COOLING:
COLD PLATE

System-level Challenges and Considerations in Transitioning from Single-Phase to Two-Phase Cooling

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Transitioning to Two-phase Cooling Requires More Than a Component Swap

1. **Working Fluid**

- Refrigerant trade-offs drive system design decisions

2. **Flow Instability**

- Boiling introduces flow excursion and oscillation risks

3. **Flow Architecture**

- Series and parallel architectures impact performance and flow distribution

4. **Cooling Loop**

- Vapor management impacts piping, pumps, and loop integration

Why Two-phase Cooling?

- Potentially higher heat transfer and better temperature uniformity
- Potentially lower flow rate requirement and pumping power
- Lower risk of damage due to leakage
- Lower corrosion and bio-growth concerns

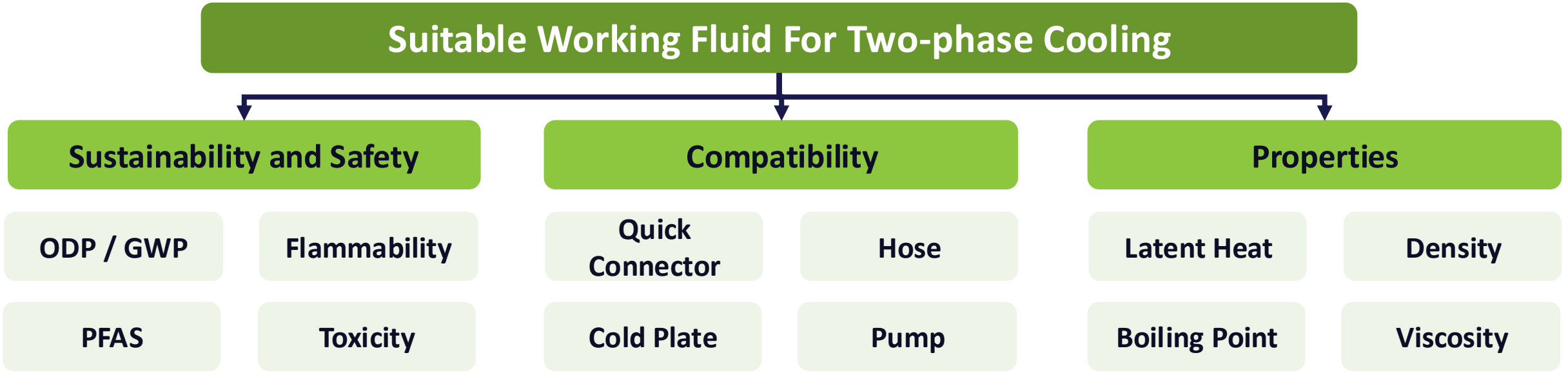
Phase	Heat Transfer Process	Heat Transfer Coefficient* ($W/m^2 \cdot K$)	Heat Absorption Capacity** (kJ/kg)
Single: Liquid	Forced Convection	100-40,000	41.8
Two: Liquid to Vapor	Convective Boiling	2,500-100,000	2,257

*Ref: Incropera, F.P. et al., Fundamentals of Heat and Mass Transfer

**Water as an example. Single-phase: $C_p \times \Delta T$ (C_p = 4.18 kJ/kg·K, ΔT = 10°C); Two-phase: h_{fg} (latent heat of vaporization)

Two-phase Working Fluid Considerations

- Ideal refrigerant: zero ODP, GWP < 1, non-PFAS, ASHRAE A1 classification
- Compatibility between the working fluid and wetted materials must be independently verified
- Lower liquid-to-vapor density ratio may enhance flow stability and performance



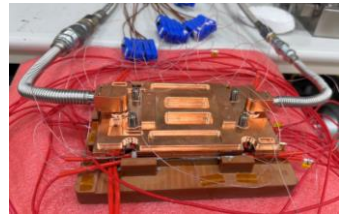
Common Two-Phase Cooling Working Fluids

- Working fluids can be broadly categorized into medium-pressure refrigerants (R134a, R1234yf, R515B) and low-pressure refrigerants (R1233zd(E))
- R515B and R1233zd(E) are key candidates in current data center two-phase cooling development and applications

