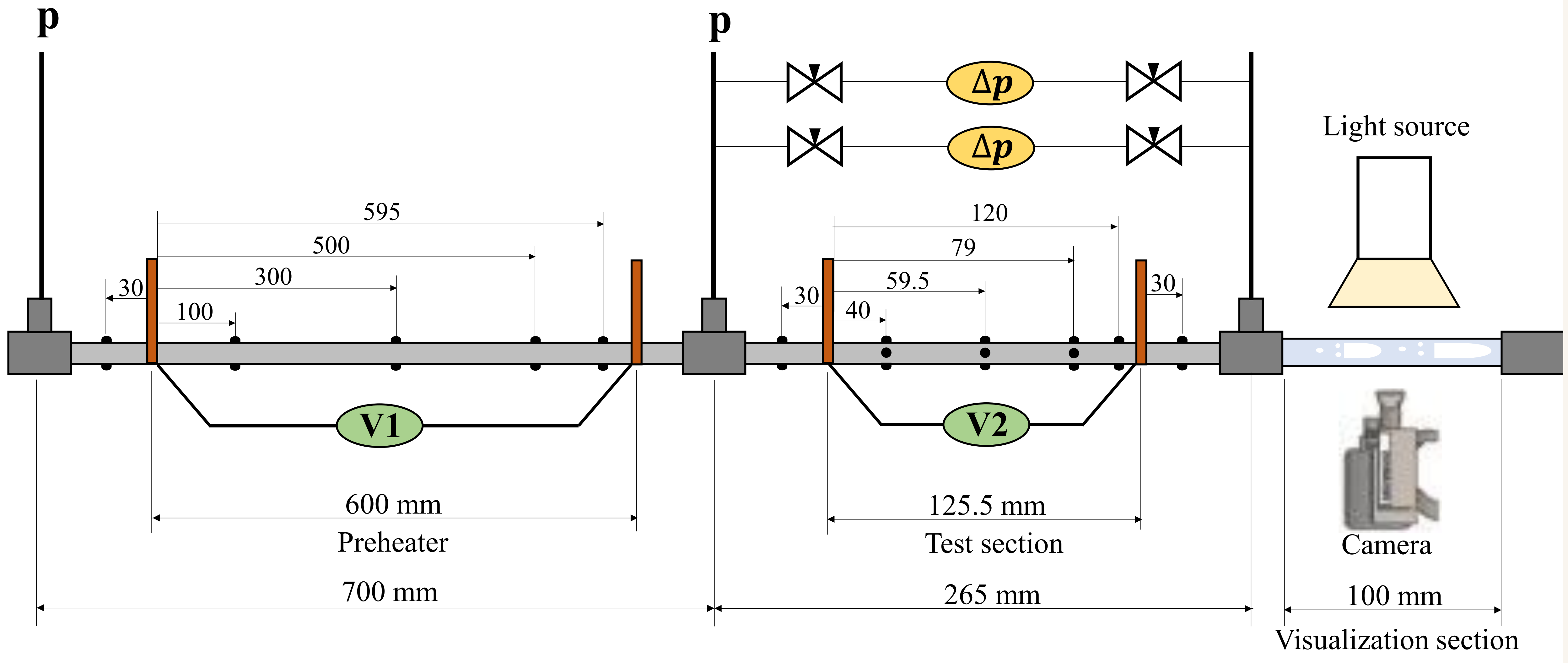
LETeF flow loop



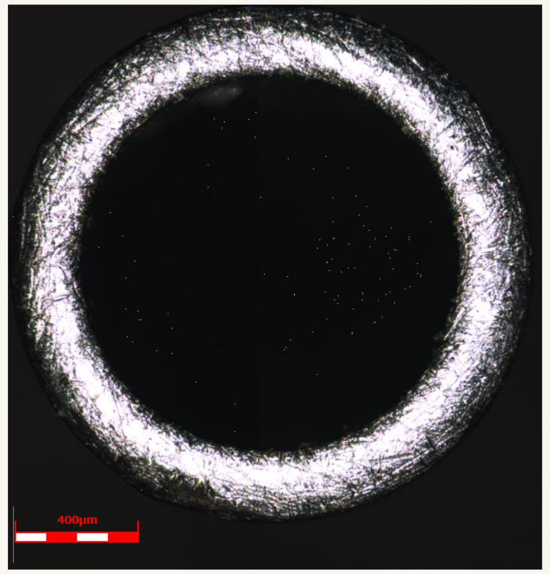


Experimental Apparatus

LETeF preheater, test and visualization sections



Tube diameter

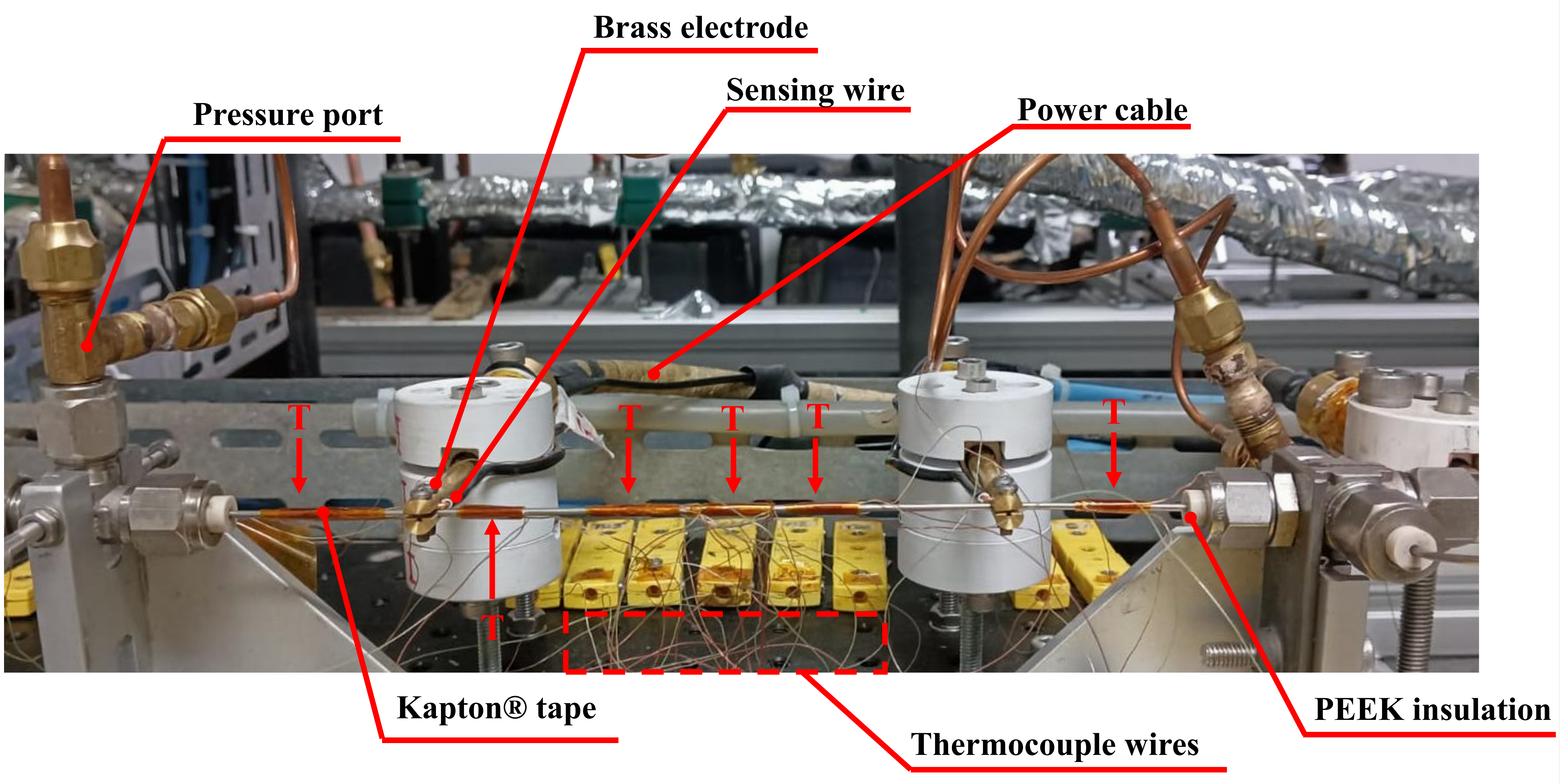


MAIN DIMENSIONS

Preheater: 1.69 mm (OD), 1.22 mm (ID)
Test section: 1.69 mm (OD), 1.22 mm (ID)
Visualization section: 8.00 mm (OD), 1.2 mm (ID)

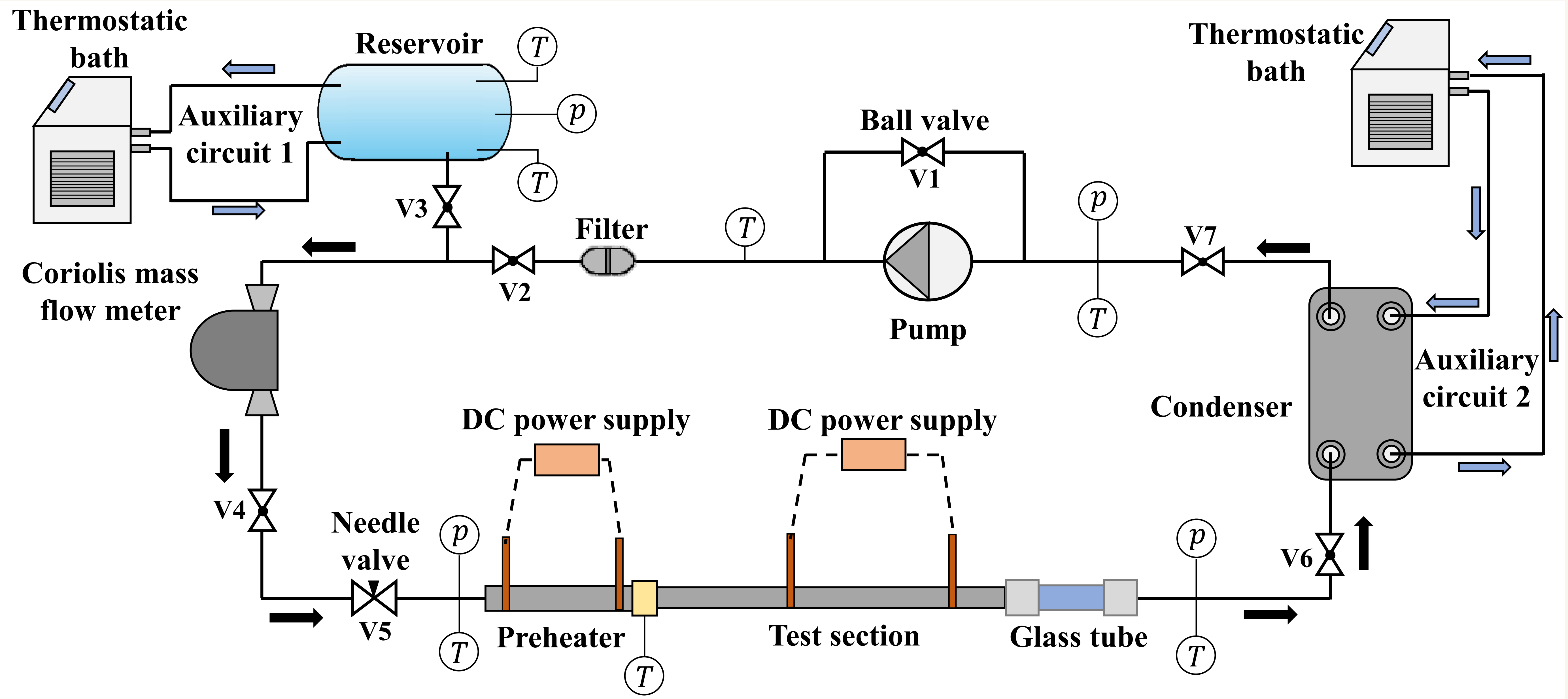
Experimental Apparatus

Test section and preheater details



Experimental Apparatus

CETHIL INSA-Lyon flow loop



Data regression and validation

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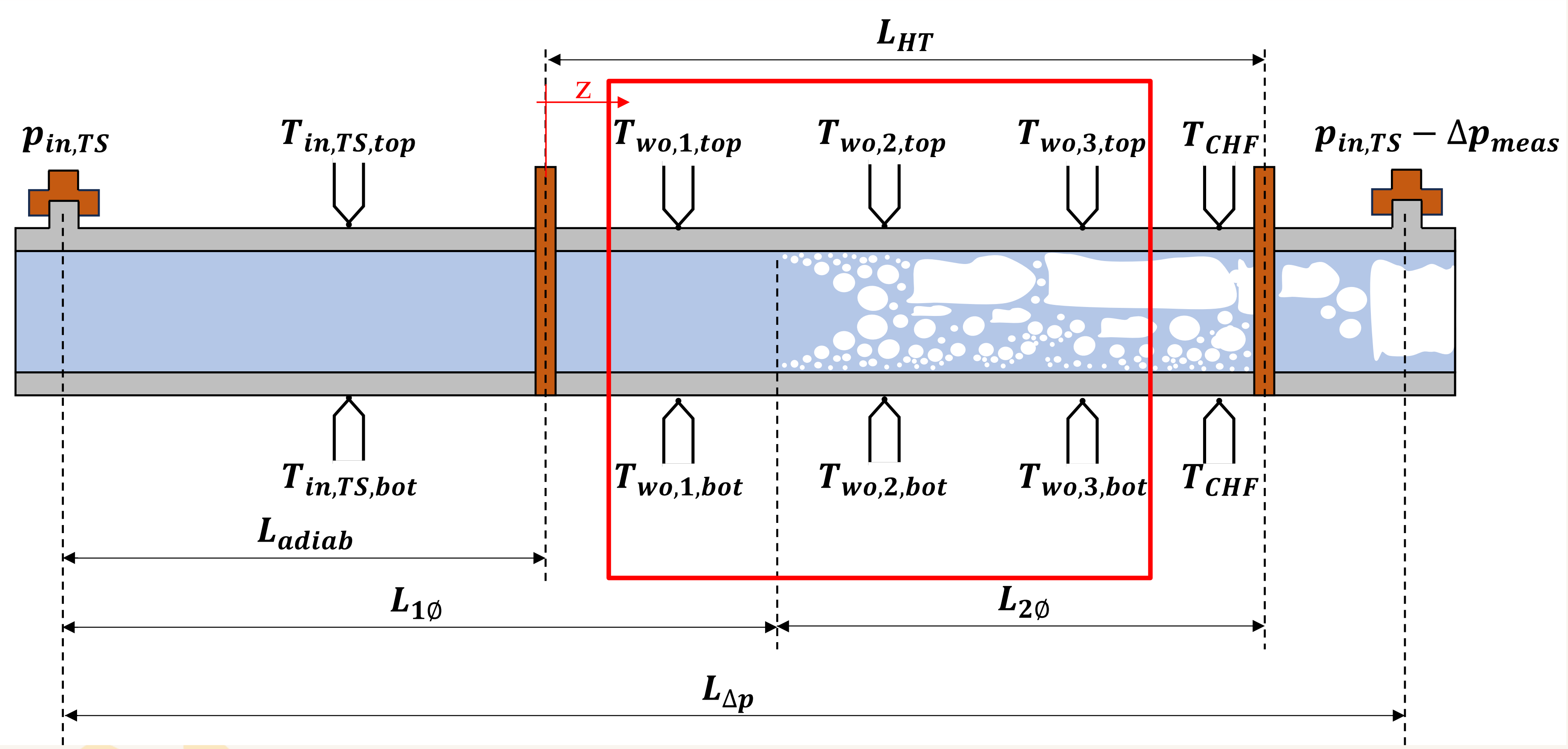

Temperatures

$$T_{fluid}(z) = T_{in,TS} + \frac{q''_{eff}.\pi.D_i.z}{\dot{m}.c_p}\ (1\emptyset) \text{ or } T_{sat}\ (2\emptyset)$$

- c_p: Heat capacity [J/kgK]
- $\dot{m}$: Mass flow rate [kg/s]
- $T_{in,TS}$: Inlet temp. [°C]

$$T_{wi}(z) = T_{wo}(z) + \frac{\dot{q}}{4k(z)}\left[\left(\frac{D_o}{2}\right)^2 - \left(\frac{D_i}{2}\right)^2\right] + \left[\frac{\dot{q}}{2k(z)}\left(\frac{D_o}{2}\right)^2 - \frac{q''_{loss}D_o}{k(z)}\right] ln\left(\frac{D_i}{D_o}\right)$$

Heat transfer Coefficient

$$h(z) = \frac{q''_{eff}}{T_{wi}(z) - T_{fluid}(z)}$$

- q''_{eff}: Effective heat flux [W/m²]
- $T_{wi}(z)$: Internal wall temperature [°C]
- $T_{wi}(z)$: Fluid temperature [°C]

$$\bar{h}(z) = \frac{h_{top}(z) + 2h_{mid}(z) + h_{bot}(z)}{4}$$

Heat flux

$$q''_{eff} = \frac{\dot{Q}_{eff}}{\pi D_i L_{HT}}$$

- $\dot{Q}_{eff}$: Effective heat transfer rate [W]
- D_i: Inner diameter of the channel [m]
- L_{HT}: Heated length [m]

$$\dot{Q}_{eff} = \dot{Q}_{el} - \dot{Q}_{loss}$$

Evaluated during single-phase flow tests

- $\dot{Q}_{el}$: Electrical power[W]
- $\dot{Q}_{loss}$: Heat losses [W]

- D_o:Outer diameter of the channel [m]
- $k(z)$: Stainless-steel termal conductivity [W/mK]
- q''_{loss}: Heat flux losses [W/m²]
- $\dot{q}$: Uniform heat generation [W/m³]

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Data regression and validation

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Vapor quality

$$x(z) = \frac{i(z) - i_l(p(z))}{i_v(p(z)) - i_l(p(z))}$$

$i(z)$: Local specific enthalpy [J/kg]

$i_l(z)$: Liquid specific enthalpy at $p(z)$ [J/kg]

$i_v(z)$: Vapor specific enthalpy at $p(z)$ [J/kg]

Linear pressure profile was assumed

$$i(z) = i_{in,TS} + \frac{q''.\pi.D_i.z}{\dot{m}}$$

$i_{in,TS}$: Specific enthalpy at TS inlet [J/kg]

$$i_{in,TS} = i_{in,PH} + \frac{\dot{Q}_{eff,PH}}{\dot{m}}$$

$i_{in,PH}$: Specific enthalpy at PH inlet [J/kg]

$\dot{Q}_{eff,,PH}$: Preheater effective heat transfer rate [W]

Working fluid properties according to REFPROP 10 (LEMMON *et al.*, 2018)

Mass velocity

$$G = \frac{4\dot{m}}{\pi.D_i^2}$$

Adiabatic frictional pressure drop

$$\left(\frac{dp}{dz}\right)_f = \frac{\Delta p_f}{L_{\Delta p}}$$

Δp_f : Frictional pressure drop [Pa]

$L_{\Delta p}$: Distance between pressure taps [m]

$$\Delta p_f = \Delta p_{meas} - \Delta p_{ac}$$

Δp_{meas}: Measured pressure drop [Pa]

Δp_{meas}: Accelerational pressure drop [Pa]

$$\Delta p_{ac} = G^2.\left\{\left[\frac{x^2}{\alpha.\rho_v} + \frac{(1-x)^2}{(1-\alpha).\rho_l}\right]_{out} - \left[\frac{x^2}{\alpha.\rho_v} + \frac{(1-x)^2}{(1-\alpha).\rho_l}\right]_{in}\right\}$$

α calculated according to Kanizawa and Ribatski (2016)

Average vapor quality

$$\bar{x} = \frac{x_{in,TS} + x_{out,TS}}{2}$$

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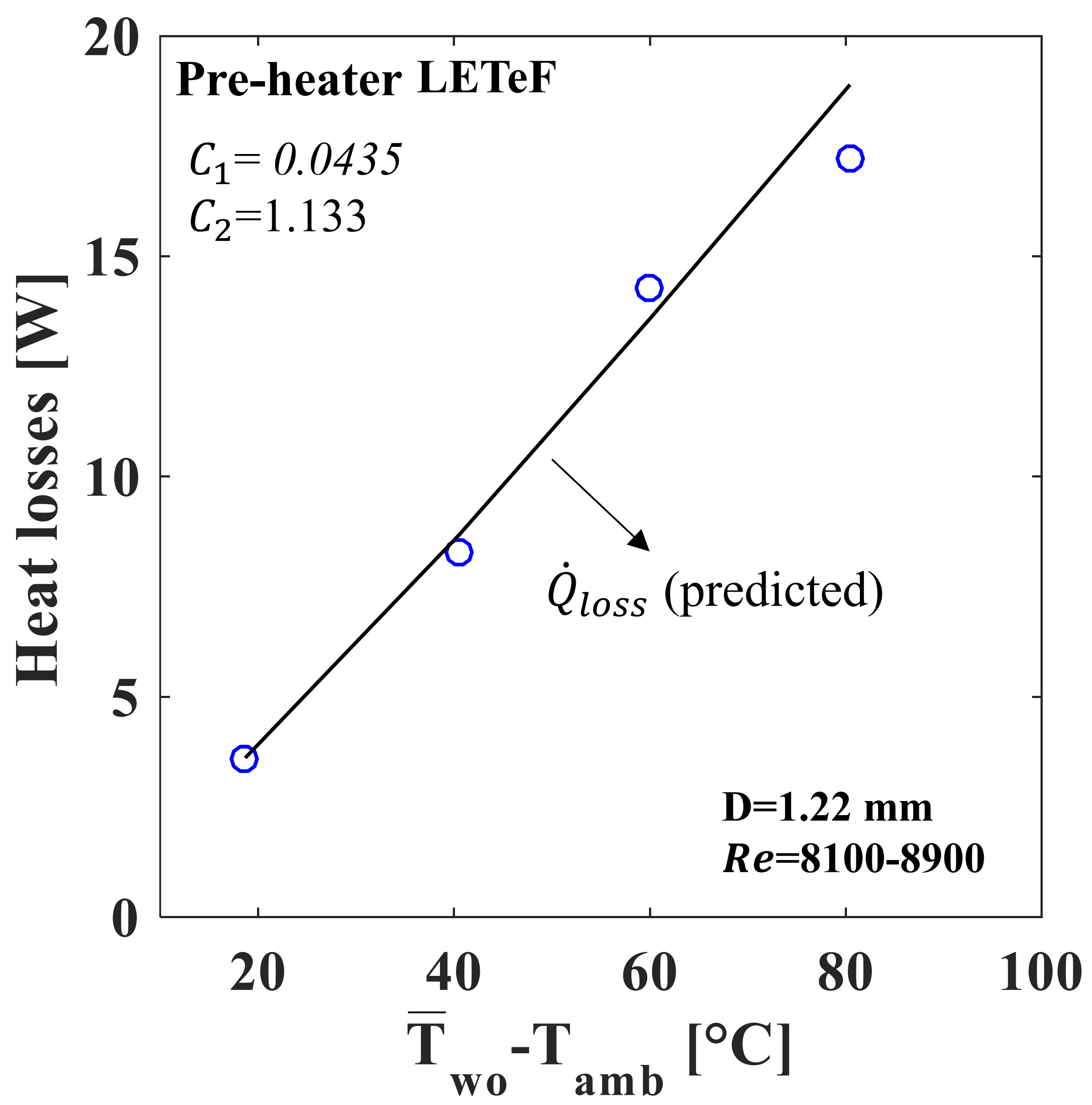
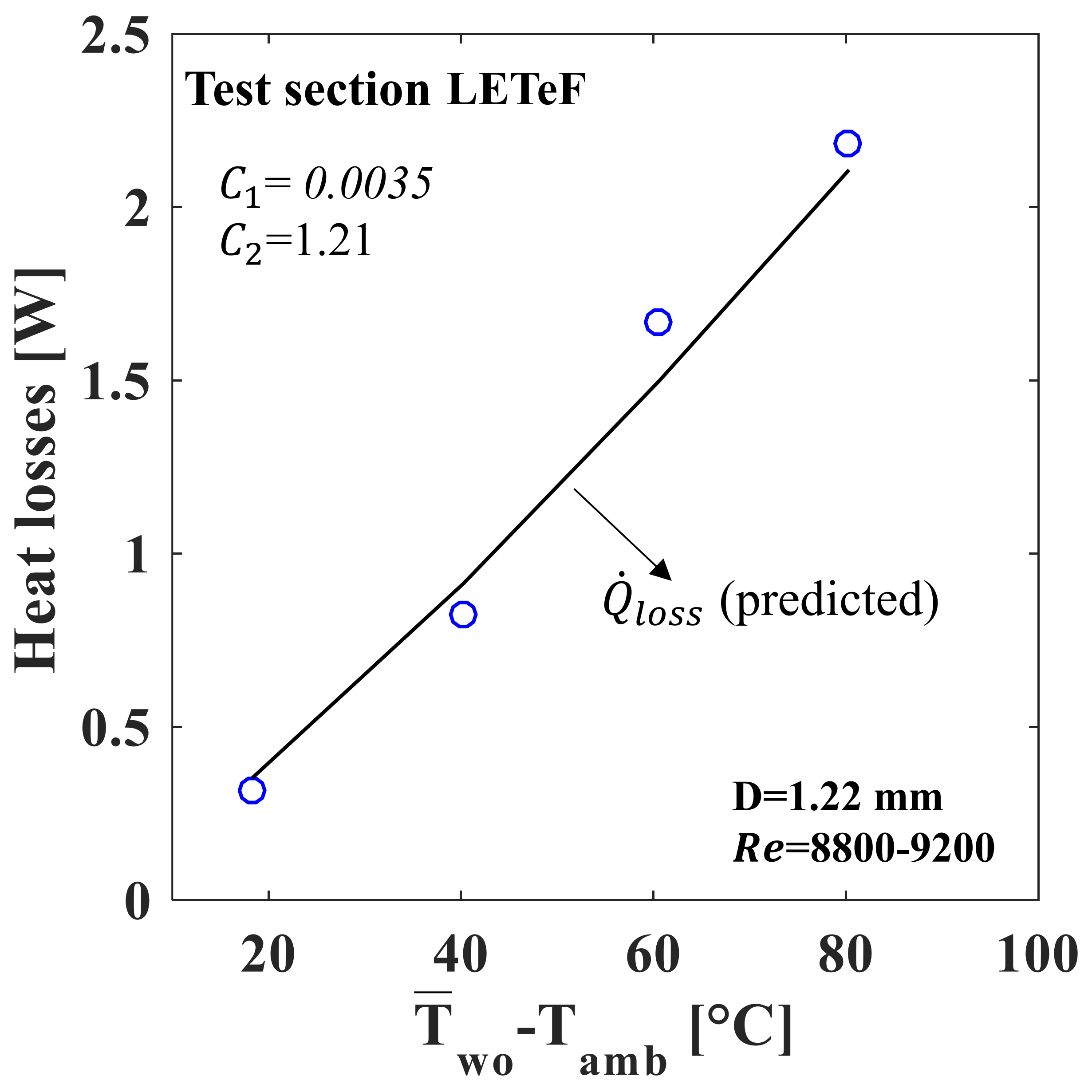
Data regression and validation

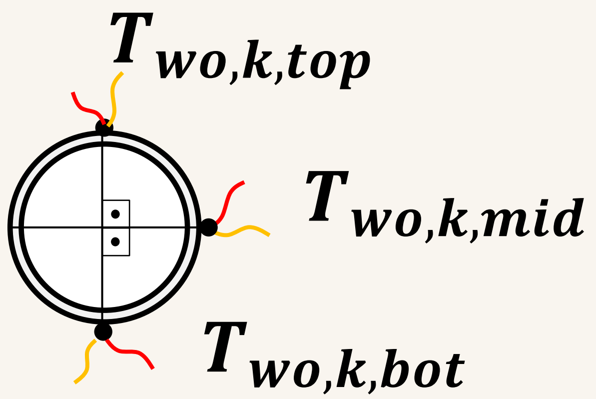
Heat losses evaluation





Fitted equation

$$\dot{Q}_{loss} = \sum_{k=1}^{3} C_1\left(\bar{T}_{wo,k} - T_{amb}\right)^{C_2}$$



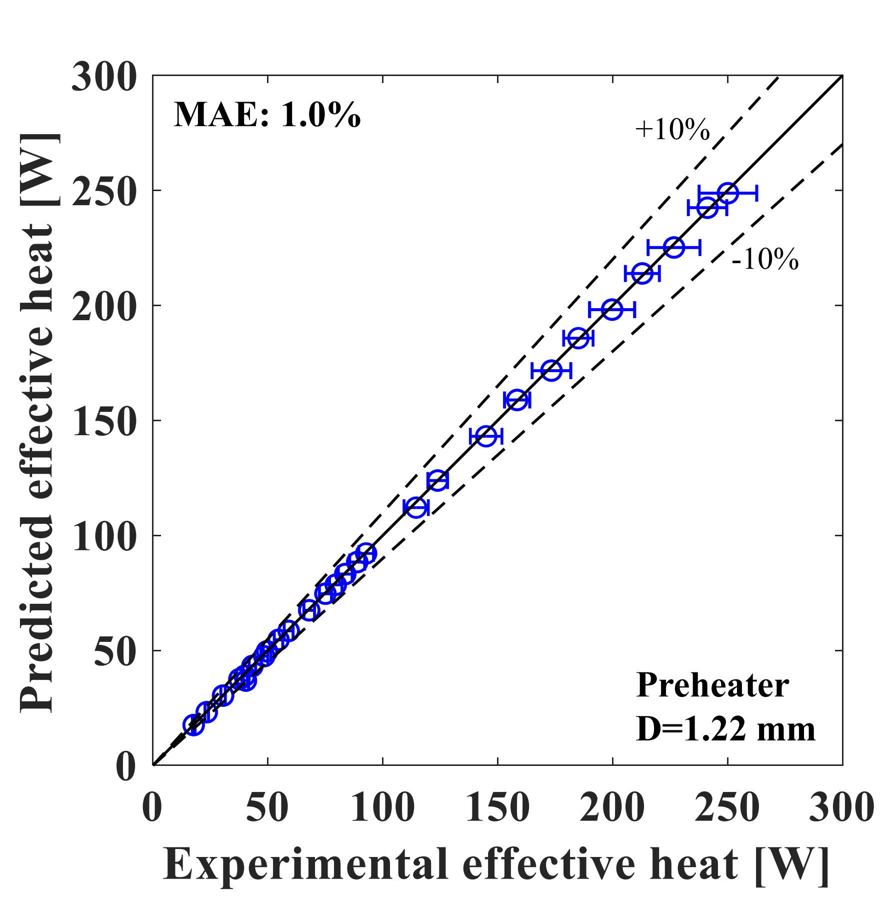
$$\bar{T}_{wo,k} = \frac{T_{wo,k,bot}+2\left(T_{wo,k,mid}\right)+T_{wo,k,bot}}{4}$$

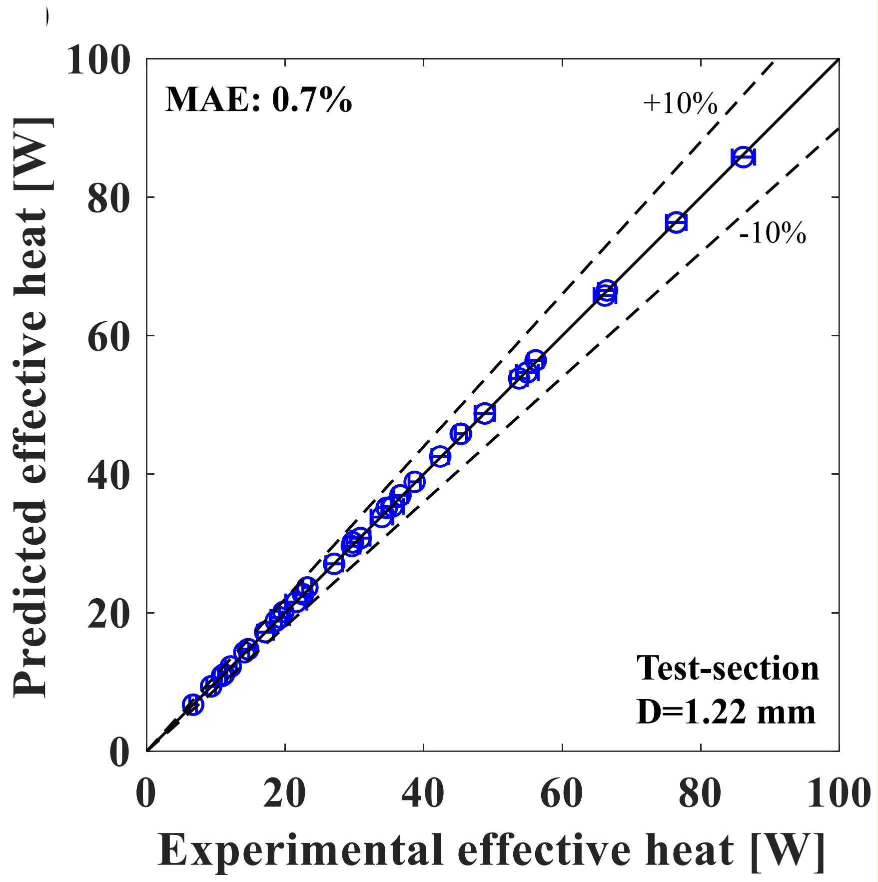
OBS: Preheater only 2 thermocouples

Data regression and validation

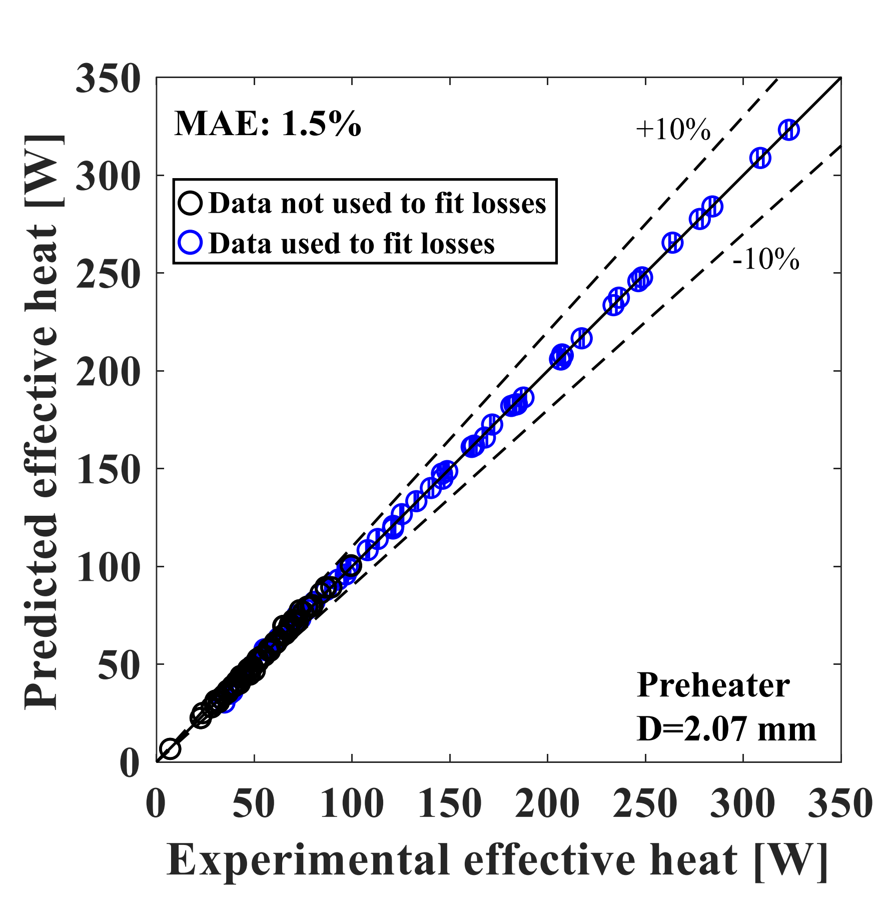
Heat losses evaluation

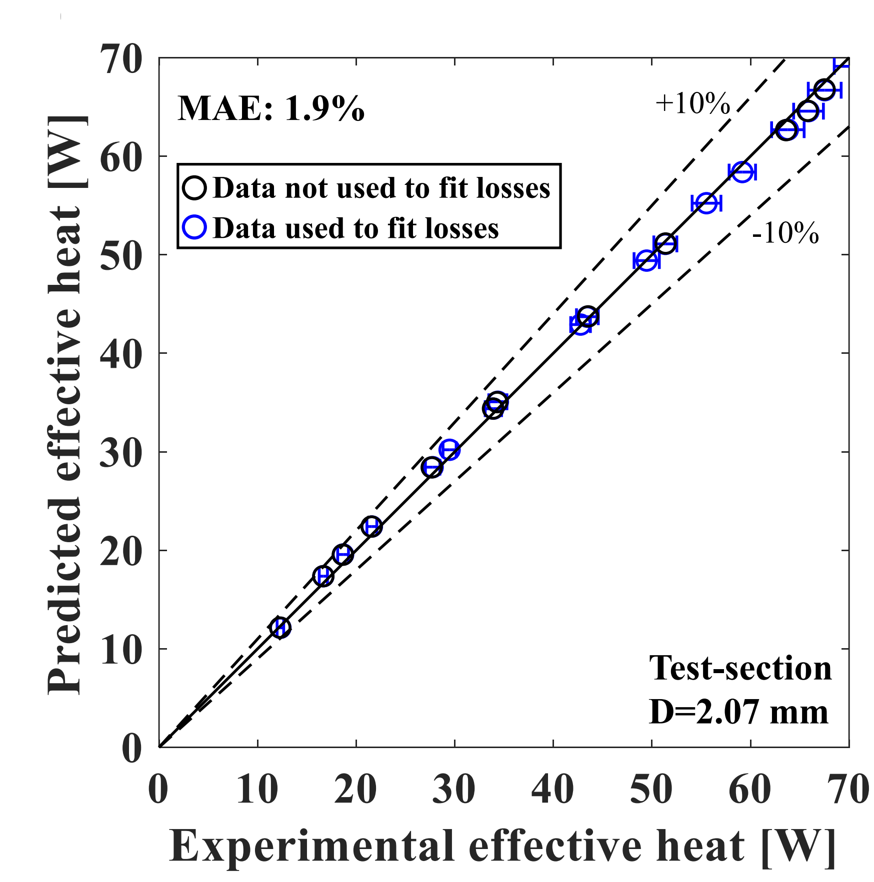
LETeF:





CETHIL





Data regression and validation

Uncertainties

- Thermocouples and pressure sensors were calibrated prior to the experiments
- Thermocouples uncertainties were calculated according to Abernethy and Thompson Jr. (1973)
- The sequential perturbation method proposed by Moffat *et al.* (1988) was followed to estimate the uncertainties of calculated parameters:

$$\delta Y = \sqrt{\sum_{n=1}^{N}\left(\frac{\delta Y}{\partial X_{meas,n}} . \delta X_{meas,n}\right)^{2}}$$

Y: Calculated parameter

X_{meas}: Measured parameter

LETeF uncertainties

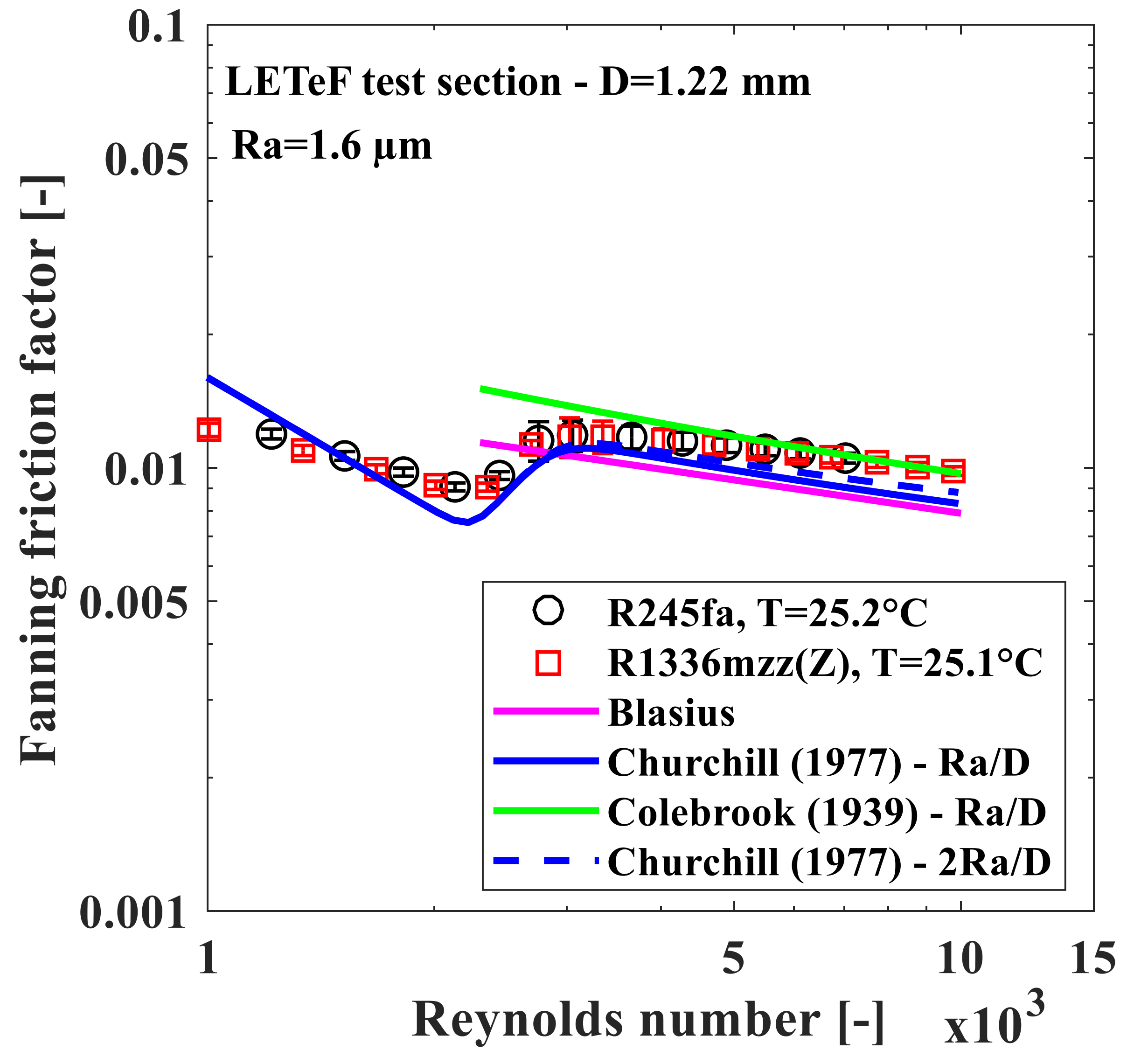
Parameter	Uncertainty
D_i	0.005 mm
L_{HT}	1 mm
$h(z)$	8.6%
G	1%
q''_{TS}	7.2%
T_{wo}	0.3°C
dp/dz	2%
$x(z)$	0.039