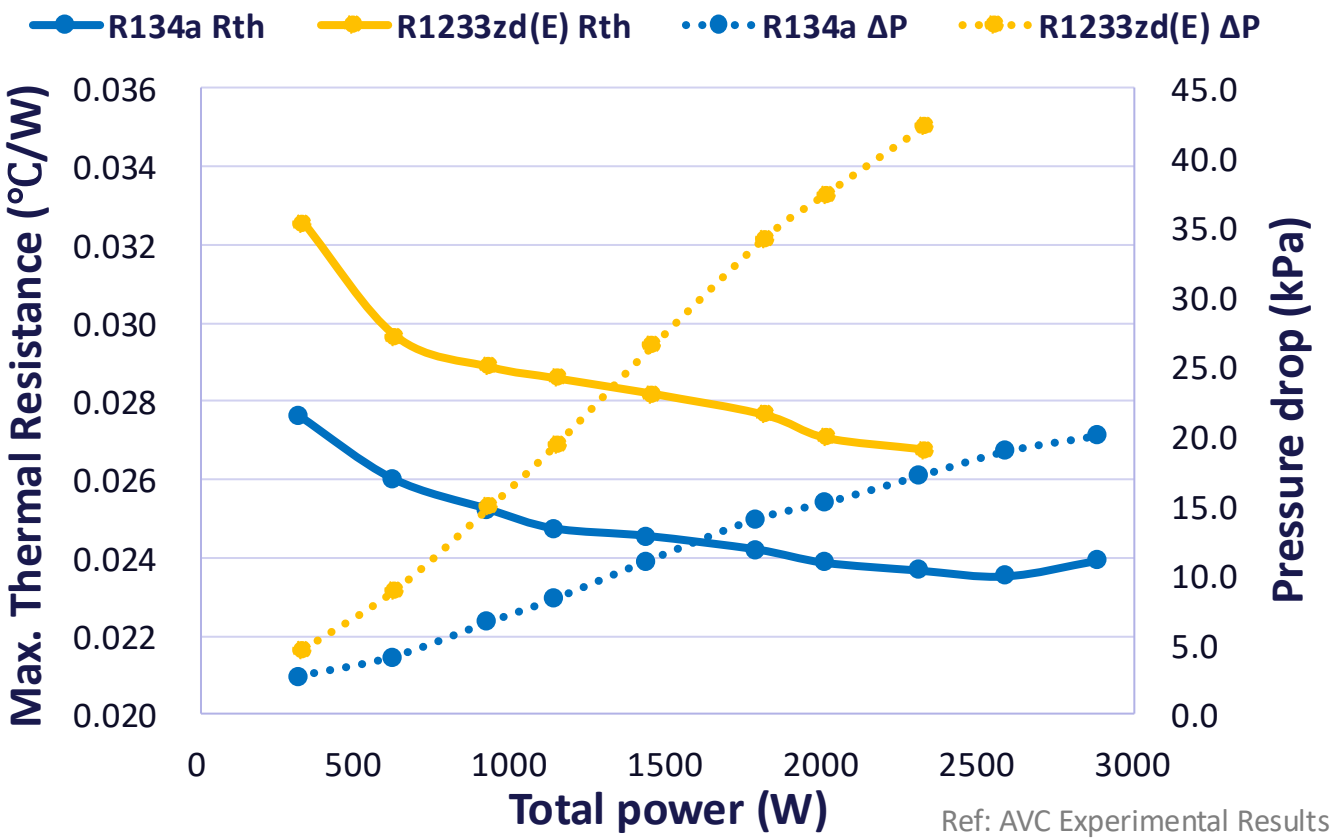
Refrigerant	Boiling Point @ 1atm (℃)	Saturated Pressure @ 45℃ (kPa)	Liquid/Vapor Density @ 45℃ (kg/m^3)	Latent heat @ 45℃ (kJ/kg)	ASHRAE Classification	IPCC AR5 GWP
R134a	-26	1155	1127/57.4	158	A1	1300
R1234yf	-29	1150	1013/66.2	128	A2L	<1
R515B	**-19**	**871**	**1109/47.9**	**146**	**A1**	**299**
R1233zd(E)	**18**	**251**	**1213/13.5**	**180**	**A1**	**1**