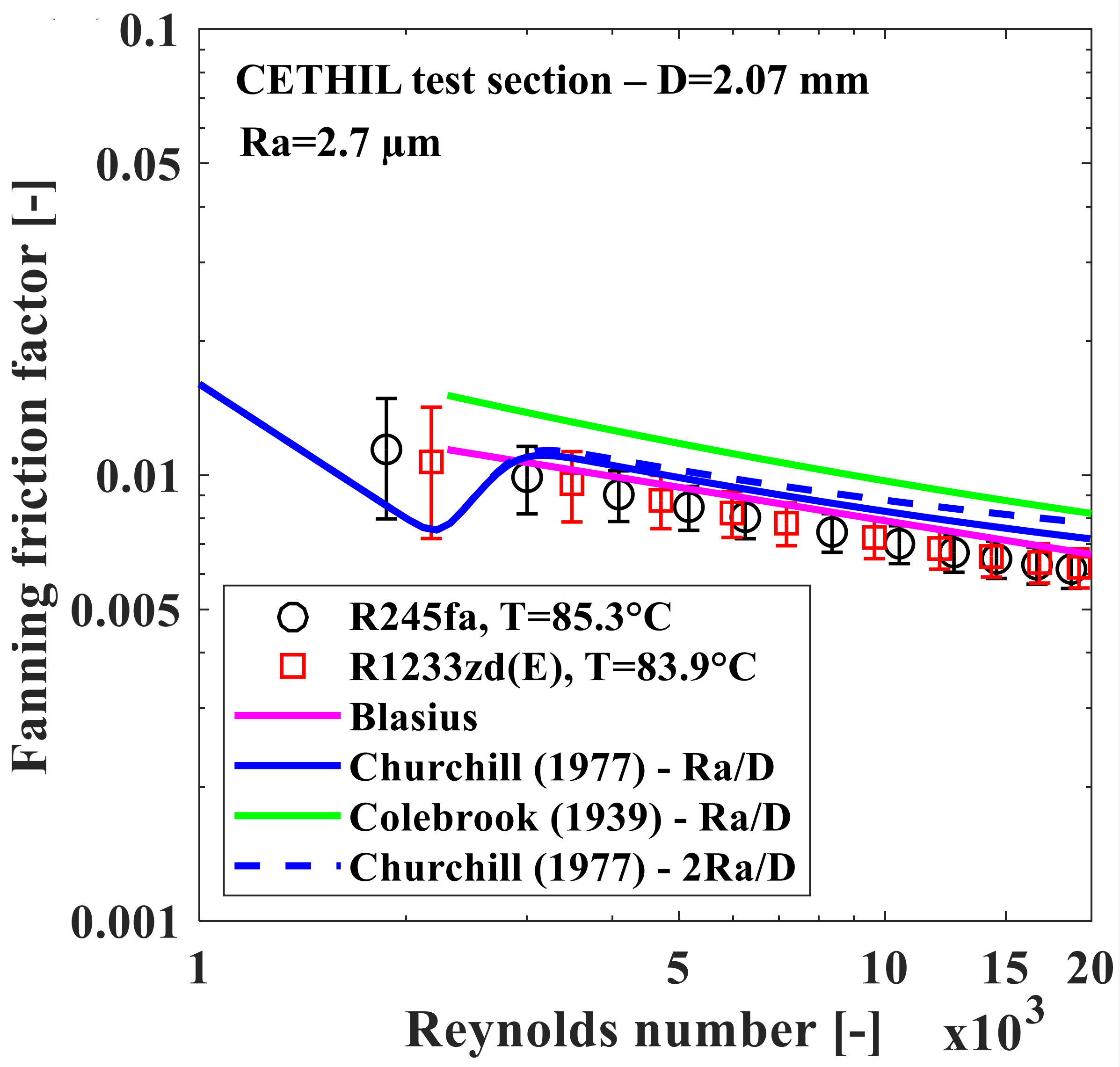
CETHIL uncertainties

Parameter	Uncertainty
D_i	0.048 mm
L_{HT}	1 mm
$h(z)$	10.1%
G	4.6%
q''_{TS}	6.5%
T_{wo}	0.15°C
dp/dz	2.1%
$x(z)$	0.035

Data regression and validation

Validation Δp

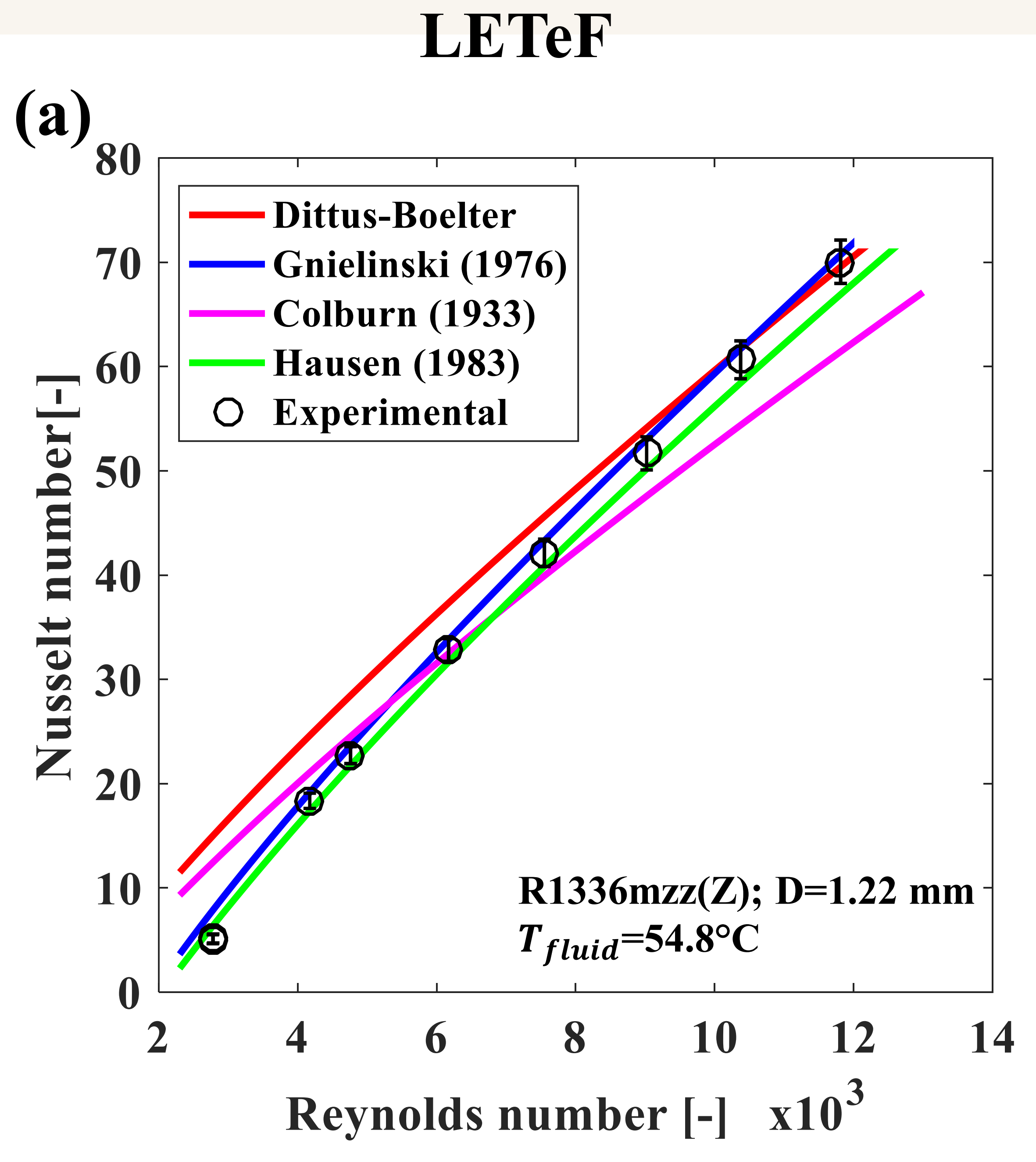


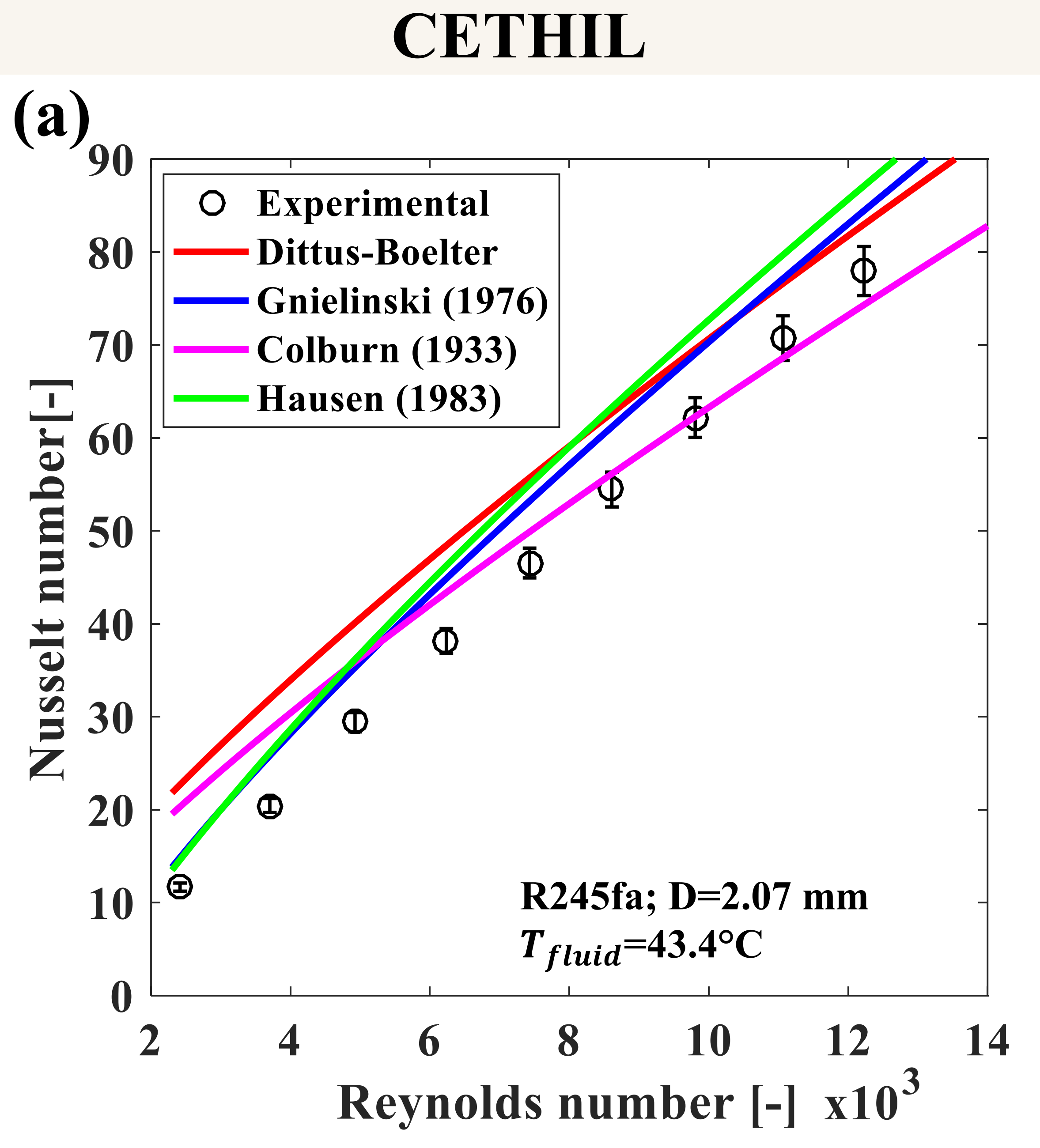


- Churchill (1977) correlation was evaluated considering 2 definitions of relative roughness;
- LETeF data presented reasonable agreement with the prediction methods of Churchill (1977) and Colebrook (1939), MAEs=9 – 13%;
- CETHIL data presented better agreement with Blasius correlation, MAE=10.8%

Data regression and validation

Validation Heat Transfer Coefficient





- At low Reynolds numbers lower deviations are verified for Hausen (1983) and Gnielinski (1976) correlations;
- The experimental Nusselt number progressively approaches Dittus-Boelter predictions with increasing Reynolds number;
- Reproducibility tests were also performed to validate the HTC measurements

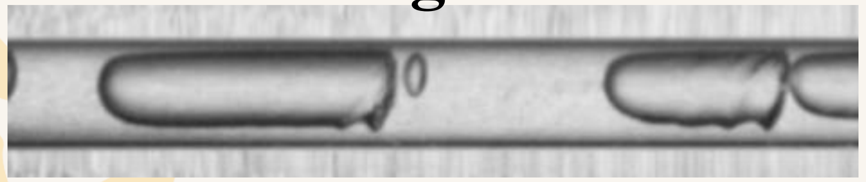
Results

Flow patterns

Bubbly

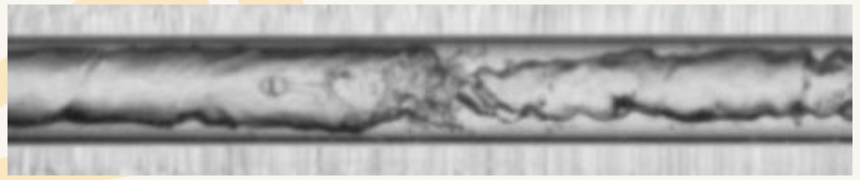
x=0.02

Plug



x=0.08

Slug



x=0.20

Annular

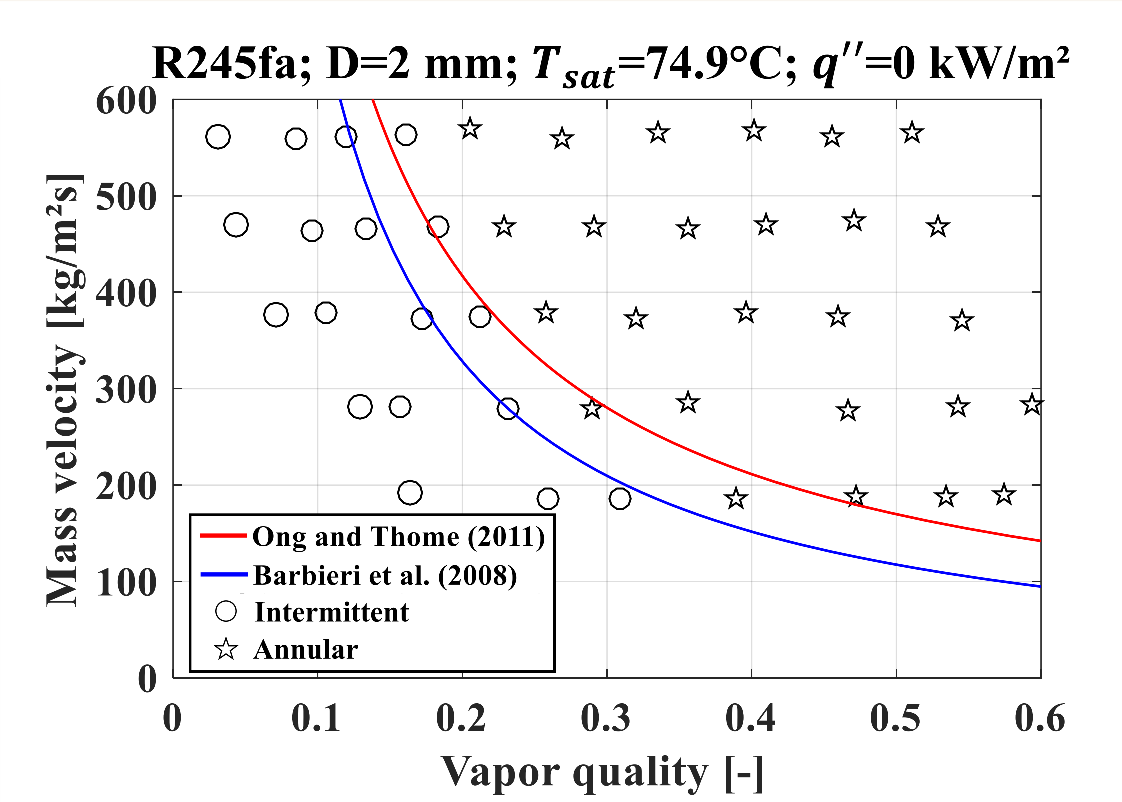
x=0.36

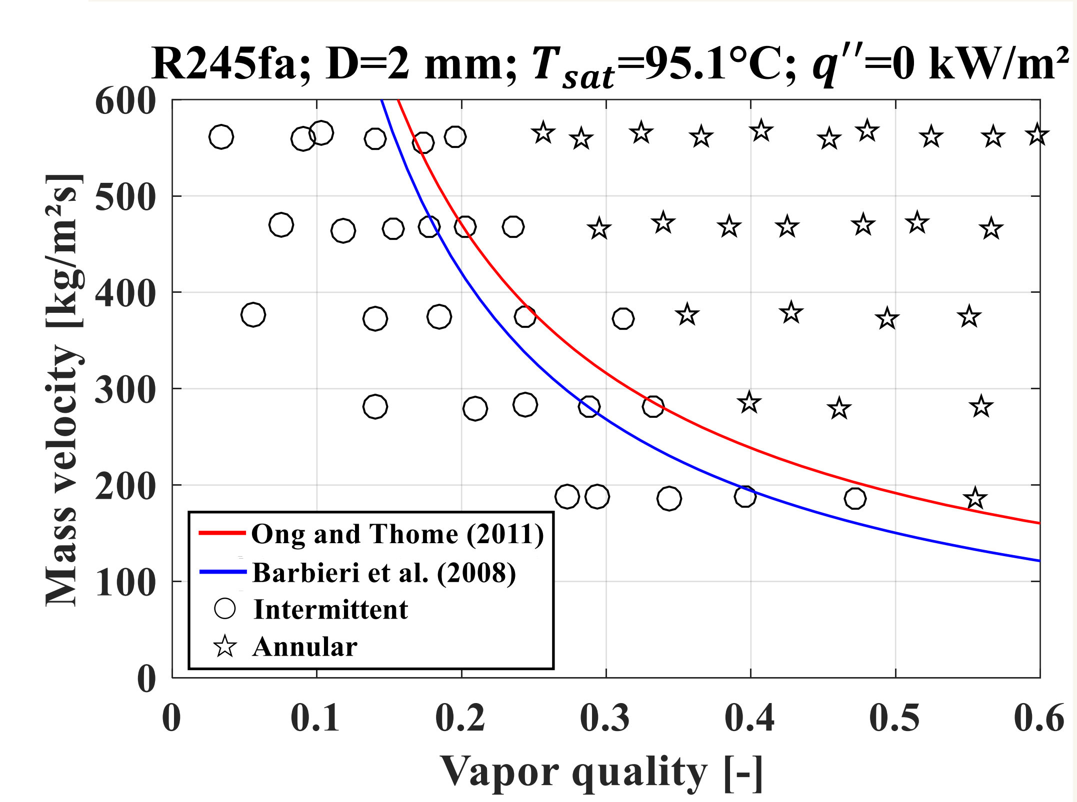
Dryout

x=0.96

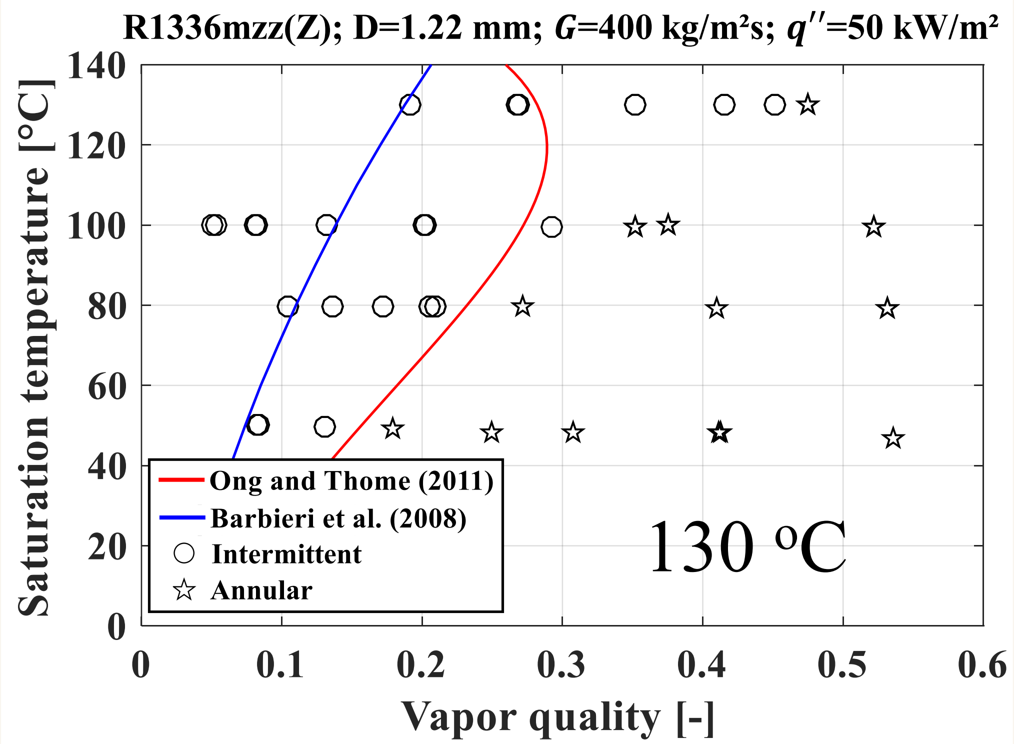
5000 fps/Playback frame rate: 100 fps

I-A transition G effect



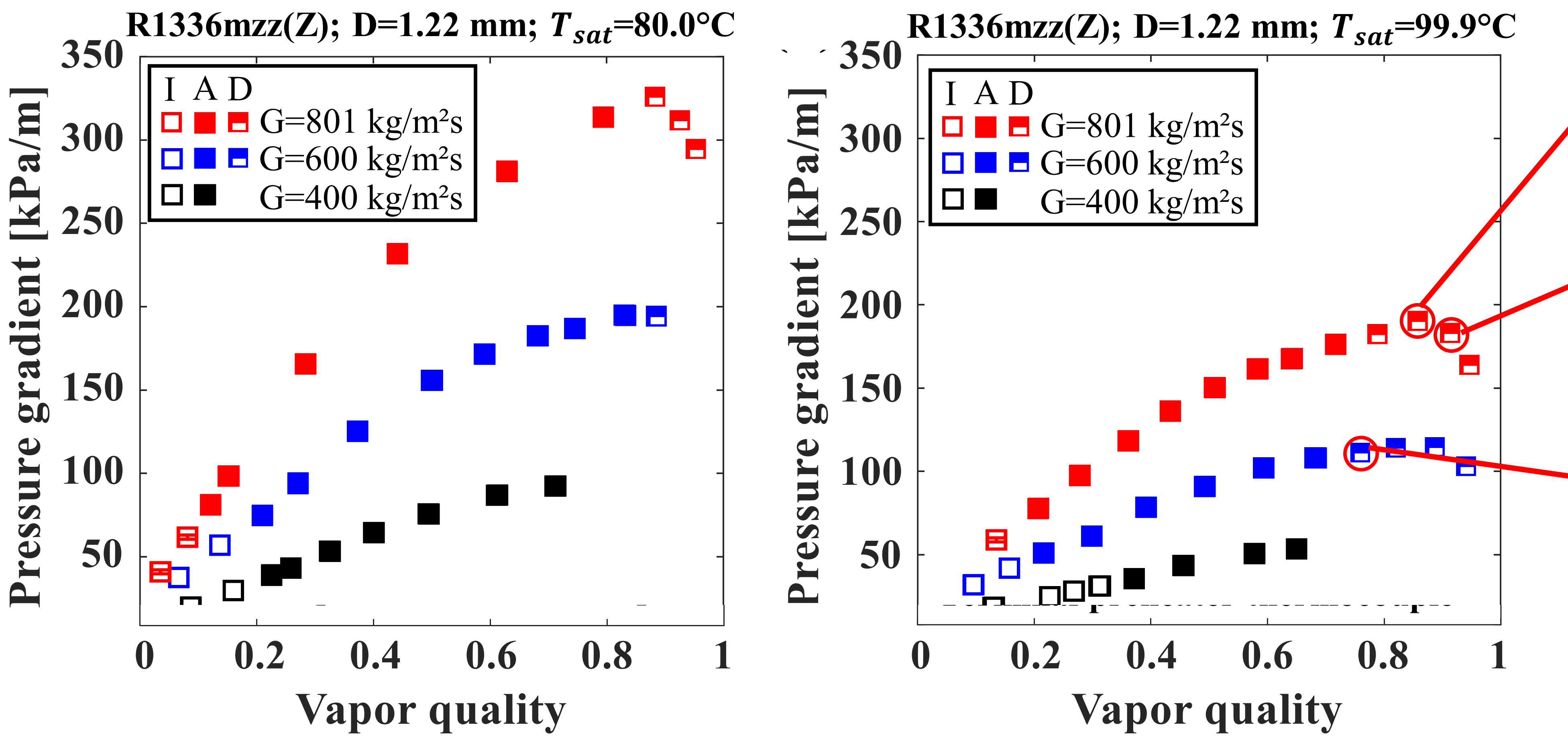


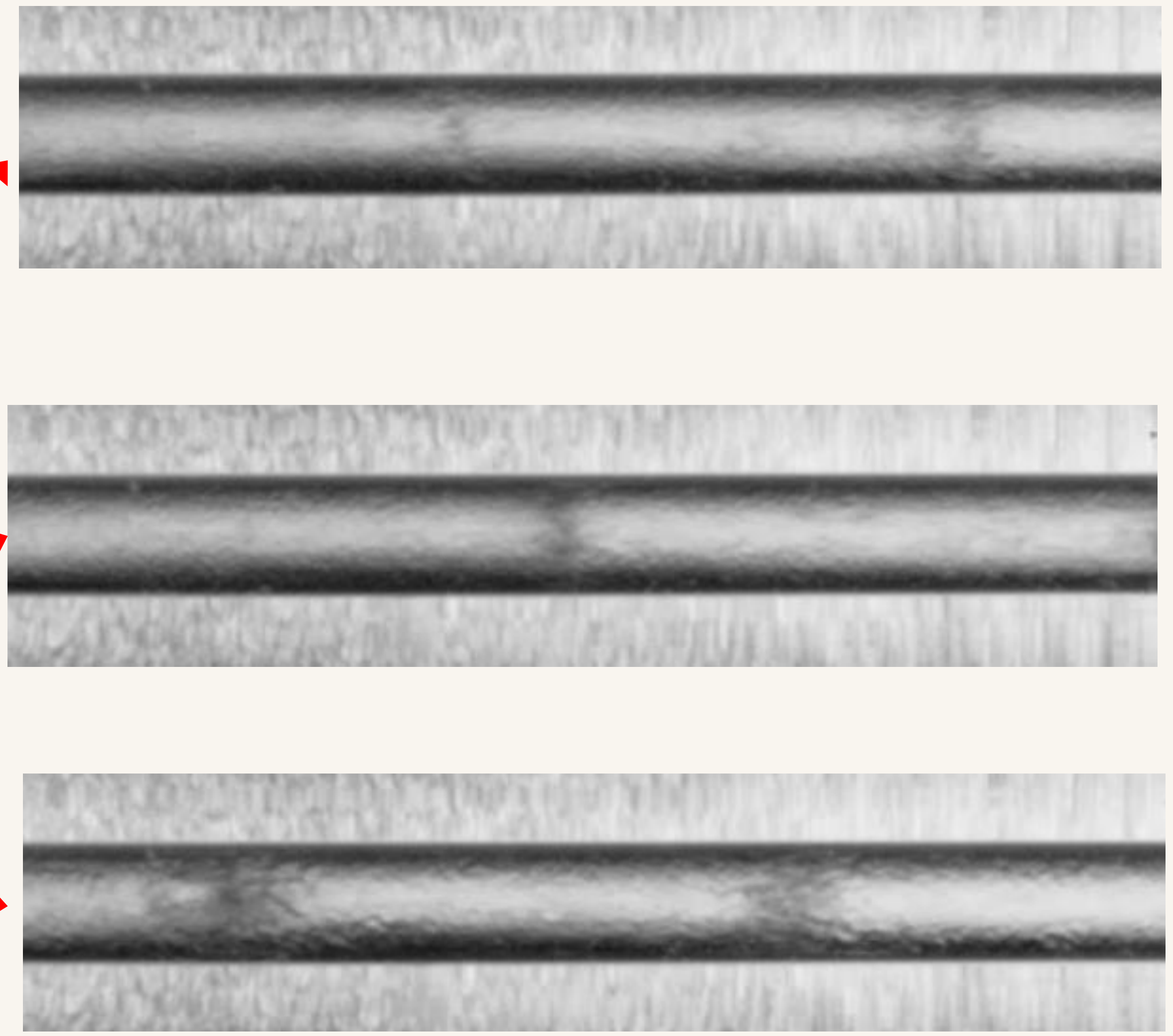
I-A transition *T_sat* effect



Results

Pressure drop: mass velocity effect

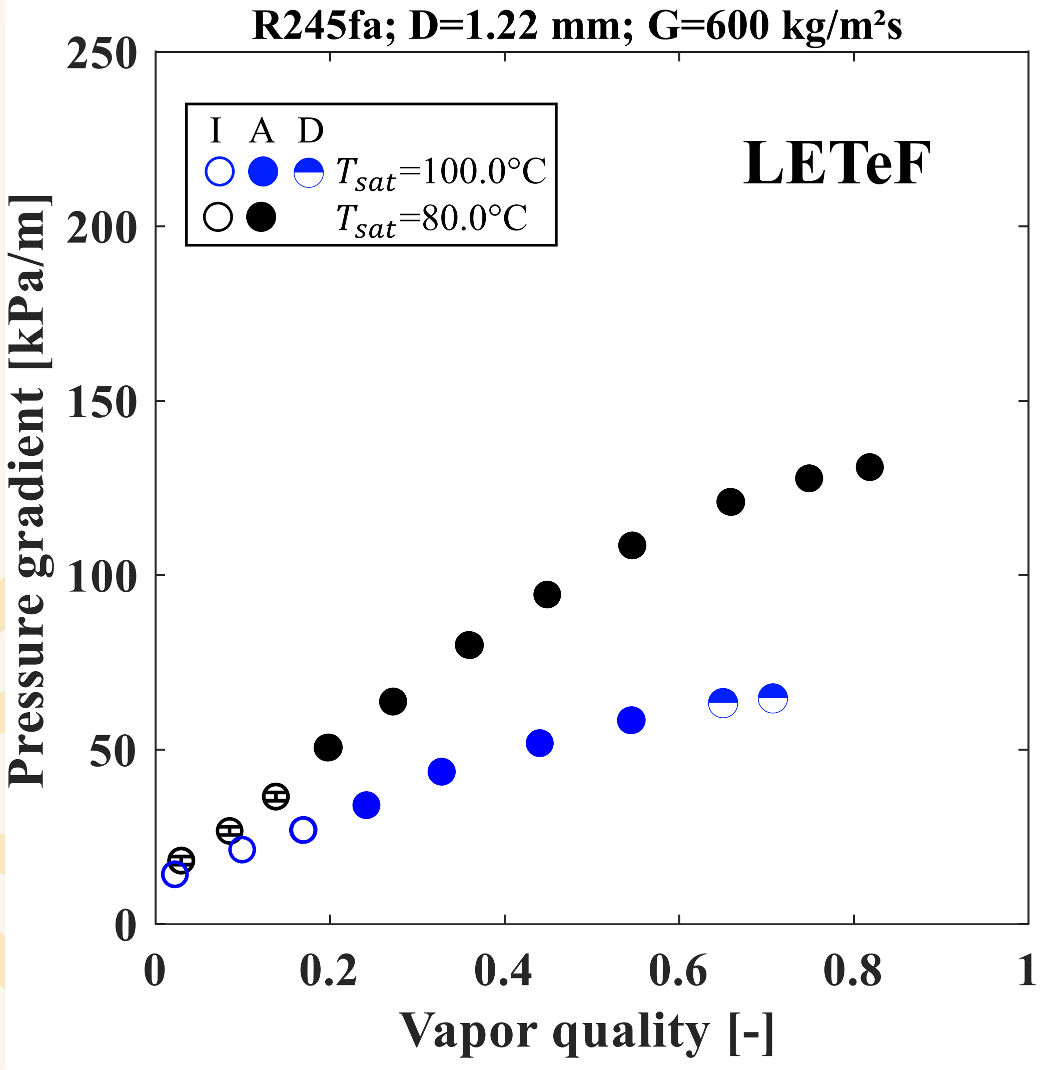


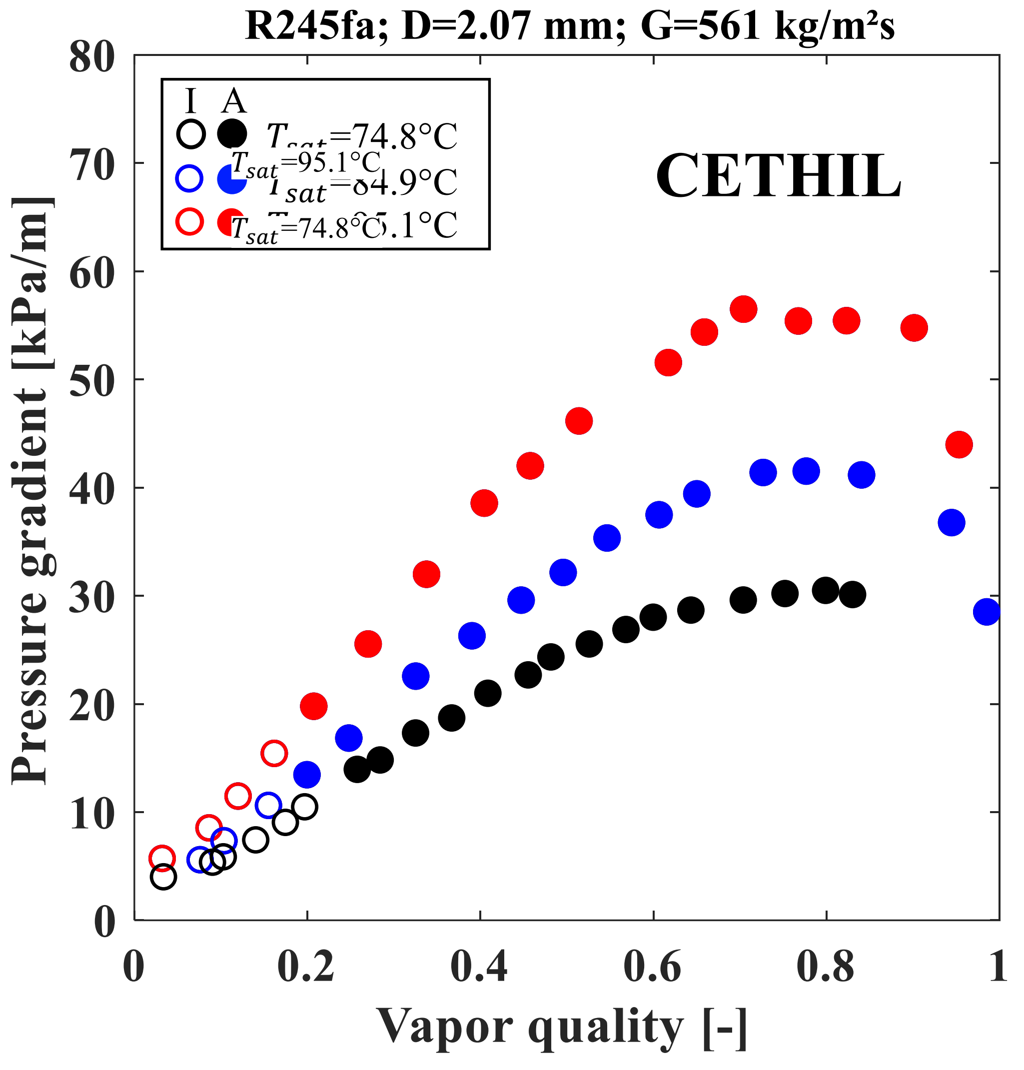


5000 fps
Playback frame rate: 100 fps

Results

Pressure drop: T_{sat} effect

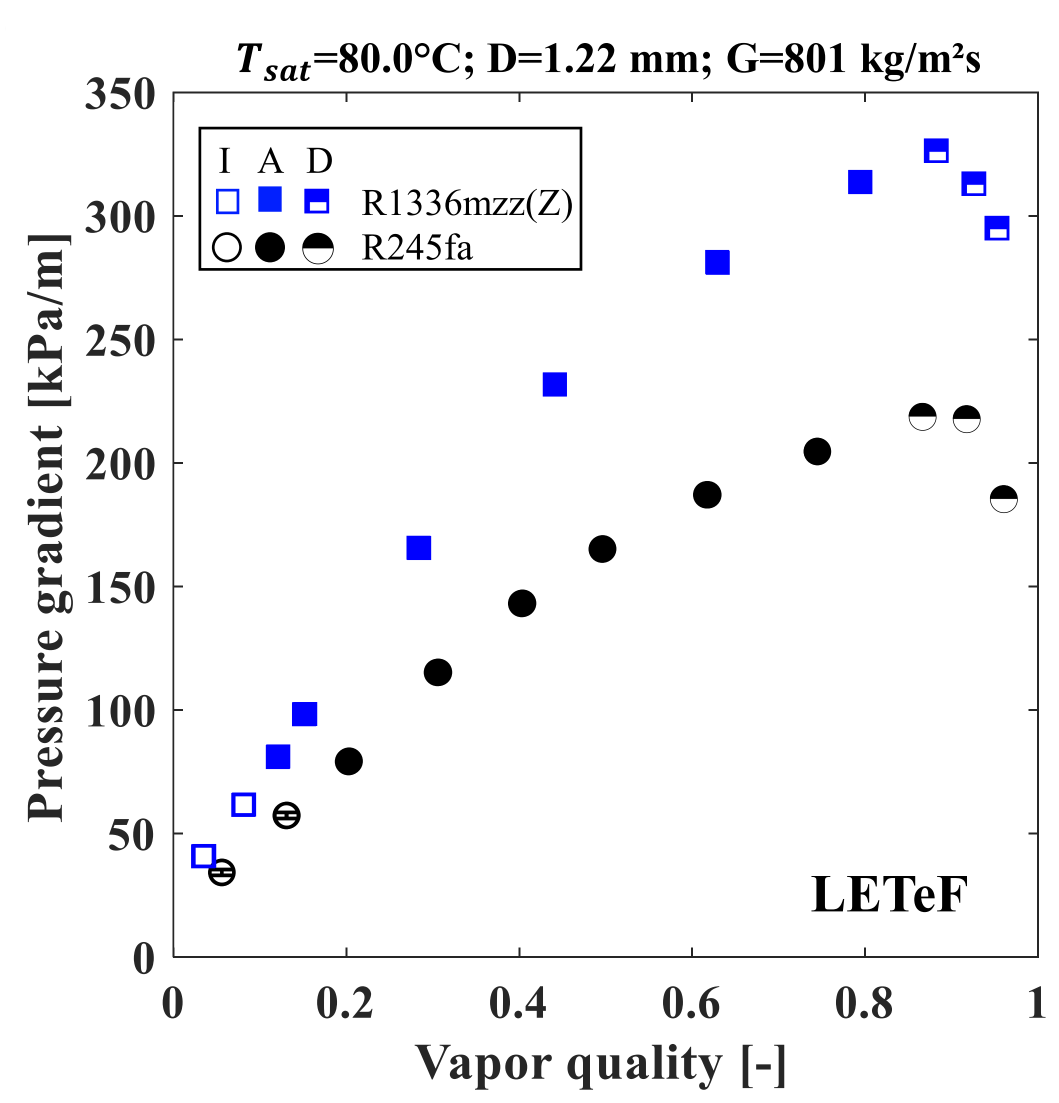


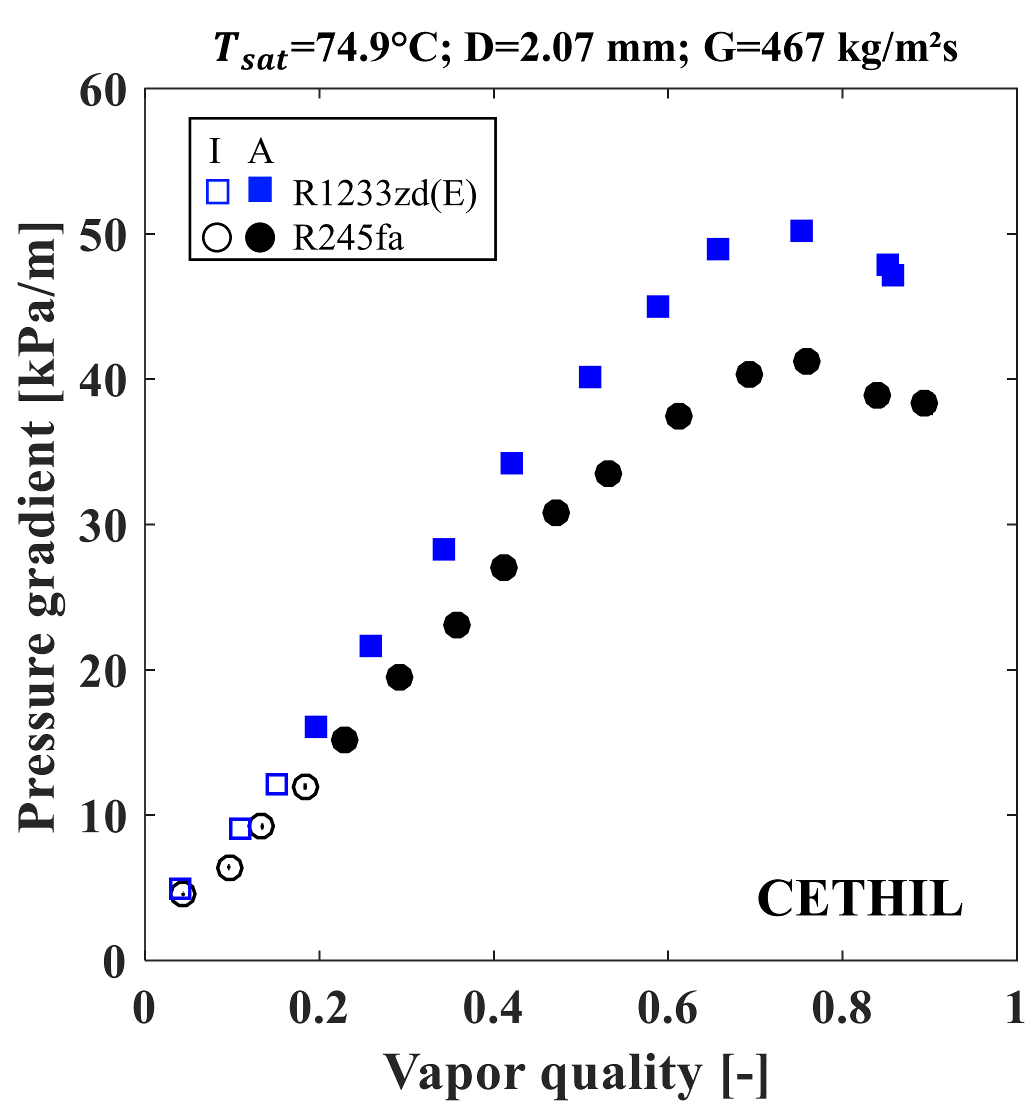


	Property	R245fa	
	T_{sat} [°C]	80	100
	p_r [-]	0.216	0.346
↑65%	ρ_l [kg/m³]	1170.5	1093.7
↓44%	ρ_v [kg/m³]	43.64	72.38
	ρ_l/ρ_v [-]	26.82	15.11
	μ_l [μPa.s]	201.2	155.1
	μ_v [μPa.s]	12.46	13.34
	μ_l/μ_v [-]	16.1	11.7