Medium vs. Low-Pressure Working Fluid Experimental Comparison

- R134a showed better thermal performance and lower pressure drop than R1233zd(E) under the same primary-side conditions and with the same cold plate



Parameter	R-134a (Medium-Pressure)	R-1233zd(E) (Low-Pressure)
Liquid/Vapor Density (kg/m³)	1127 / 57.4	1213 / 13.5
Latent Heat (kJ/kg)	158	180
Liquid Thermal Conductivity (mW/m·K)	72.6	76.9
Vapor Dynamic Viscosity (μPa·s)	12.6	11.0

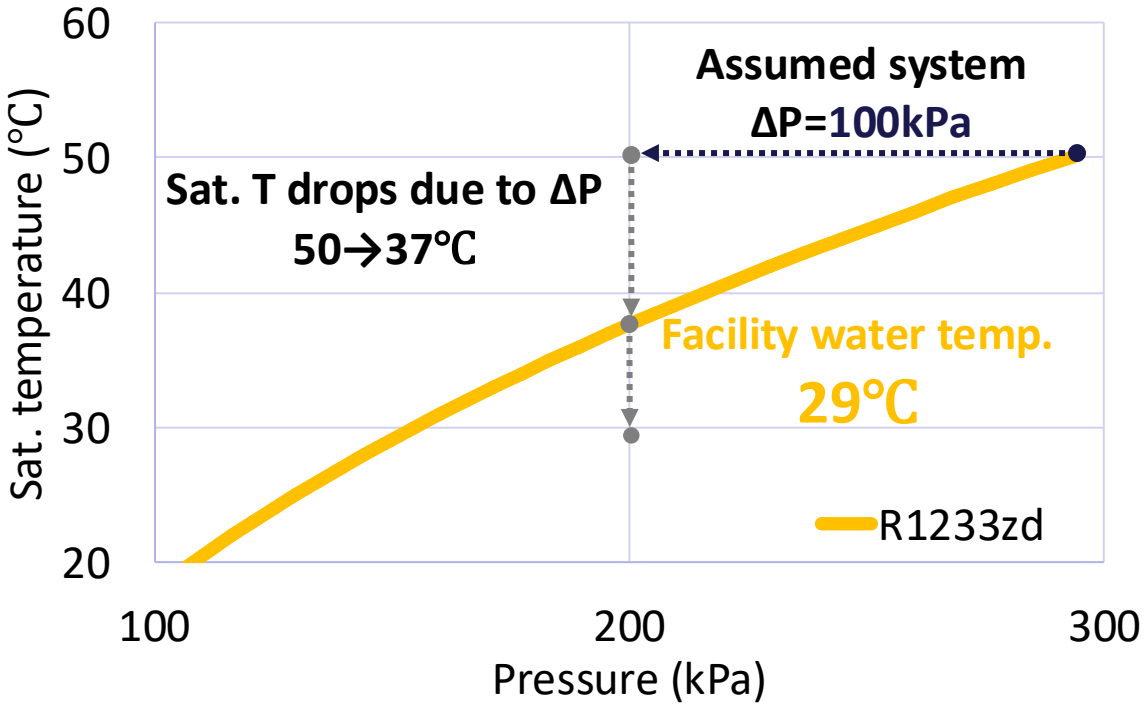


Ref: AVC Experimental Results
- Both at 0.7LPM

Technical Consideration 1: System Pressure Effect

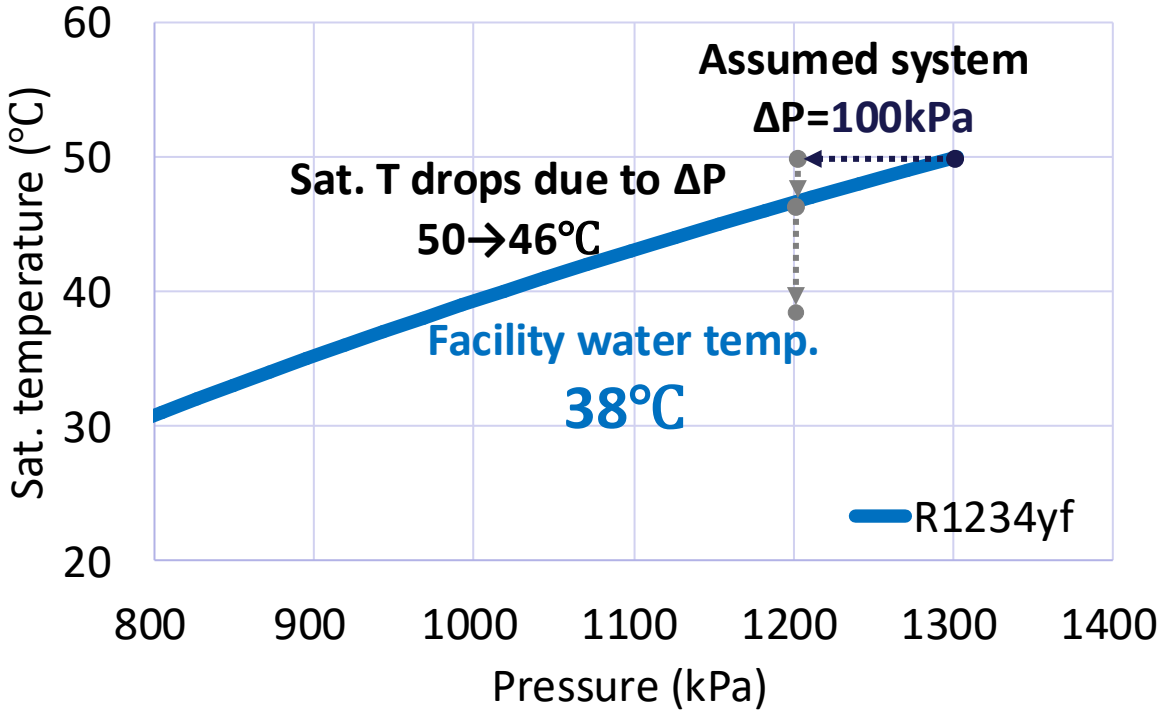