Results

Pressure drop: working fluid effect





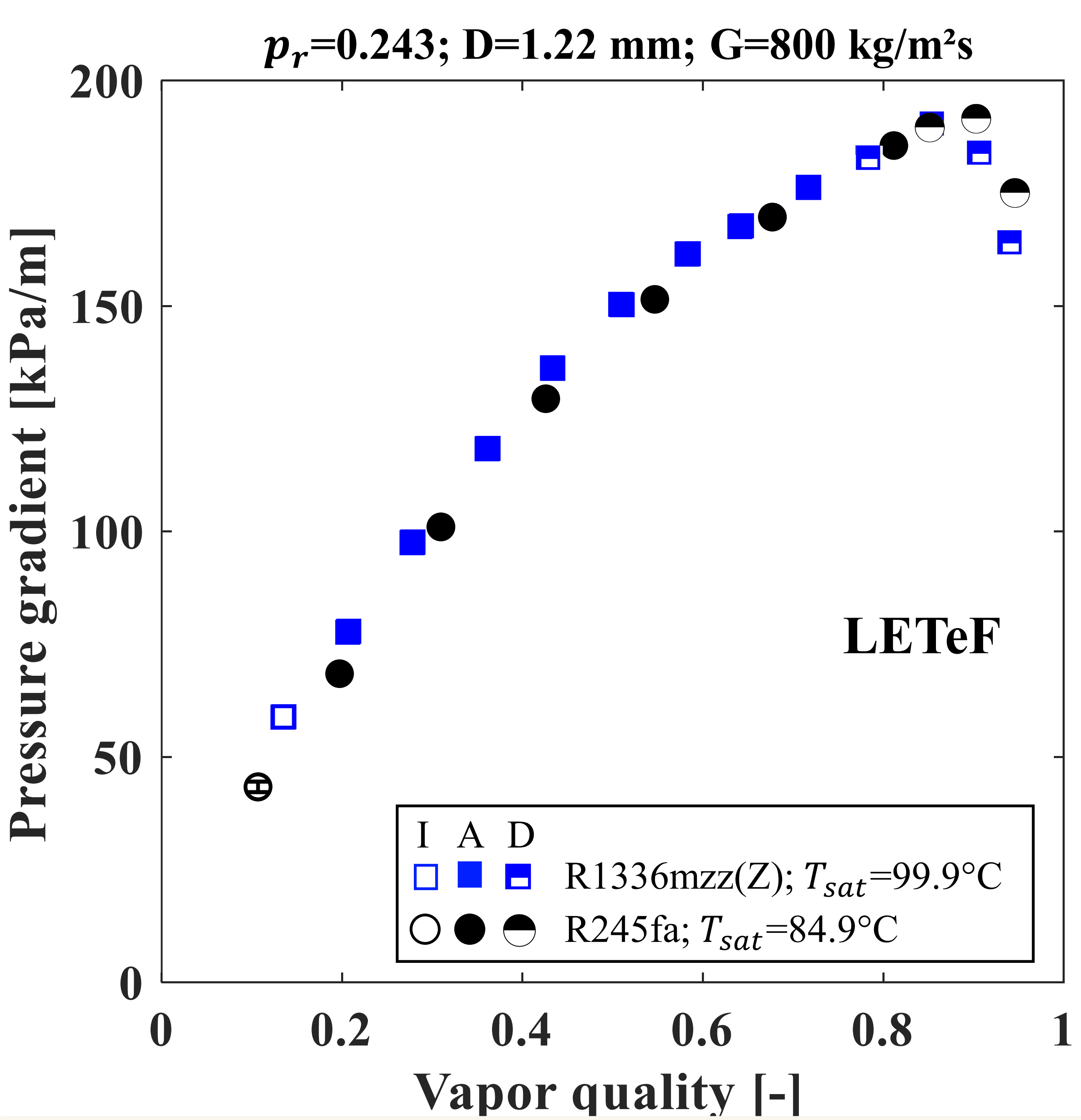
	Property	Fluid	
		R245fa	R1336mzz(Z)
	T_{sat} [°C]	80	80
	p_r [-]	0.216	0.148
	ρ_l [kg/m³]	1170.5	1207.1
≠ 56%	ρ_v [kg/m³]	43.64	27.93
≠61%	ρ_l/ρ_v [-]	26.82	43.22

	Property	Fluid	
		R245fa	R1233zd(E)
	T_{sat} [°C]	75	75
	p_r [-]	0.190	0.160
≠25%	ρ_v [kg/m³]	38.3	30.69
≠19%	ρ_l/ρ_v [-]	31.02	36.81

Results

Pressure drop: working fluid effect

At similar p_r



Property	Fluid	
	R245fa	R1336mzz(Z)
T_{sat} [°C]	84.8	100
p_r [-]	0.243	0.243
ρ_l [kg/m³]	1153.4	1139.1
≠ 6% ρ_v [kg/m³]	49.33	46.47
≠4% ρ_l/ρ_v [-]	23.4	24.5
μ_l [µPa.s]	189.6	157.19
μ_v [µPa.s]	12.65	12.79
μ_l/μ_v [-]	15.0	12.3

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Results

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Pressure drop: Comparison against prediction methods

21 Evaluated methods

Homogeneous-based

McAdams *et al.* (1942)
Cicchitti *et al.* (1962)
Dukler *et al.* (1964)
Beattie e Whalley (1982)
Lin *et al.* (1991)
Awad and Muzychka (2008)
Ducoulombier *et al.* (2011)

Two-phase multiplier-based

Lockhart and Martinelli (1949)
Chisholm (1973)
Friedel (1979)
Mishima and Hibiki (1996)
Zhang and Web (2001)
Yamamoto *et al.* (2007)
Ducoulombier *et al.* (2011)
Kim and Mudawar (2013)

Empirical

Müller-Steinhagen and Heck (1986)
Xu e Fang (2012)
Sempértegui-Tapia and Ribatski (2017)
Tibiriçá *et al.* (2017)

- The frictional pressure drop prediction methods evaluated were not equally accurate for both channel diameters;
- The prediction methods of Xu and Fang (2012) and Tibiriçá *et al.* (2017) predicted the LETeF (D=1.22 mm) database with MAEs of 6.7% and 15.9%, respectively;
- The prediction methods of Nie *et al.* (2023) and Muller-Steinhagen and Heck (1987) predicted the CETHIL (D=2.07 mm) database with MAEs of 12.2% and 14.8%, respectively;
- Prediction methods based on homogeneous-model presented progressive reduction of deviations with increasing saturation temperature.

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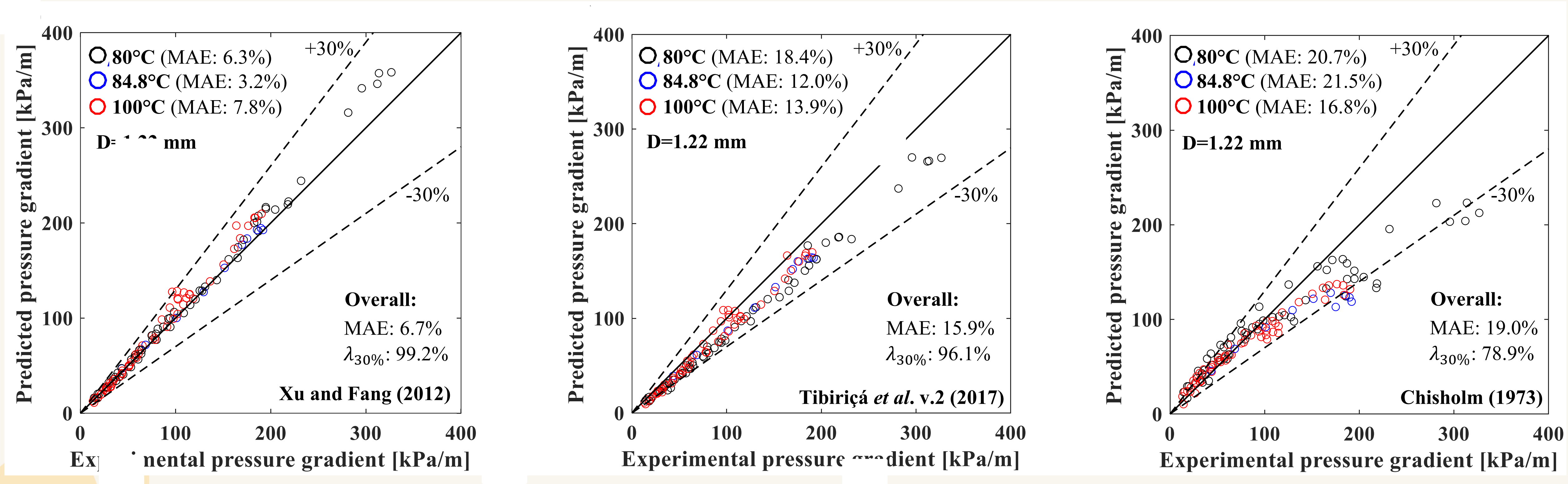
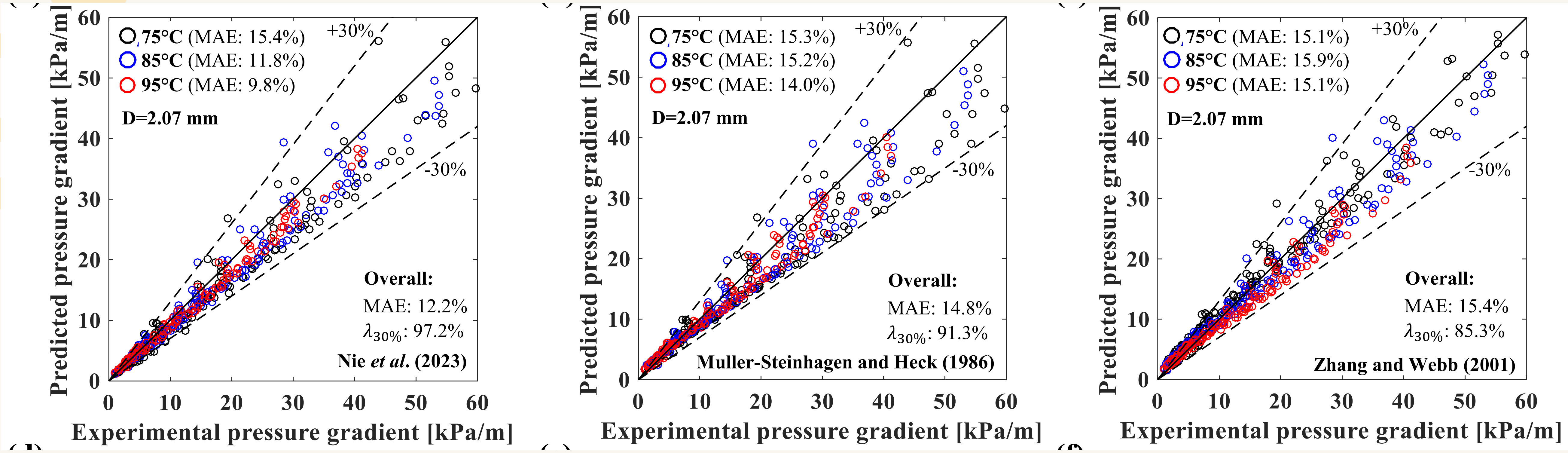
Experimental Apparatus

Pressure drop: Comparison against prediction methods



LETeF database



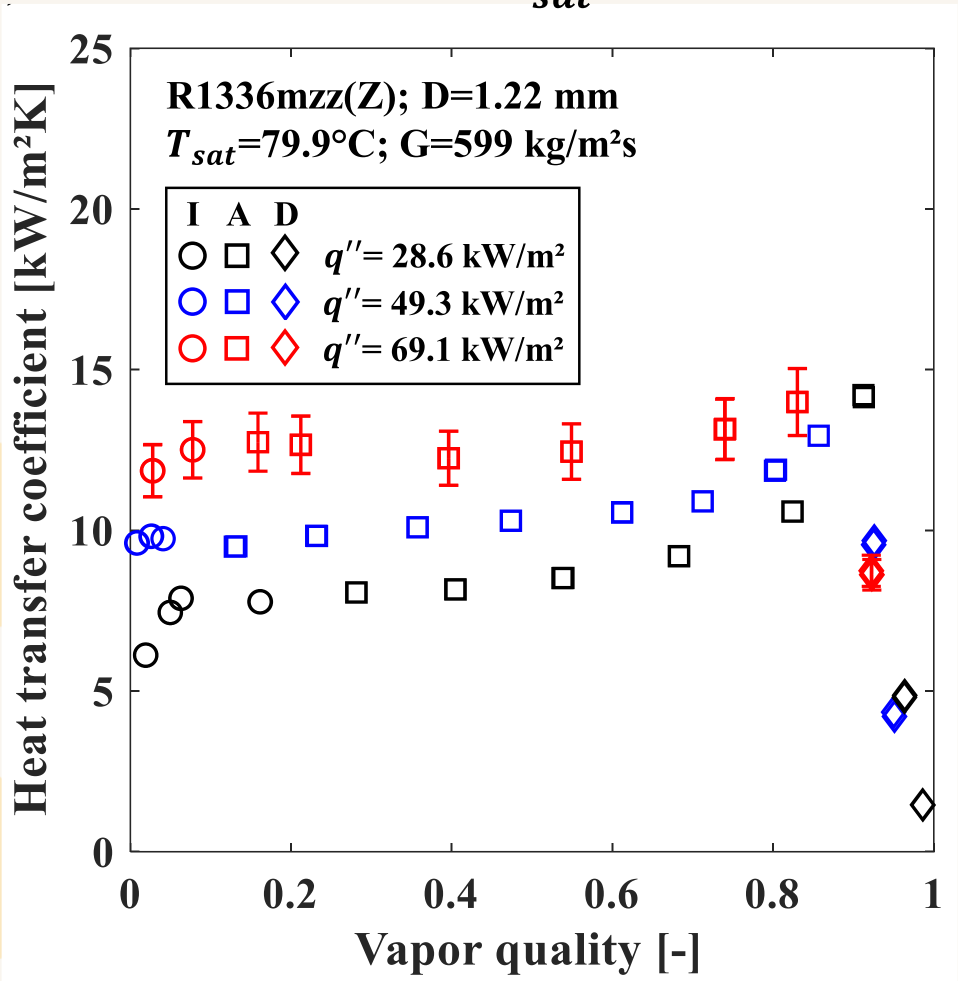
CETHIL database

Results

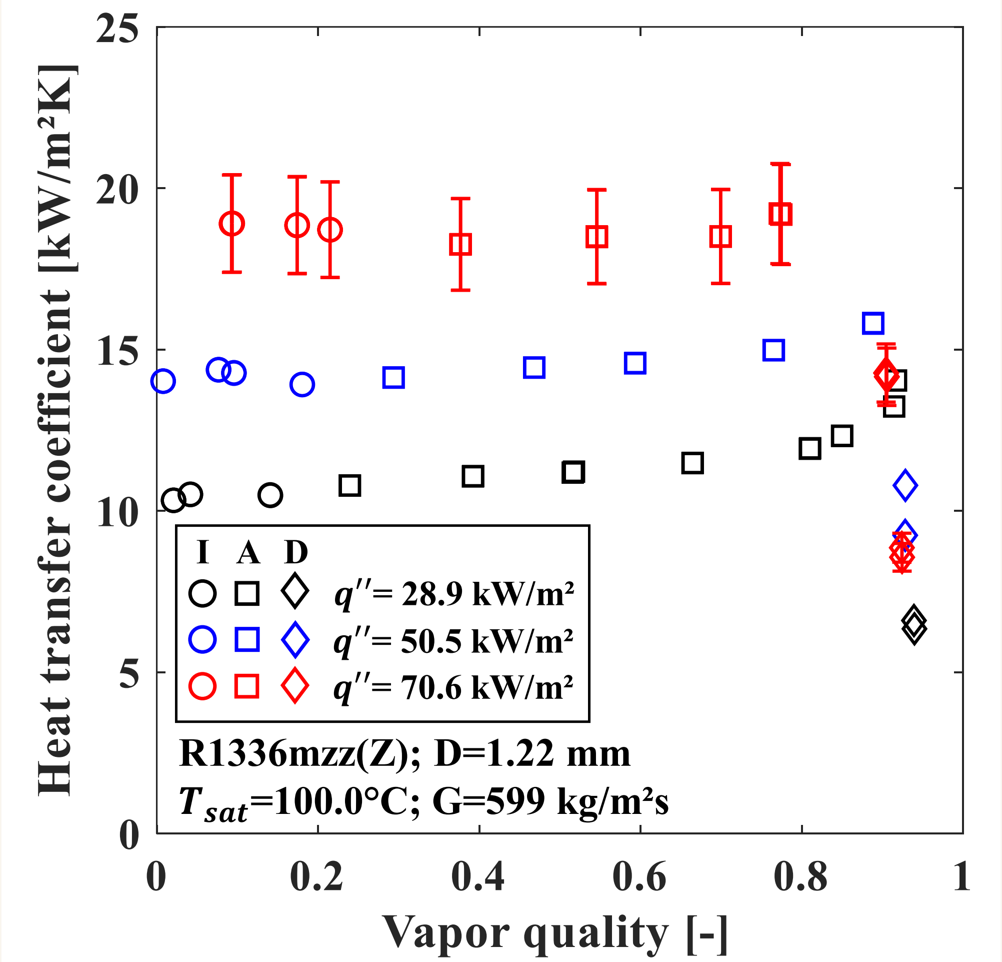
Heat transfer coefficient: Heat flux effect

LETeF data:

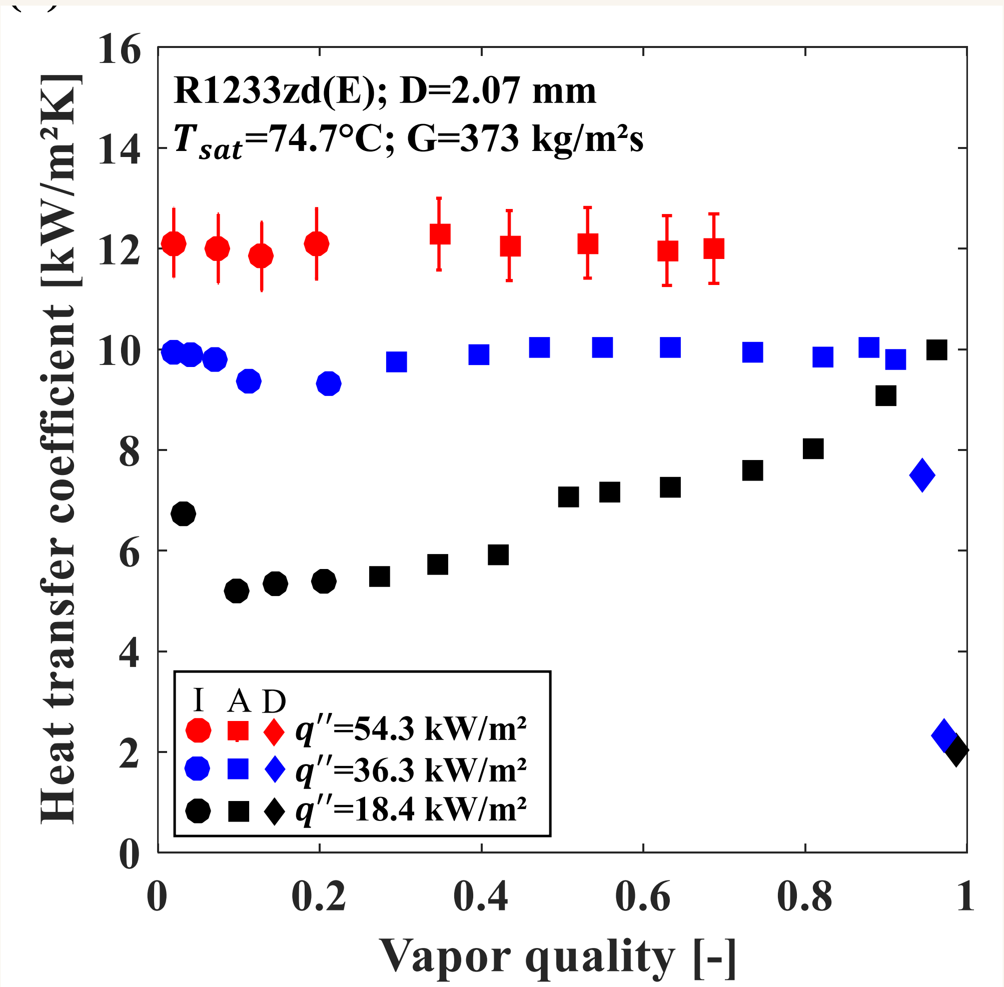
Low T_{sat}



High T_{sat}



CETHIL data

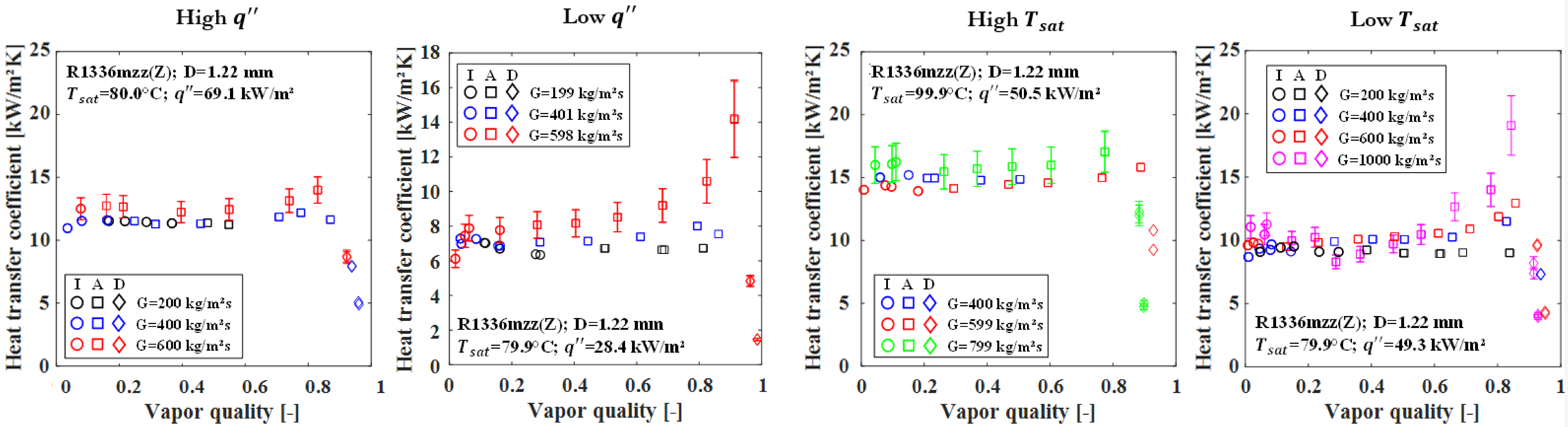


- Increasing heat flux increases the HTC regardless of vapor quality. However, the effect is more pronounced at low x

Results

Heat transfer coefficient: Mass velocity effect

LETeF data:

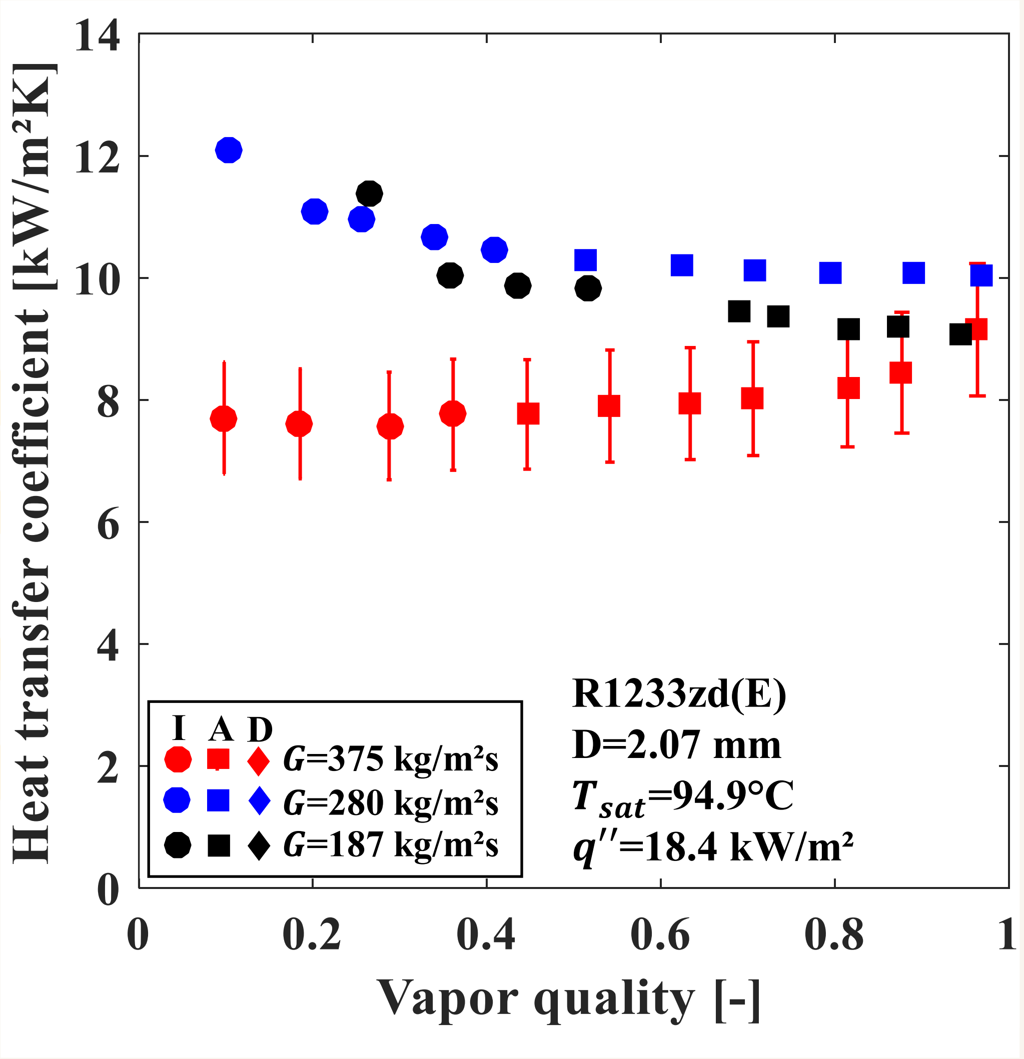


Results

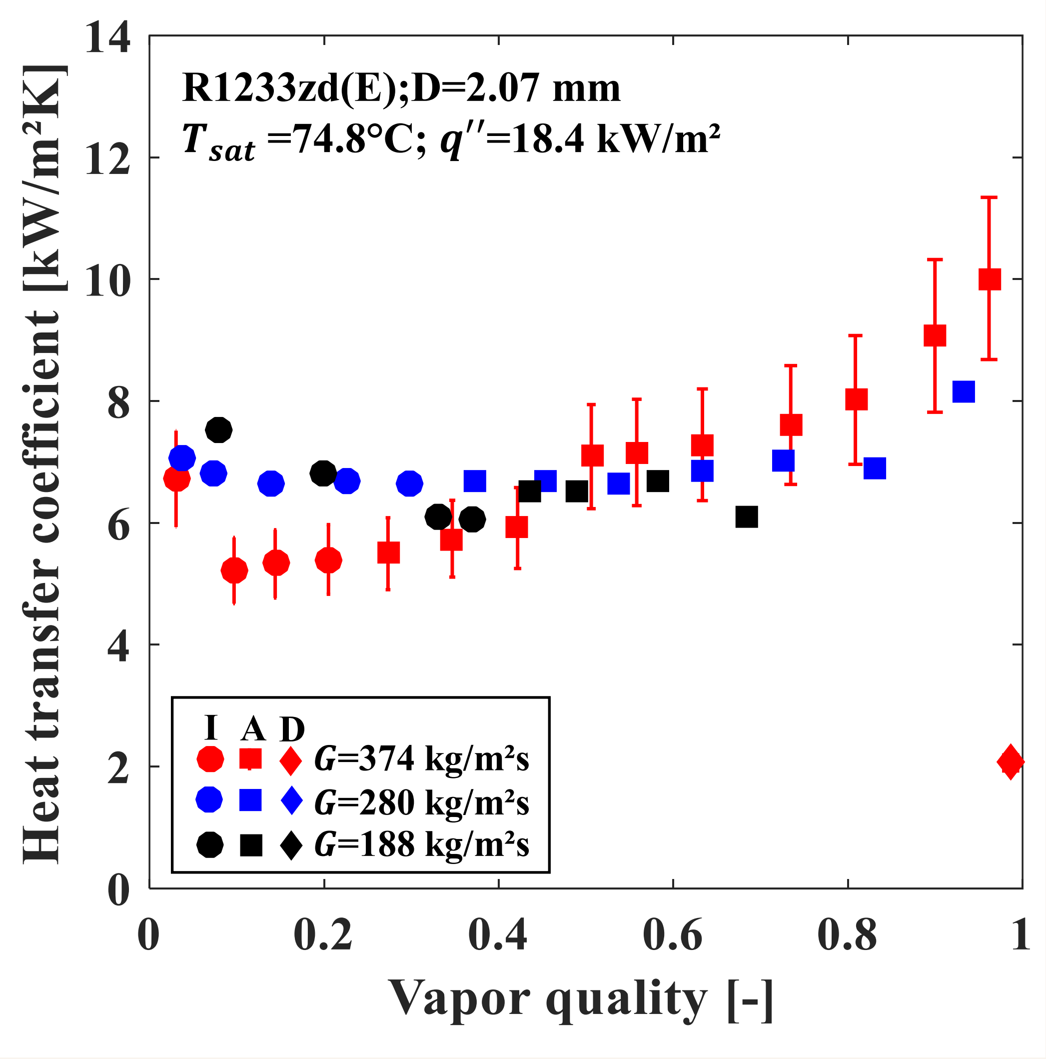
Heat transfer coefficient: Saturation temperature effect

CETHIL data:

High T_{sat}



Low T_{sat}



- The effect of mass velocity strongly depends on the governing heat transfer mechanism

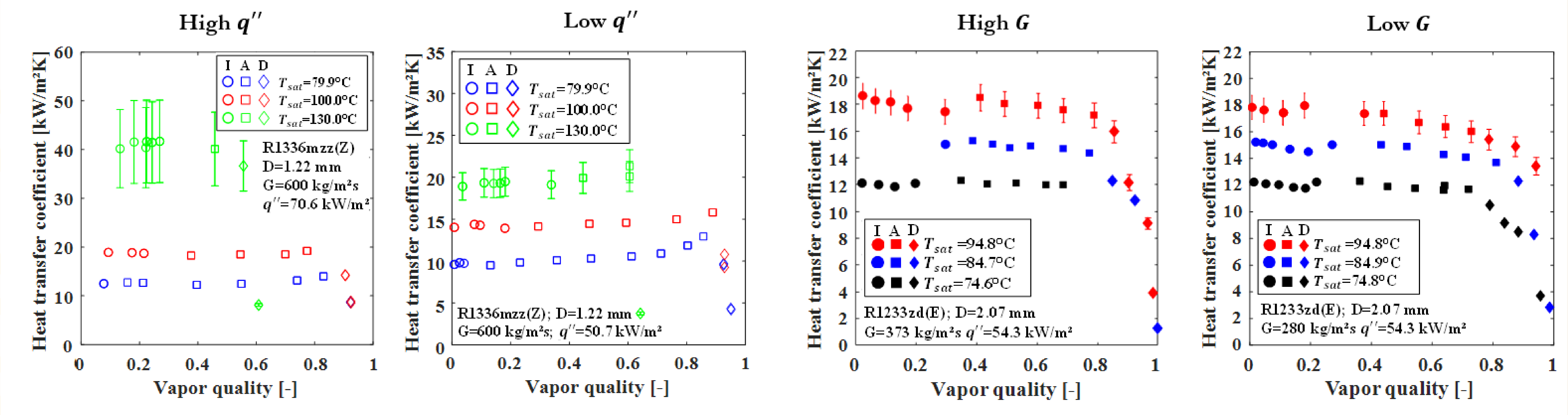
CONVECTIVE DOMINANCE: ↑ G→ ↑ HTC

NUCLEATE BOILING DOMINANCE: ↑ G→ ↓ HTC or = HTC

Results

Heat transfer coefficient: Saturation temperature effect

LETeF data | **CETHIL data**



- Increasing saturation temperature increases the heat transfer coefficient, regardless of vapor quality.

Results

Heat transfer coefficient: Fluid and diameter effects

Fluid effect:

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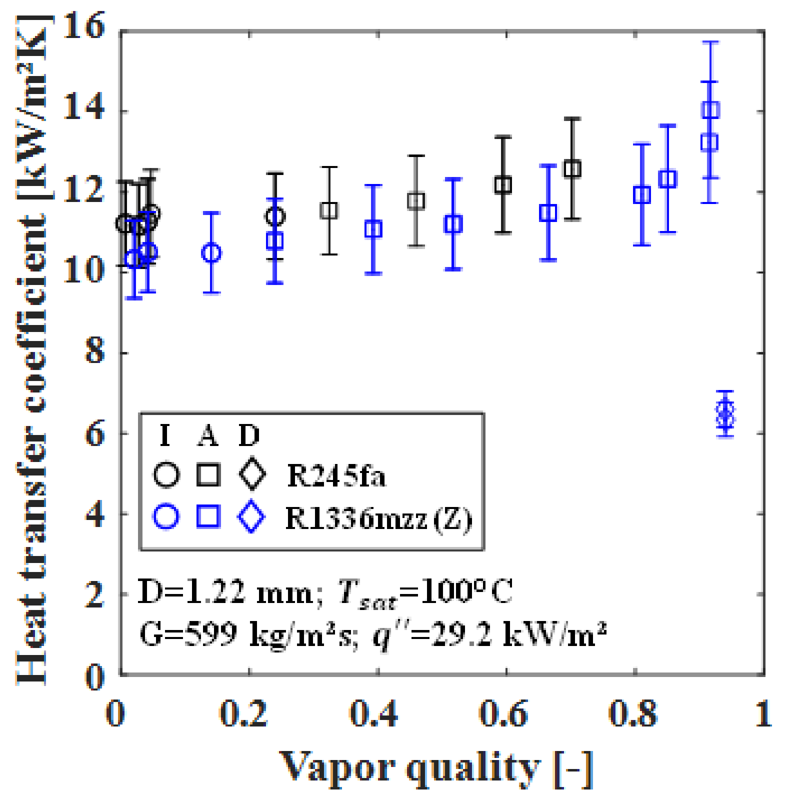
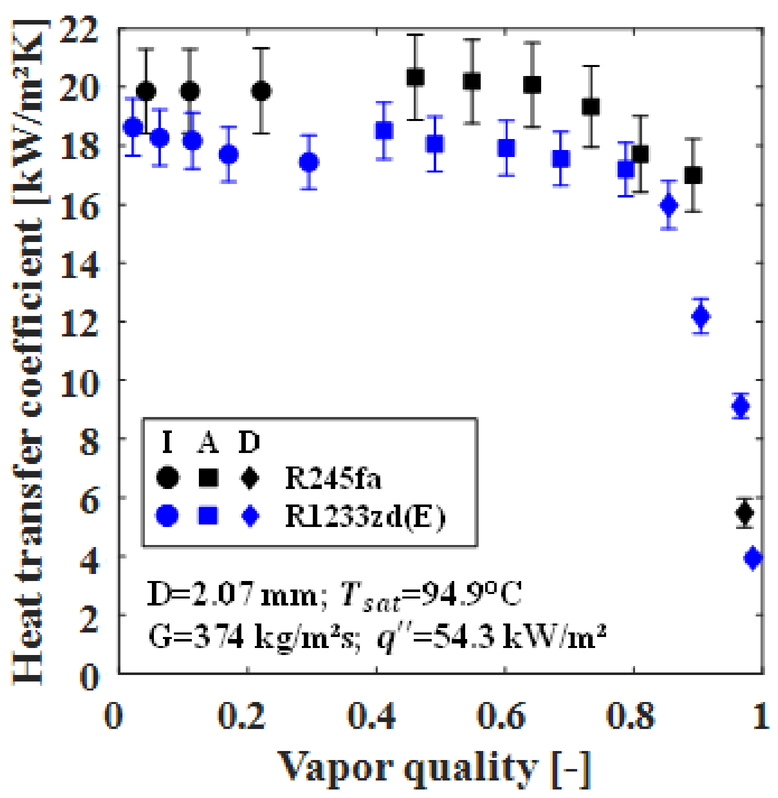
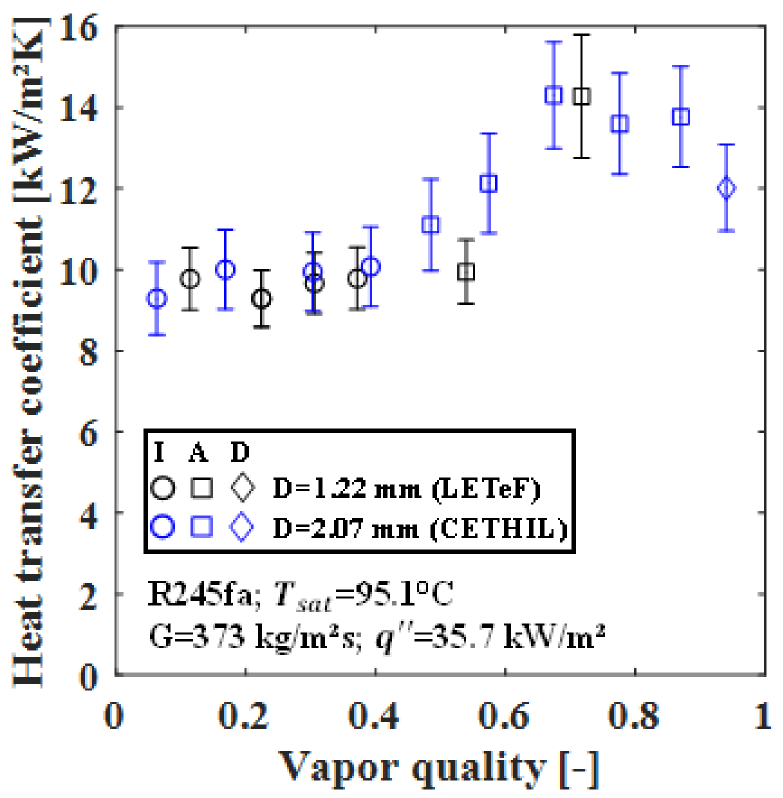
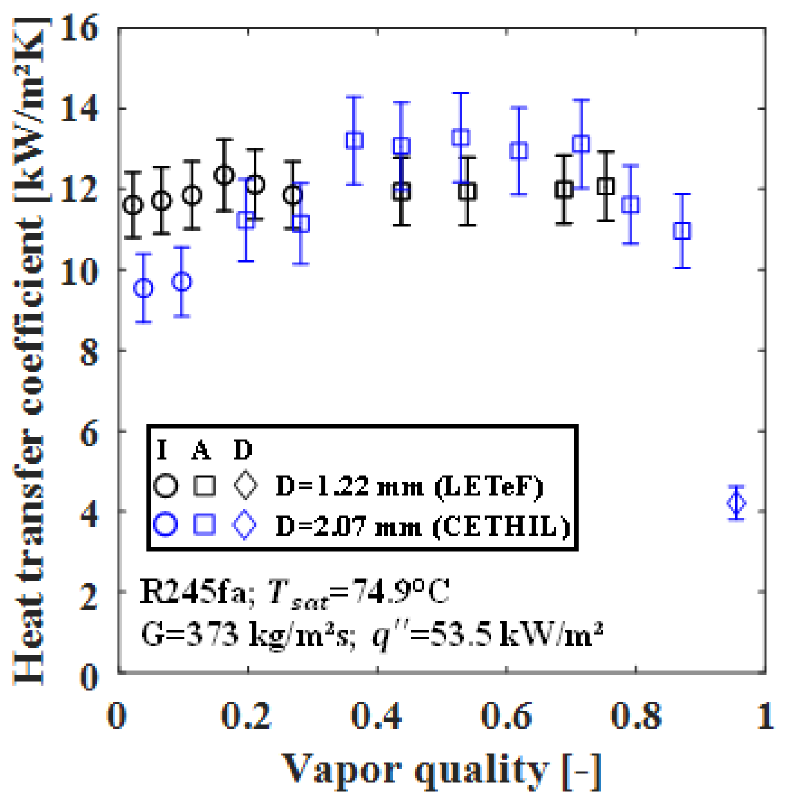


CETHIL



Diameter effect:





Fluid	T_{sat}	p_r	i_{lv}	σ	ρ_v	k_l	$\in_l$
R245fa	95°C	0.31	140 kJ/kg	5.2 mN/m	64 kg/m³	0.066 W/mK	339 Ws$^{0.5}$/m²K
R1336mzz(Z)	95°C	0.22	130 kJ/kg	7.0 mN/m	41 kg/m³	0.058 W/mK	306 Ws$^{0.5}$/m²K
R1233zd(E)	95°C	0.26	146 kJ/kg	6.1 mN/m	50 kg/m³	0.063 W/mK	306 Ws$^{0.5}$/m²K

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Results

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Heat transfer coefficient: Comparison against prediction methods

27 evaluated methods

Empirical

Lazarek e Black (1982)
Tran *et al.* (1996)
Kew and Cornwell (1997)
Sun and Mishima (2009)
Tibiriçá *et al.* (2017)
Keniar *et al.* (2020)

Convective and nucleate boiling superposition

Shah (1982,2017)
Gungor and Winterton (1986,1987)
Jung *et al.* (1989)
Liu e Winterton (1991)
Choi *et al.* (2007)
Saitoh *et al.* (2007)
Ozawa *et al.* (2009)
Ducoulombier *et al.* (2011)
Del Col (2010)
Kim and Mudawar (2013)
Kanizawa *et al.* (2016)
Billiet *et al.* (2018)
Wang *et al.* (2019)
Luo *et al.* (2023)

Phenomenological

Wojtan *et al.* (2005)
Zhang *et al.* (2018)

Pool boiling

Cooper (1983)
Yun *et al.* (2005)
Cheng *et al.* (2006)*

- In general, the heat transfer coefficient prediction methods evaluated were not equally accurate for both channel diameters;
- Progressive loss of accuracy with increasing T_{sat} was identified for most of prediction methods evaluated. The exceptions were those proposed for CO2
- The prediction method of Gungor and Winterton (1986) and its modifications (Wang *et al.*, 2019; Luo *et al.*, 2023) were the most accurate in the saturation temperatures range of 75-100°C, considering both channel diameters

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Results

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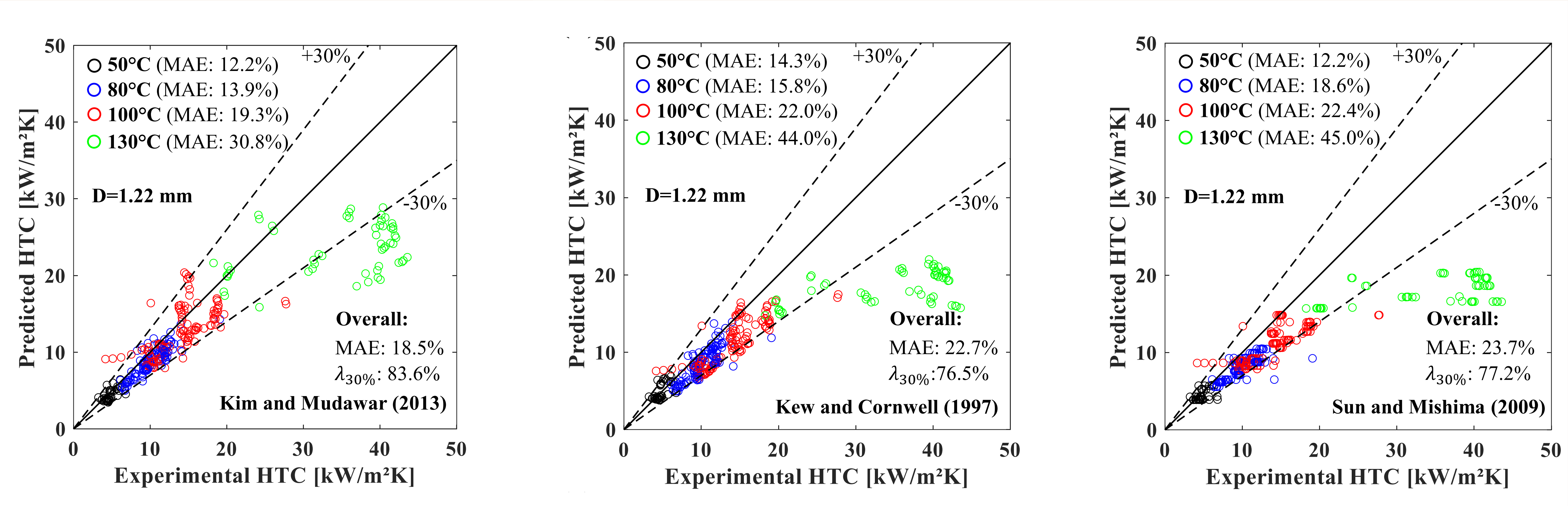
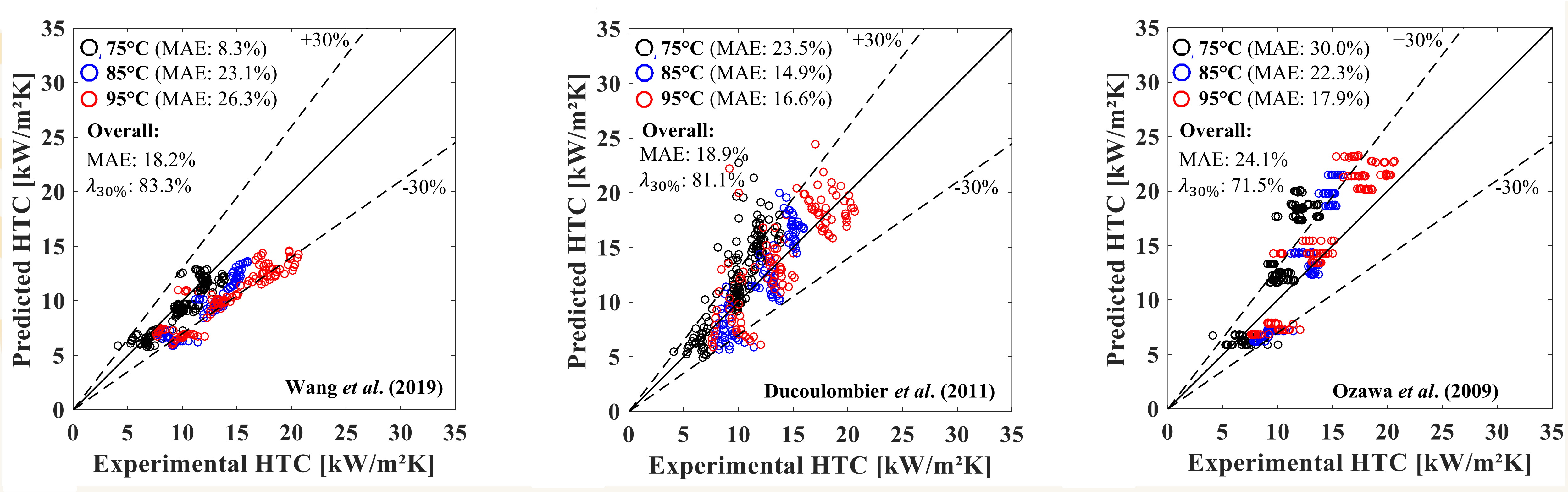
Heat transfer coefficient: Comparison against prediction methods