Low-pressure Refrigerant

- More sensitive to pressure drop, requiring stricter facility water conditions
- Lower component pressure rating required



Medium-pressure Refrigerant

- More tolerant of pressure drop, enabling easier facility water integration
- Higher component pressure rating required

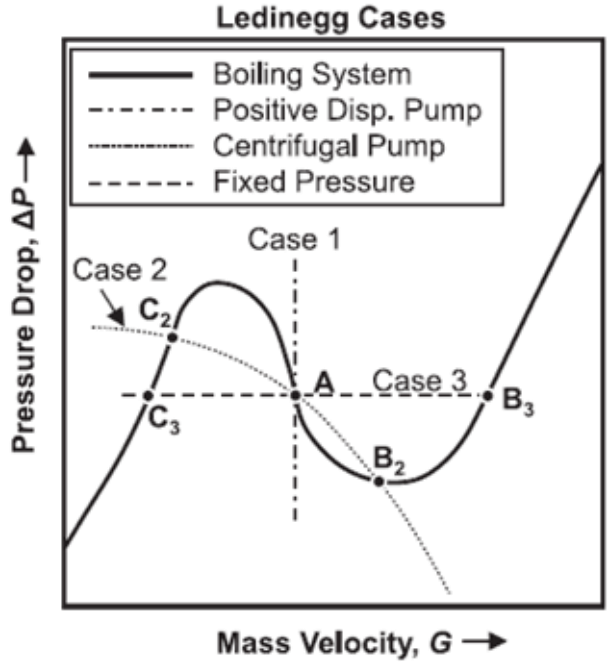


Technical Consideration 2: Flow Instability

- Two-phase flow instability may occur across multiple scales, from cold plate channels to trays and racks
- Potential mitigation approaches include inlet flow restrictors, expanded outlet channels/piping, and optimized manifold designs