LETeF database



CETHIL database

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Heat transfer modeling

The heat flux partitioning model:

- RPI model (KRUL & PODOWSKI, 1990):

$$q'' = q''_{evap} + q''_{quench} + q''_{Conv}$$

- Evaporation:

$$q''_{evap} = f_b N'' \frac{\pi}{6} D_b^3 \rho_v i_{lv}$$

f_b: Departure Frequency [1/s]
N'': Nucleation sites density [1/m²]
D_b: Bubble departure diameter [m]

- Quenching:

$$q''_{quench} = f_b N'' K_b \frac{\pi D_b^2}{4} 2\varepsilon_l \sqrt{\frac{t_{wait}}{\pi}} \Delta T_{wall}$$

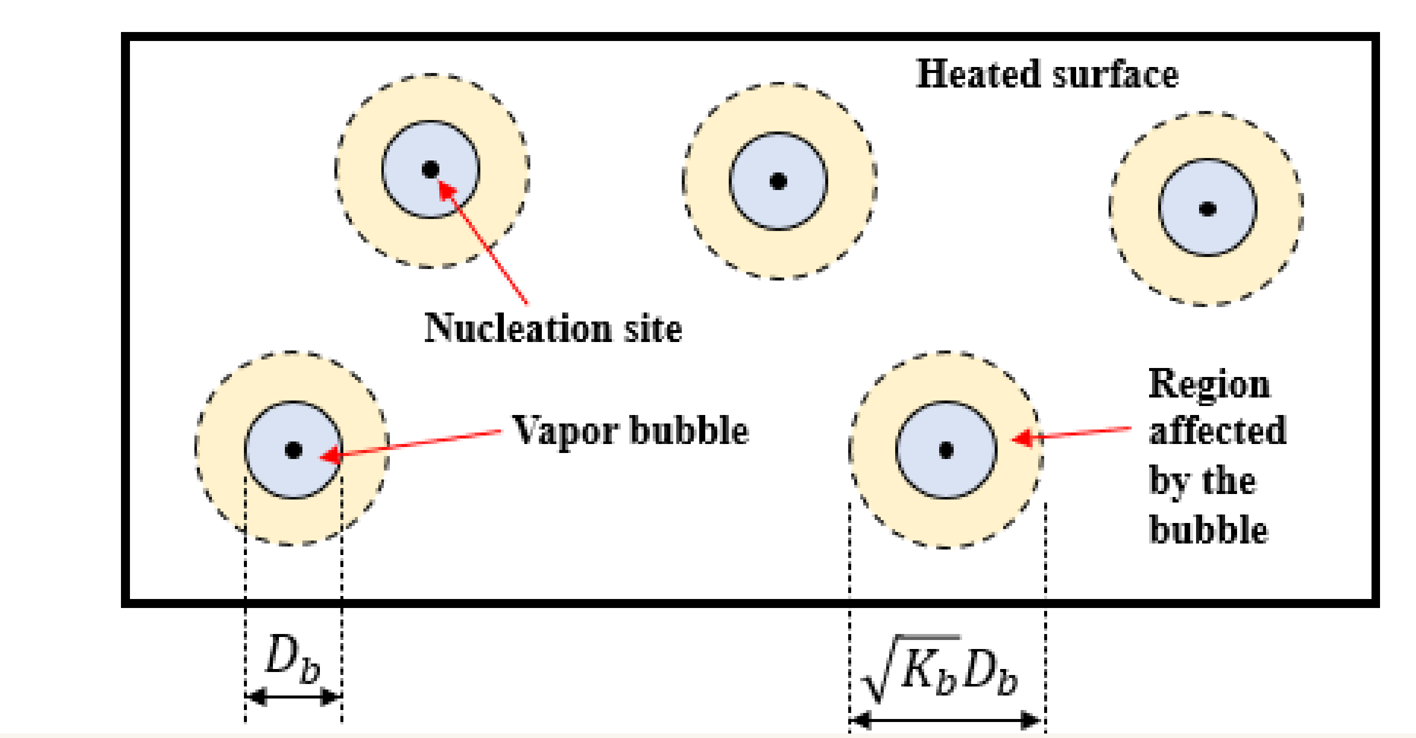
K_b: Area influence factor [-]
t_{wait}: Waiting period [s]
ε_l: Liquid effusivity [W$\sqrt{s}$/m²K]

- Convection:

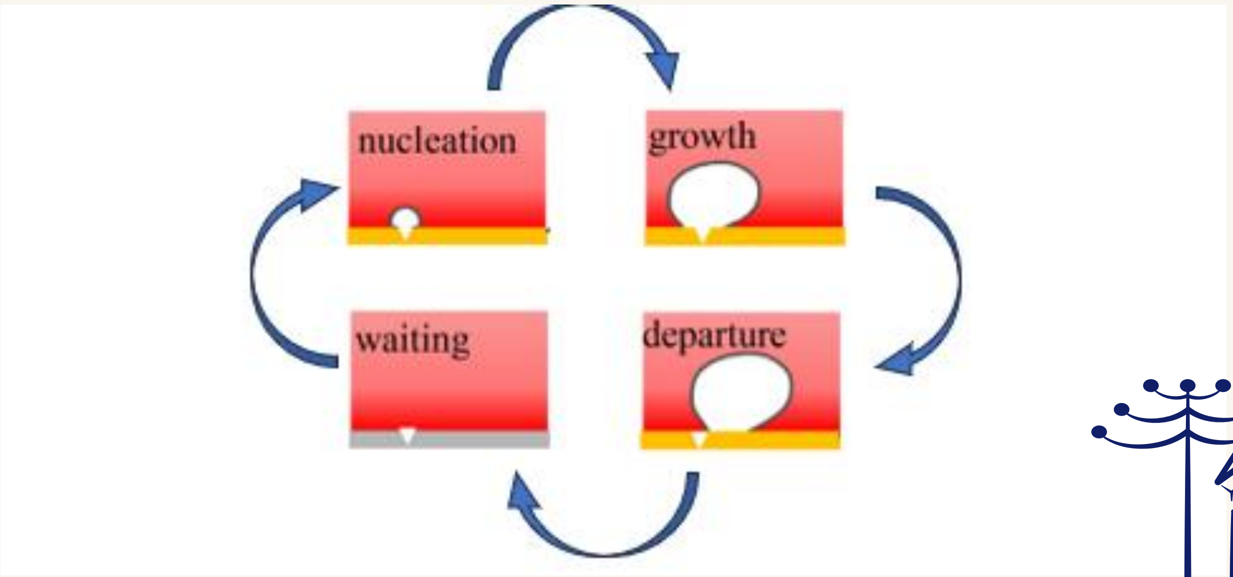
$$q''_{Conv} = \underbrace{\left(1 - N'' K_b \frac{\pi D_b^2}{4}\right)}_{A_{Conv}} h_{Conv} \Delta T_{wall}$$

A_{Conv}: Convection dimensionless area [-]

h_{Conv}: Convection HTC [W/m²K]



Bubble cycle:



Adapted from Zhou *et al.* (2021)

Heat transfer modeling

The heat flux partitioning model:

- **RPI model (KRUL & PODOWSKI, 1990):**

$$q'' = q''_{evap} + q''_{quench} + q''_{Conv}$$

- **Evaporation:**

$$q''_{evap} = f_b N'' \frac{\pi}{6} D_b^3 \rho_v i_{lv}$$

f_b: Departure Frequency [1/s]
N'': Nucleation sites density [1/m²]
D_b: Bubble departure diameter [m]

- **Quenching:**

$$q''_{quench} = f_b N'' K_b \frac{\pi D_b^2}{4} 2\varepsilon_l \sqrt{\frac{t_{wait}}{\pi}} \Delta T_{wall}$$

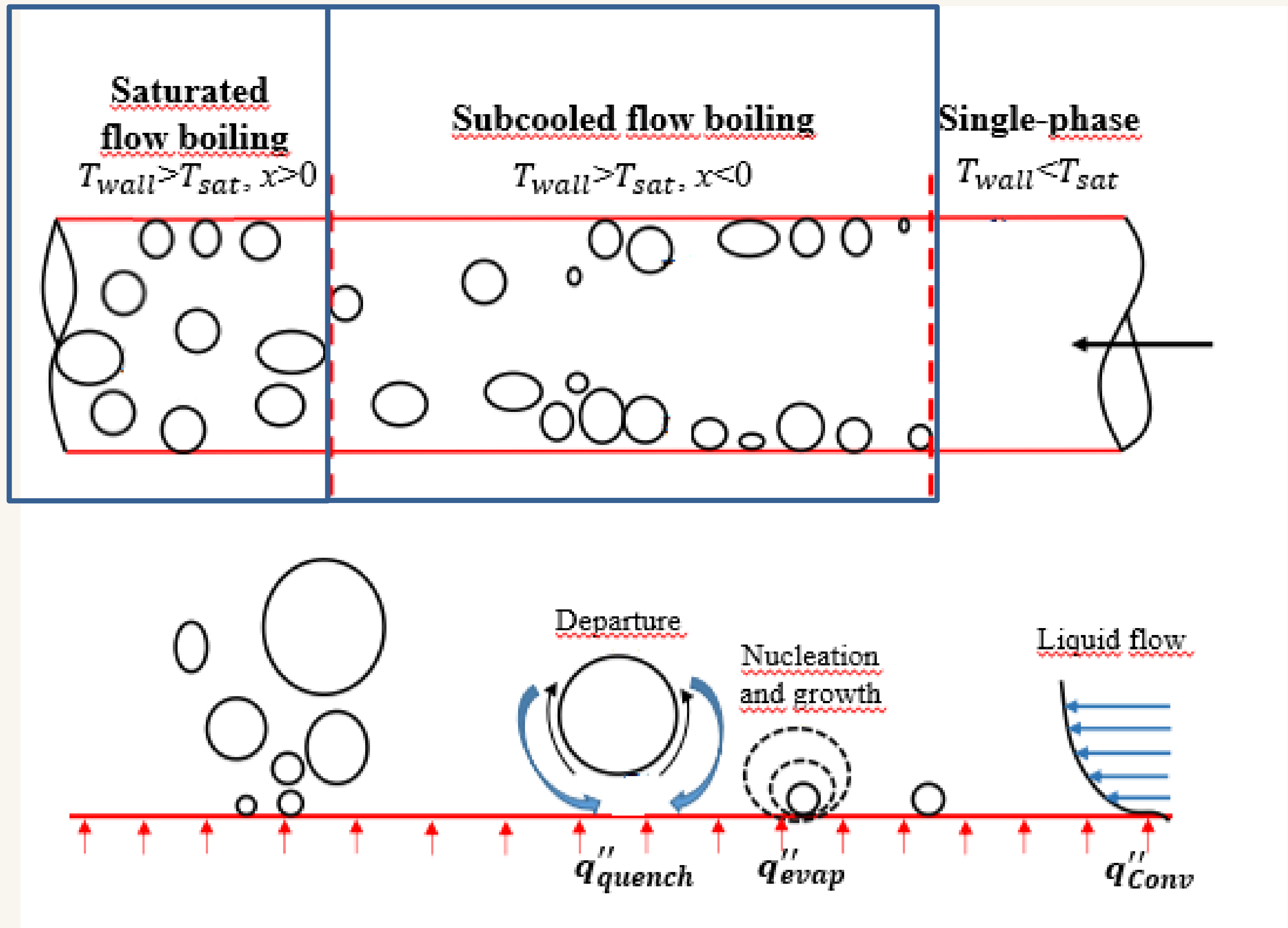
K_b: Area influence factor [-]
t_{wait}: Waiting period [s]
ε_l: Liquid effusivity [W$\sqrt{s}$/m^{2}K]

- **Convection:**

$$q''_{Conv} = \underbrace{\left(1 - N'' K_b \frac{\pi D_b^2}{4}\right)}_{A_{Conv}} h_{Conv} \Delta T_{wall}$$

A_{Conv}: Convection dimensionless area [-]

h_{Conv}: Convection HTC [W/m²K]

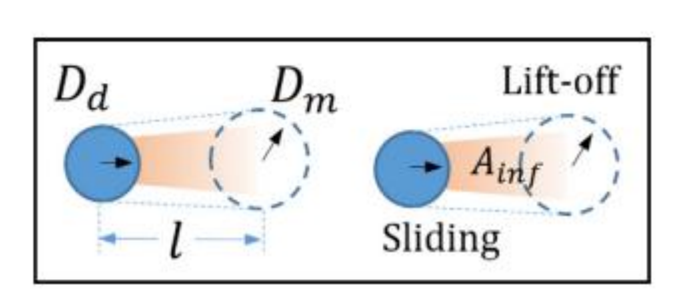


Adapted from Gu *et al.* (2017)

Heat transfer modeling

Hypothesis of the proposed model

- Saturated flow boiling;
- Annular flow pattern;
- Uniform HTC along the tube perimeter: $\bar{h}$;
- Uniform wall temperature along the tube perimeter;
- Interfacial disturbance waves neglected;
- Additional heat flux partitioning mechanisms neglected: sliding or lift-off induced transient conduction (WANG *et al.*, 2023);
- Superposition of bubbles influence area neglected



Hoang et al. (2017)

Heat transfer modeling

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Bubble dynamic parameters

Departure diameter

Pioro *et al.* (2004)

$$D_b = \frac{8.13}{10^6} p_r^{-1.09}$$

Valid for:
- Pool boiling
- p_r=0.0003-0.73
- Different surface materials
- Working fluids including water, organic refrigerants, ethanol and others

Departure frequency

Tibiriçá and Ribatski (2014)

$$D_b f_b^{0.5} = 3.457 * 10^{-3}$$

Valid for:
- Subcooled flow boiling
- Horizontal circular channels
- D=0.4-2 mm
- p_r=0.05-0.19
- R134a and R245fa

Nucleation sites density

Yang *et al.* (2016):

$$N'' = 2800 \Delta T_{wi}^{2.66}$$

Valid for:
- Subcooled flow boiling
- Vertical rectangular channels
- ΔT_{wall}<11.2°C
- Water, atmospheric pressure

Proposed modification:

$$N'' = a * \Delta T_{wi}^{b} * p_r^{c} * Re_{l0}^{d}$$

$a = 1.801 * 10^{10}\ [m^{-2}K^{-1.19}]$,
$b = 1.19$ [-], $c = 2.2$ [-],
$d = -0.67$ [-]

Waiting time

Murallidharan *et al.* (2016):

$$t_{wait} = 0.8\left(\frac{1}{f_b}\right)$$

This value is considered in CFD simulation codes, including in the high-pressure model proposed by Murallidharan *et al.* (2016)

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Heat transfer modeling

Convection heat transfer coefficient

- Based on the model proposed by Cioncolini and Thome (2011) for annular flow.
- This model accounts only for convective effects:

$$Nu_{l,film} = \frac{h_{Conv}\,\delta}{k_l} = 0.0776\delta^{+0.9} Pr_l^{0.52}$$

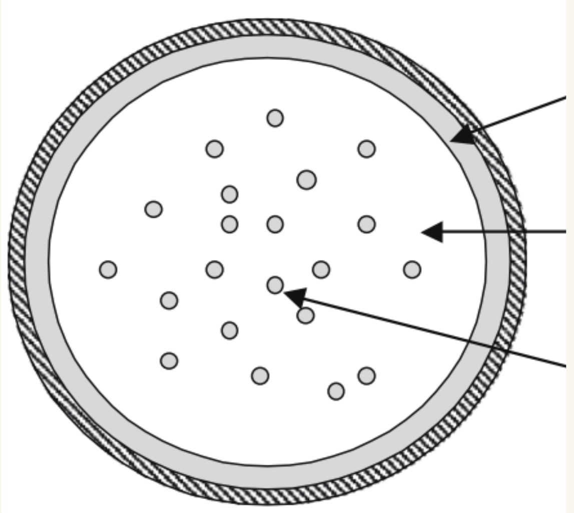
$$\delta^{+} = \frac{\delta}{\frac{\mu_l}{\rho_l V^*}} \qquad V^* = \sqrt{\tau_w/\rho_l} \qquad \tau_w = \left(-\frac{dp}{dz}\right)_f \frac{D_i}{4}$$

$$\delta^{+} = \max\left[\sqrt{\frac{2(1-ent_f)(1-x)GD_i}{4\mu_l}}, 0.066\frac{(1-ent_f)(1-x)GD_i}{4\mu_l}\right]$$

Entrainment fator (ent_f) according to Cioncolini and Thome (2010)
Frictional pressure gradient according to Cioncolini and Thome (2009)

$$\delta = \frac{1}{2}\left(D_i - \sqrt{\frac{4A_{film}}{\pi} - D_i^2}\right)$$

Proposed modifications:



$$\boldsymbol{A_{film} = (1-\gamma)(1-\alpha)A}$$

$$\boldsymbol{A_v = \alpha A}$$

$$\boldsymbol{A_{l,ent} = \gamma(1-\alpha)A}$$

Adapted from Cioncolini and Thome (2010)

Liquid droplet hold-up: $\gamma = ent_f\left(\frac{\alpha}{1-\alpha}\right)\left(\frac{1-x}{x}\right)\left(\frac{\rho_v}{\rho_l}\right)$
slip effects neglected

Entrainment factor (ent_f) according to Wang *et al*. (2020)
Void fraction (α) according to Kanizawa and Ribatski (2016)
Frictional pressure gradient according to Müller-Steinhagen and Heck (1987)

$$Nu_{l,film} = \frac{h_{Conv}\,\delta}{k_l} = m\,\delta^{+0.9} Pr_l^{0.52}$$

m=0.0647 [-]