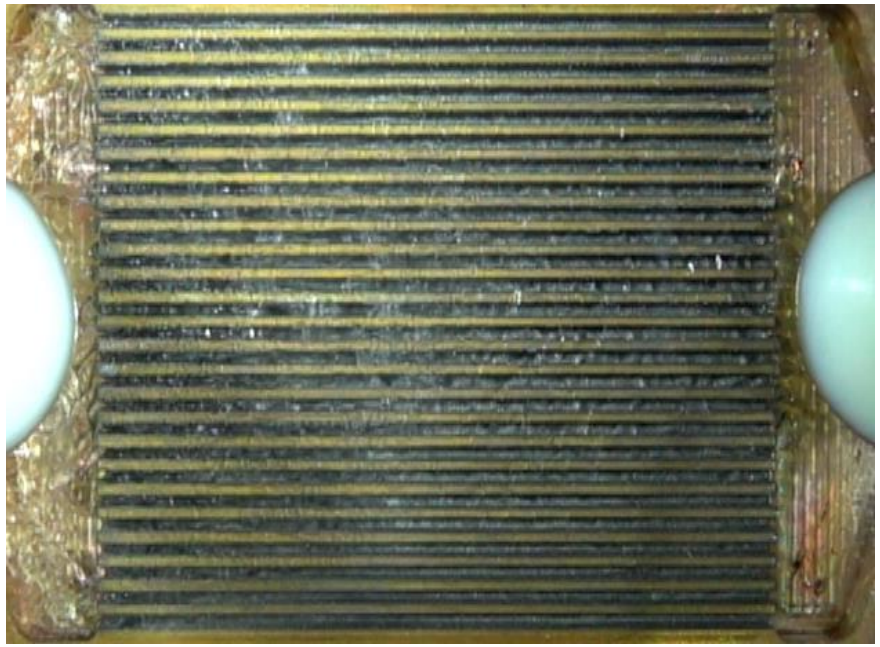
Ledinegg Instability

Negative P-Q curve → Flow excursion



Elongated Bubble Oscillation

Confined bubble growth → Flow Oscillation

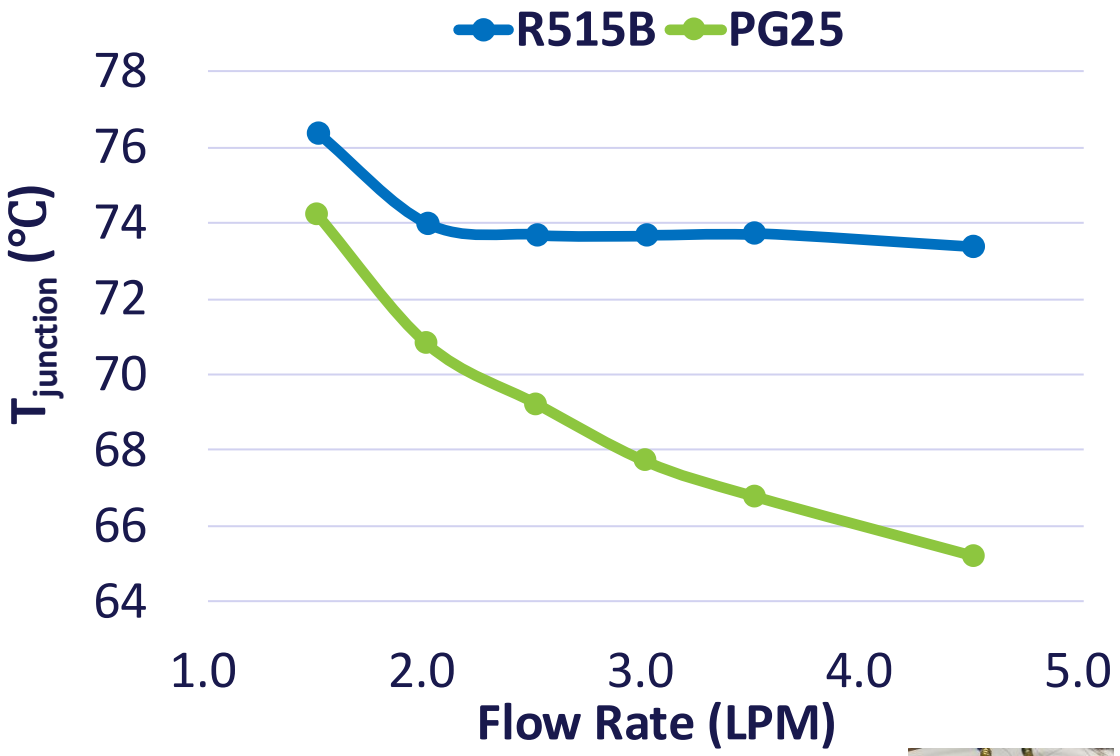


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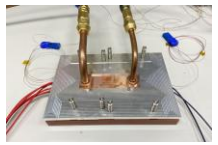
1. L. E. O'Neill and I. Mudawar, Review of two-phase flow instabilities in macro- and micro-channel systems, International Journal of Heat and Mass Transfer 157 (2020) 119738
2. AVC Experimental Study

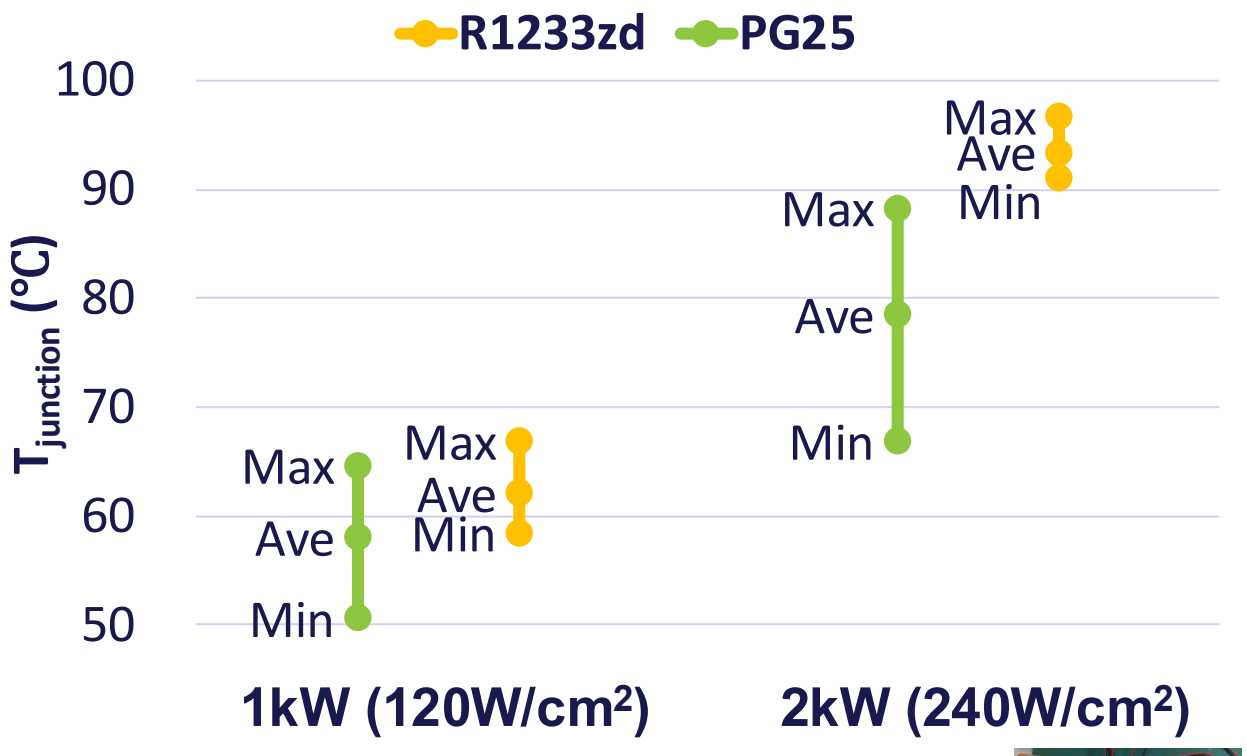
Further Cold Plate Optimization May Improve Two-Phase Performance