Heat transfer modeling

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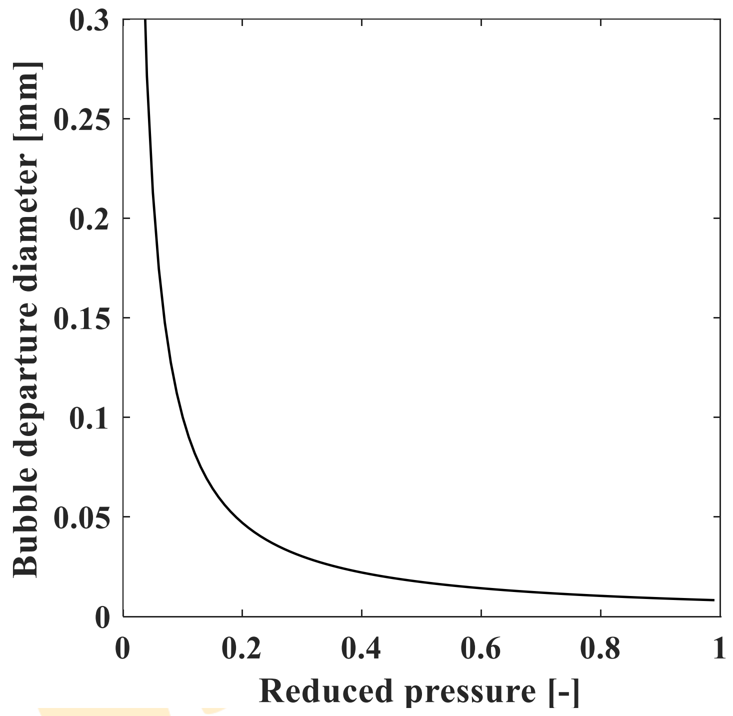
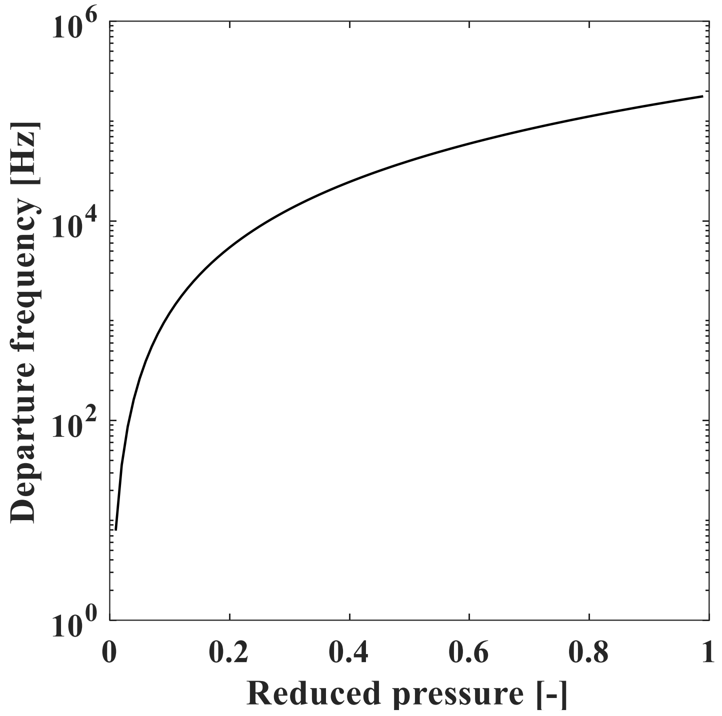
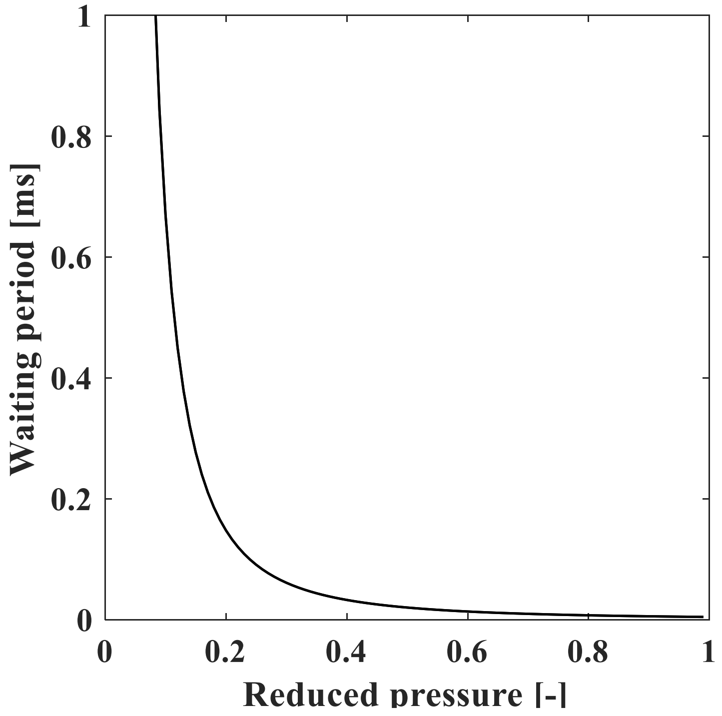
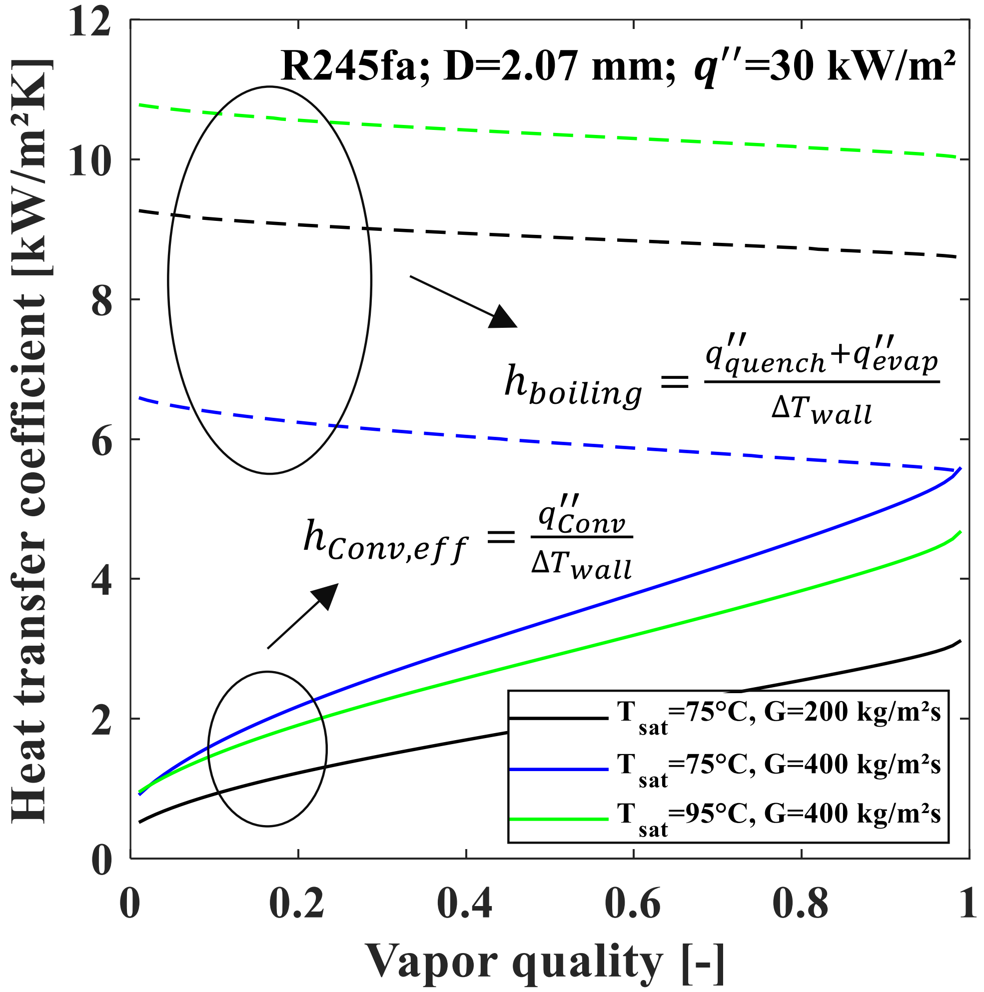
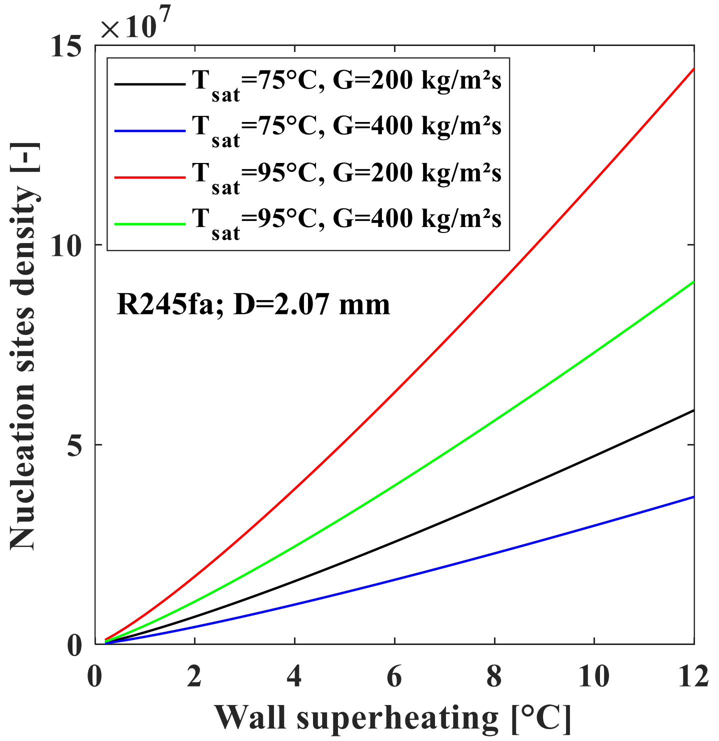
Model parameters

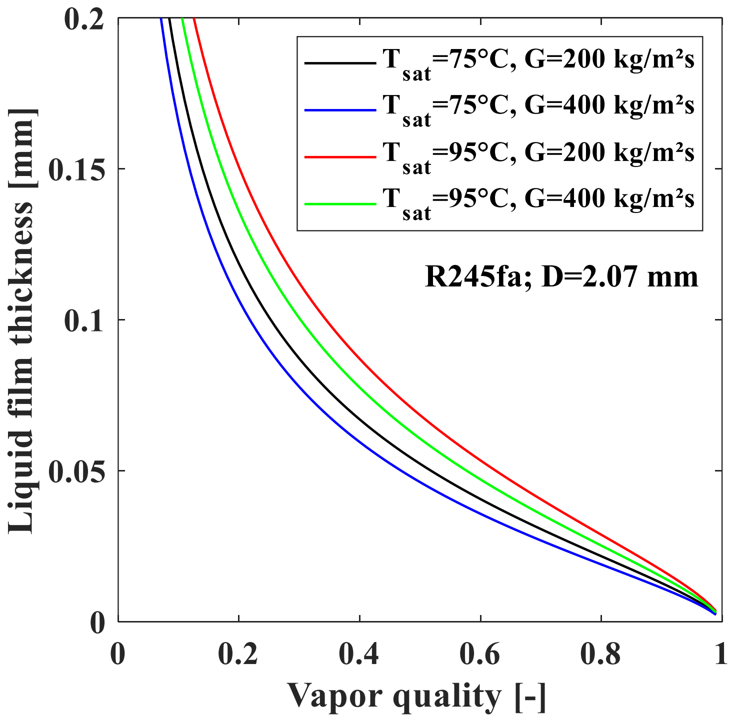












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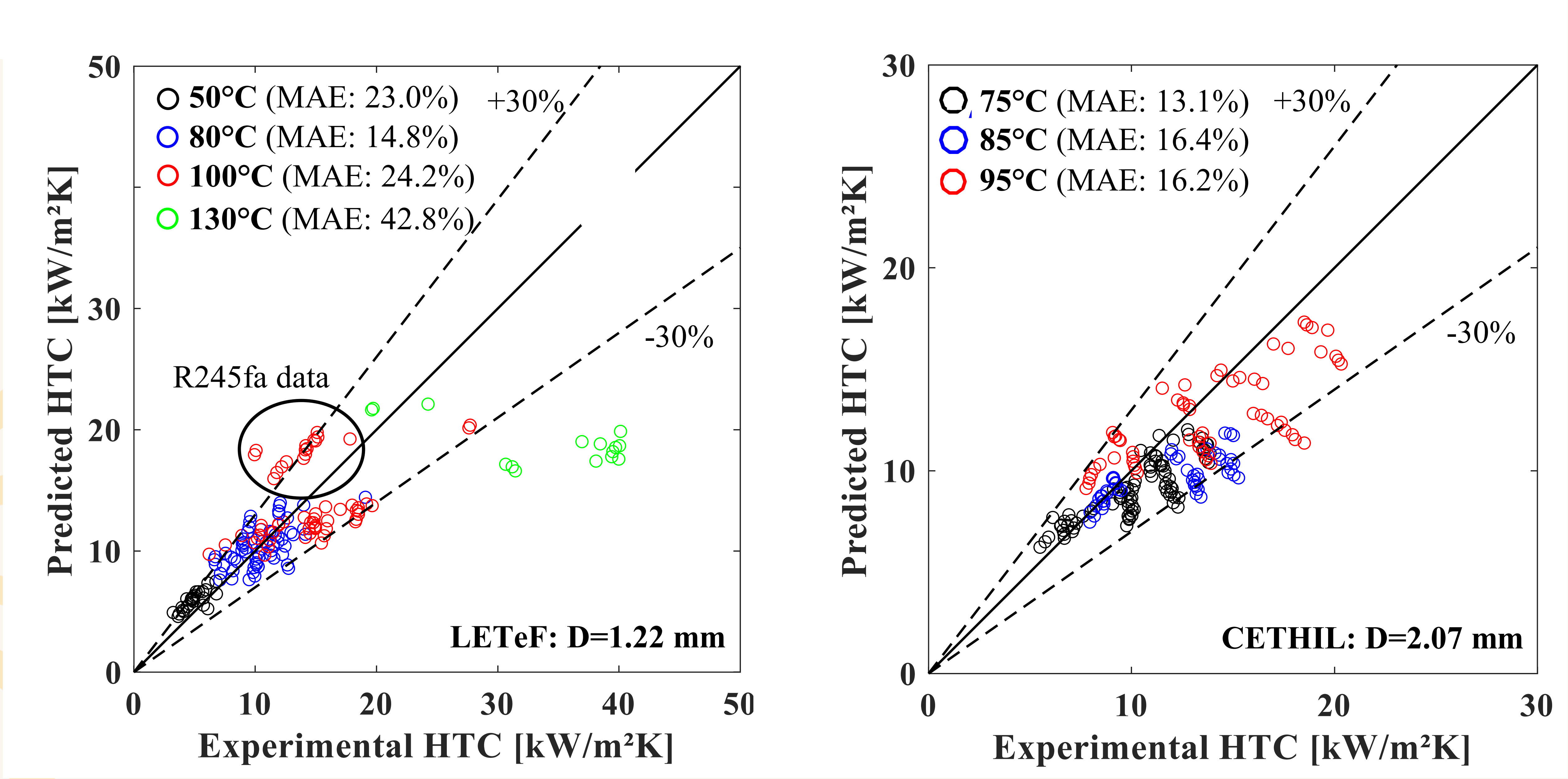
Heat transfer modeling

Comparison against database



- **Overall:**

  MAE: 18.2%, $\lambda_{30\%}$: 86.0%

- **LETeF:**

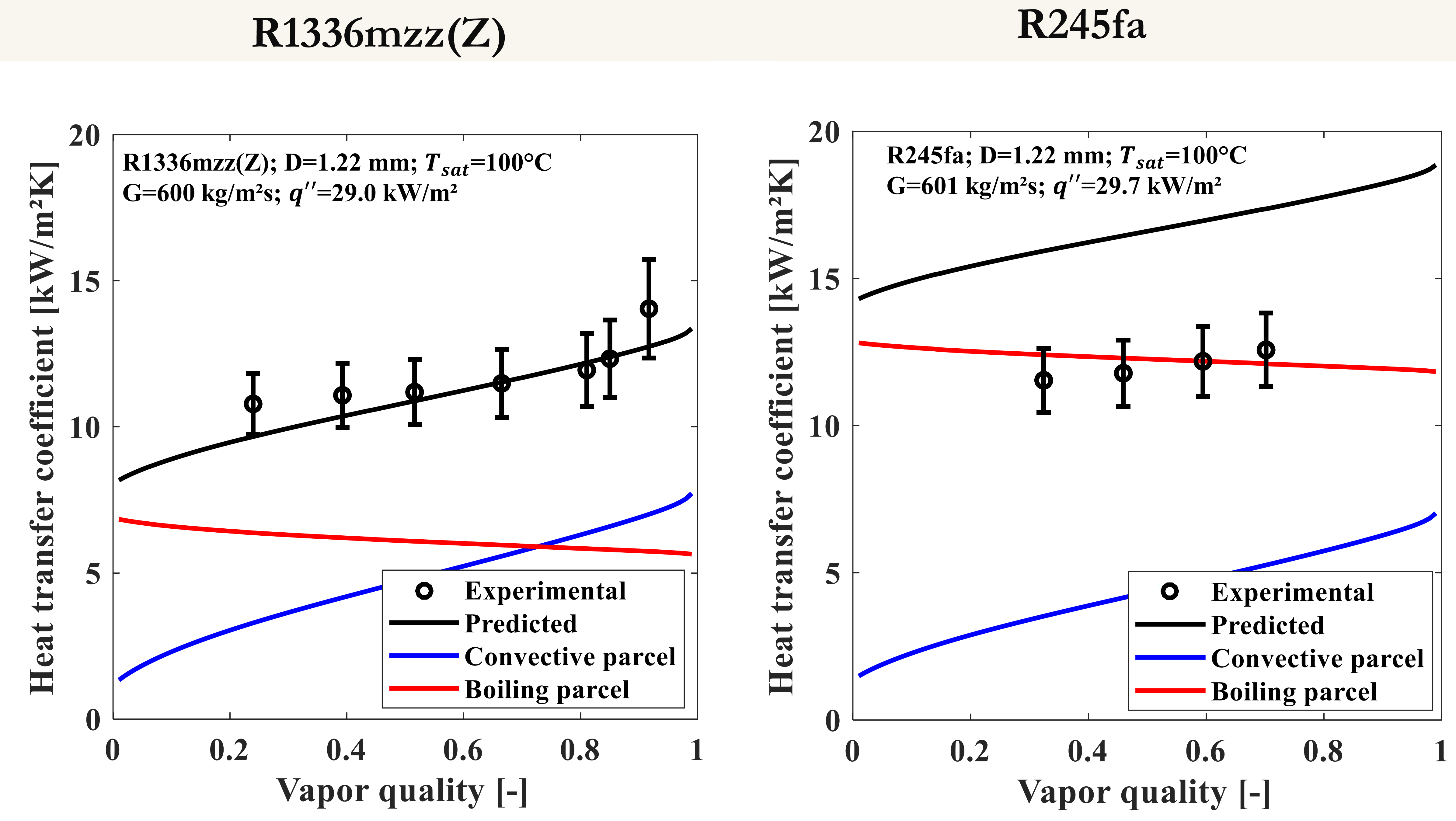
  MAE: 22.0%, $\lambda_{30\%}$: 78.7%

- **CETHIL:**

  MAE: 14.9%, $\lambda_{30\%}$: 92.3%

Heat transfer modeling

Comparison against database



- **Overall:**

  MAE: 18.2%, $\lambda_{30\%}$: 86.0%

- **LETeF:**

  MAE: 22.0%, $\lambda_{30\%}$: 78.7%

- **CETHIL:**

  MAE: 14.9%, $\lambda_{30\%}$: 92.3%

Heat transfer modeling

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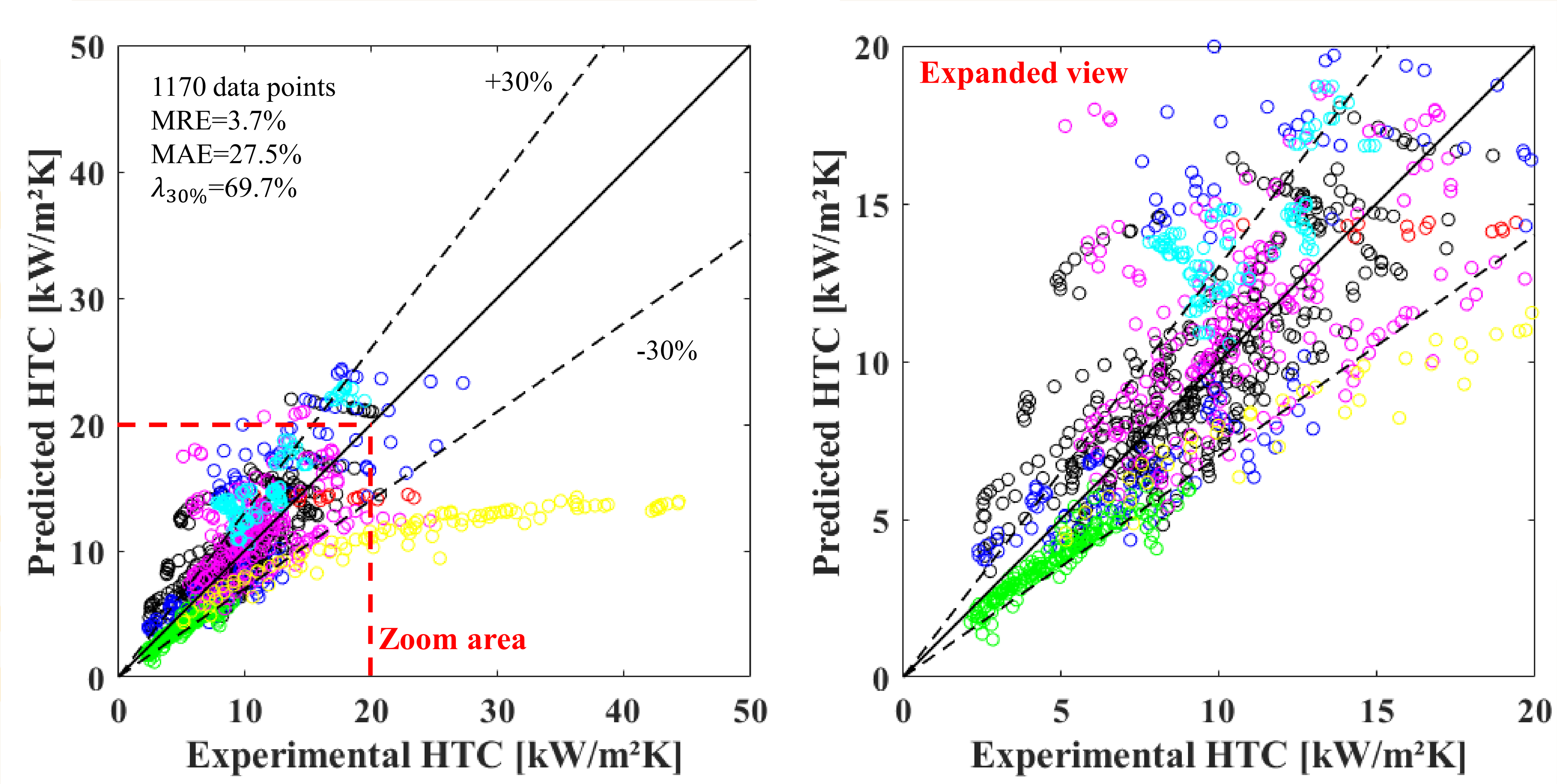
Comparison against independent databases



- Charnay *et al*. (2014, 2015)
- Billiet *et al*. (2018)
- Pysz *et al*. (2023), Pysz and Mikielewicz (2023)
- Roldão and Tibiriçá (2024)
- Mawatari and Mori (2016)
- Lillo *et al*. (2019)
- Bediako *et al*. (2024)

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Conclusions

- Intermittent-annular transition was anticipated with increasing G and reducing T_{sat}. Prediction methods underpredicted the transition vapor quality at high T_{sat};
- The onset of liquid film dryout was characterized through temperature measurements and it was anticipated with increasing G, T_{sat} and q'';
- The frictional pressure gradient increased with increasing G, and reducing D and T_{sat};
- Two-phase frictional pressure gradient prediction methods were not equally accurate for both channel diameters;
- Increasing T_{sat} did not compromise the accuracy of the pressure drop prediction methods;
- The heat transfer coefficient increased with increasing T_{sat} and q'', while the effect of G was strongly dependent on the governing heat transfer mechanism;
- The channel diameter exhibited only a marginal influence on the heat transfer coefficient;
- In general, heat transfer coefficient prediction methods were not equally accurate for both channel diameters;
- The increase in T_{sat} reduced the accuracy of the HTC prediction methods evaluated;

Conclusions

- Under similar operating condition, R245fa showed equivalent or slightly higher HTCs and lower pressure drops compared to its low-GWP substitutes;
- A new analytical-empirical model, based on the heat flux partitioning approach, was proposed to predict the flow boiling heat transfer coefficient during annular flow in small diameter channels;
- The proposed model achieved the lowest MAE for the overall experimental database obtained in the present study, 18.2%, predicting 86.0% of the data within error bands of ±30%. The proposed model also showed reasonable accuracy for independent datasets from literature;
- The incorporation of a convective heat transfer model based on the liquid film thickness variation into the heat flux partitioning approach seems to be a promising methodology during annular flow pattern.

Recommendations for future studies

- Extend the reduced pressure range evaluated, reaching near-critical values
- Perform critical heat flux experiments;
- Develop flow-pattern transition criteria valid under the conditions evaluated in the presente study;
- Evaluate bubble dynamics and two-phase flow constructive parameters (IR + high-speed imaging);
- Improve the proposed model incorporating correlations valid for saturated flow boiling at high temperatures;
- Extend the application of the proposed model to intermittent and dryout patterns, coupling it with a flow pattern prediction method valid flow boiling of refrigerants at high T_{sat} in small diameter channels.

Acknowledgments

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Thank you!!!!!!!!!