- PG25 shows better thermal performance than R515B and R1233zd(E) on both platforms, with the same PG25-designed cold plate and primary-side conditions for each comparison



Ref: AVC Experimental Results
- 3kW TTV (60 × 60mm²)
- 25°C PG25 inlet temperature





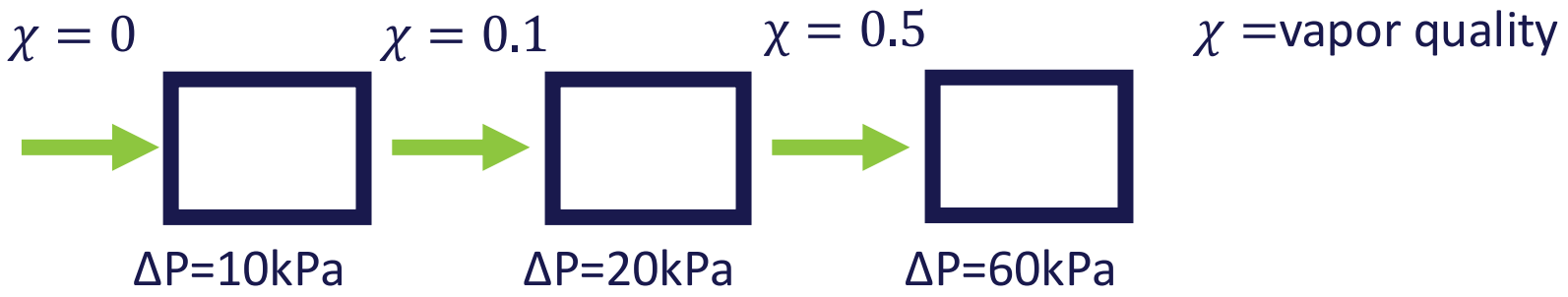
Ref: Wiwynn Experimental Results
- Silicon TTV (26 × 32mm²), 9 $T_{junction}$ measurement points recorded
- 30°C PG25 inlet temperature, 1.5 LPM/kW



Technical Consideration 3: Flow Architecture (Flow Boiling)

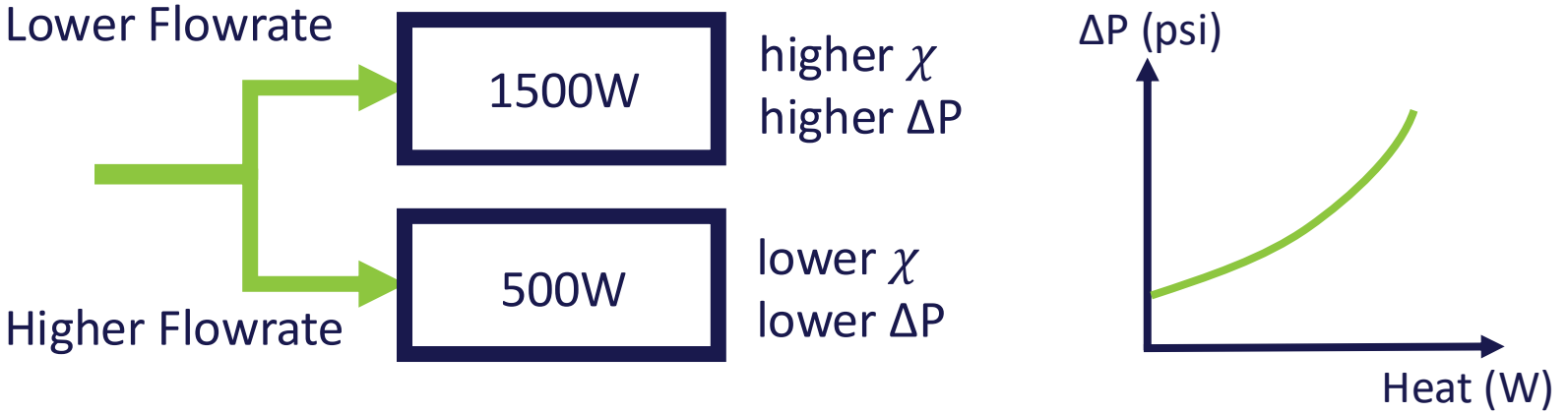
In Series

High pressure drop &
Uneven cold plate performance



In Parallel

Flow maldistribution under
non-uniform heat loads



Technical Consideration 4: Two-phase Cooling Loop

1. **Flow instability and non-uniform distribution**

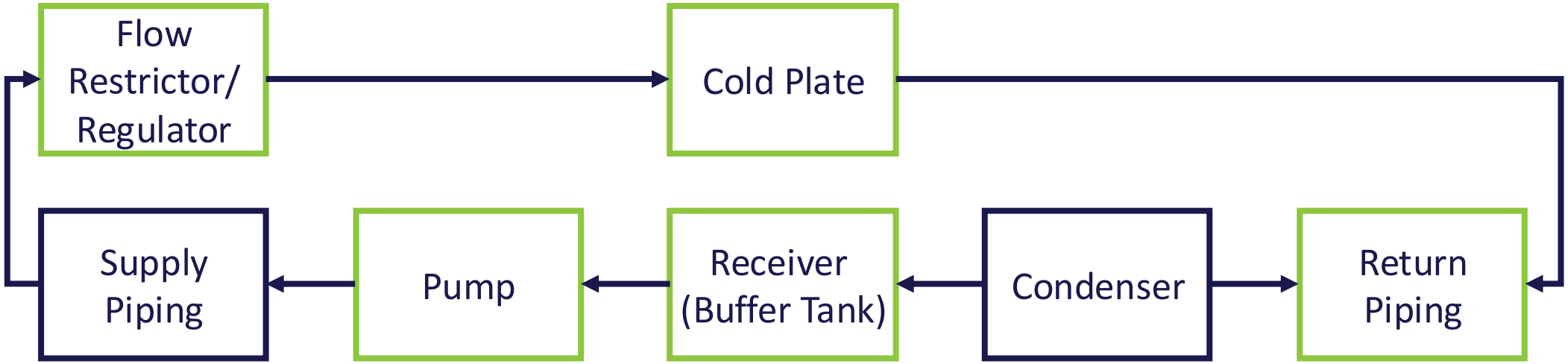
Flow restrictors/regulators may be required → Higher pump head requirement

2. **High vapor volume**

Larger return piping to reduce flow velocity & pressure drop → Larger manifold size/CDU footprint

3. **Pump cavitation**

A low-NPSHr pump and large receiver may be required → Larger CDU footprint



***NPSHr:** Net Positive Suction Head Required

Call to Action

Problem to solve

- Two-Phase optimized cold plate development
- System-level experiments to identify and mitigate flow instabilities in real applications
- Standardization and ecosystem interoperability

Get involved in Cooling Environments – Cold Plate Sub-project

- Every second Wednesday of the month from 8 to 9AM PT
- https://www.opencompute.org/w/index.php?title=Cooling_Environments/Cold_Plate

Thank You!

Wiwynn Whitepaper

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