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FEDERALE DE LAUSANNE

Micro Scale Heat Transfer for Electronics Cooling

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Lausanne, Switzerland

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Overview of Lecture, Topics and Sponsors:

<u>Topics to be Addressed</u>:

- **Overview of two-phase cooling of electronics and computer chips.**
- **Videos of boiling in multi-microchannel cooling elements.**
- **Flow patterns in microchannels.**
- **Design considerations to make two-phase cooling work.**
- **Heat transfer in a silicon cooling element with microchannels.**
- **Comparisons of two-phase vs. single-phase cooling of *targets*.**

***Sponsors of Microscale Two-Phase Flow and Heat Transfer at LTCM*:**
Swiss National Science Foundatin, Swiss CTI agency, IBM, ABB, European Madame Curie, European Space Agency, etc.

Website of My Free Web e-Book on Heat Transfer:

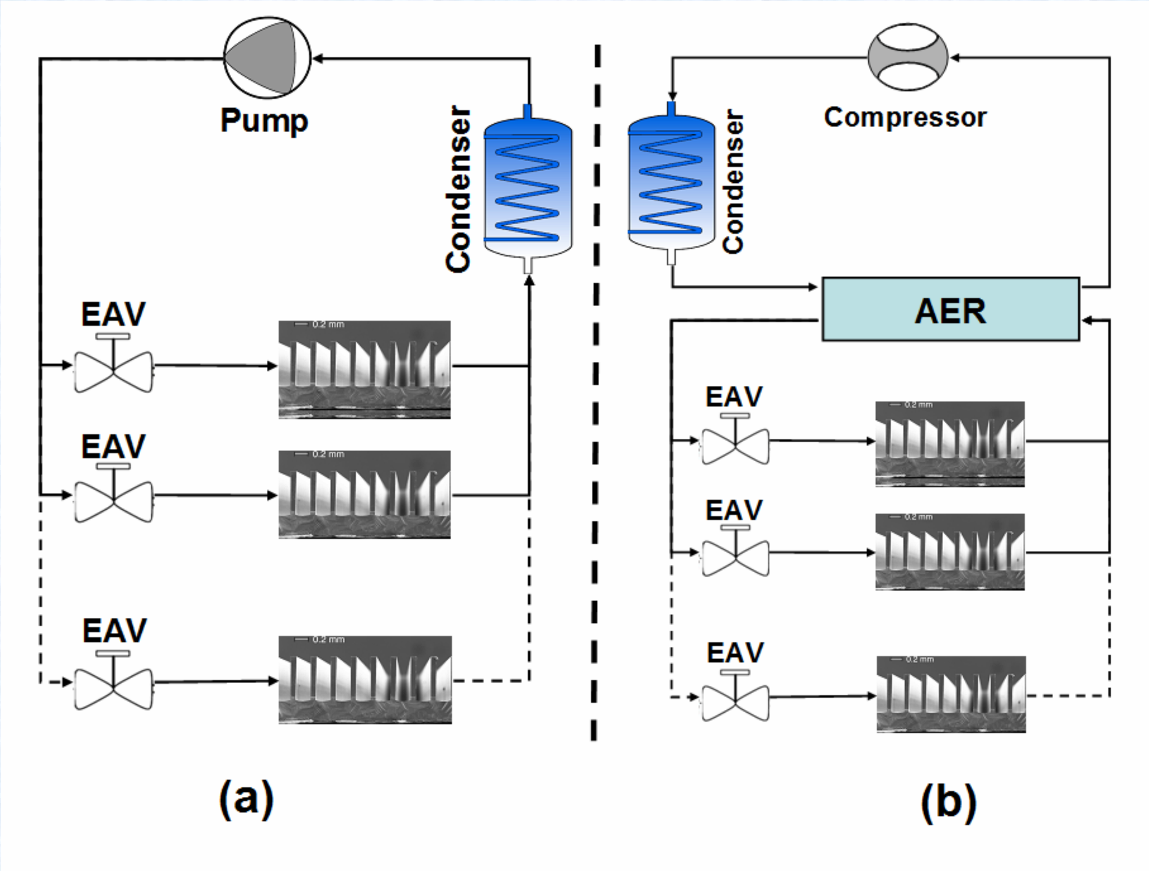
***Engineering Data Book III*:**

a **free e-book at website**

www.wlv.com

Then go to the online book page. It includes 20 chapters on micro and macro scale single-phase and two-phase flow and heat transfer, over 200 videos in Chapter 1, an Excel calculation program, etc.

Liquid Pumped & Heat Pumped Two-Phase Systems



Left: Pump-based system; **Right**: Compressor-based system. **EAV** refers to electronic actuated valve, **AER** refers to accumulator, heat exchanger and receiver components.

Models Required for Design of Micro-Evaporators

- Flow boiling model and flow pattern map: LTCM has 3-zone model and flow map.
- Two-phase pressure drop and void fraction models for microchannels: Must cover laminar, transition and turbulent regimes for circular and rectangular channel shapes. LTCM has 1-d turbulence model for annular microchannel flows.
- CHF model for circular/non-circular, single and multi-channel heat sinks: LTCM has theoretical and empirical methods and is improving them.
- Stable two-phase flow with large scale instabilities: LTCM has method that stabilizes flow in our test sections with up to 134 parallel channels.
- 3-d numerical modeling of conduction in heat sinks: commercial codes are okay.
- Temperature overshoots at startup: LTCM has limited data and a passive method to avoid overshoot, and has patented an active method to avoid them.
- Numerical model to simulate distribution of single and two-phase flows in the inlet and outlet headers: LTCM is working on this.
- Transient simulation capability and hot spots: LTCM has limited data on first topic and has first method to simulate CHF at hot spots with tests now starting.
- Comprehensive simulation tool has been developed within LTCM.

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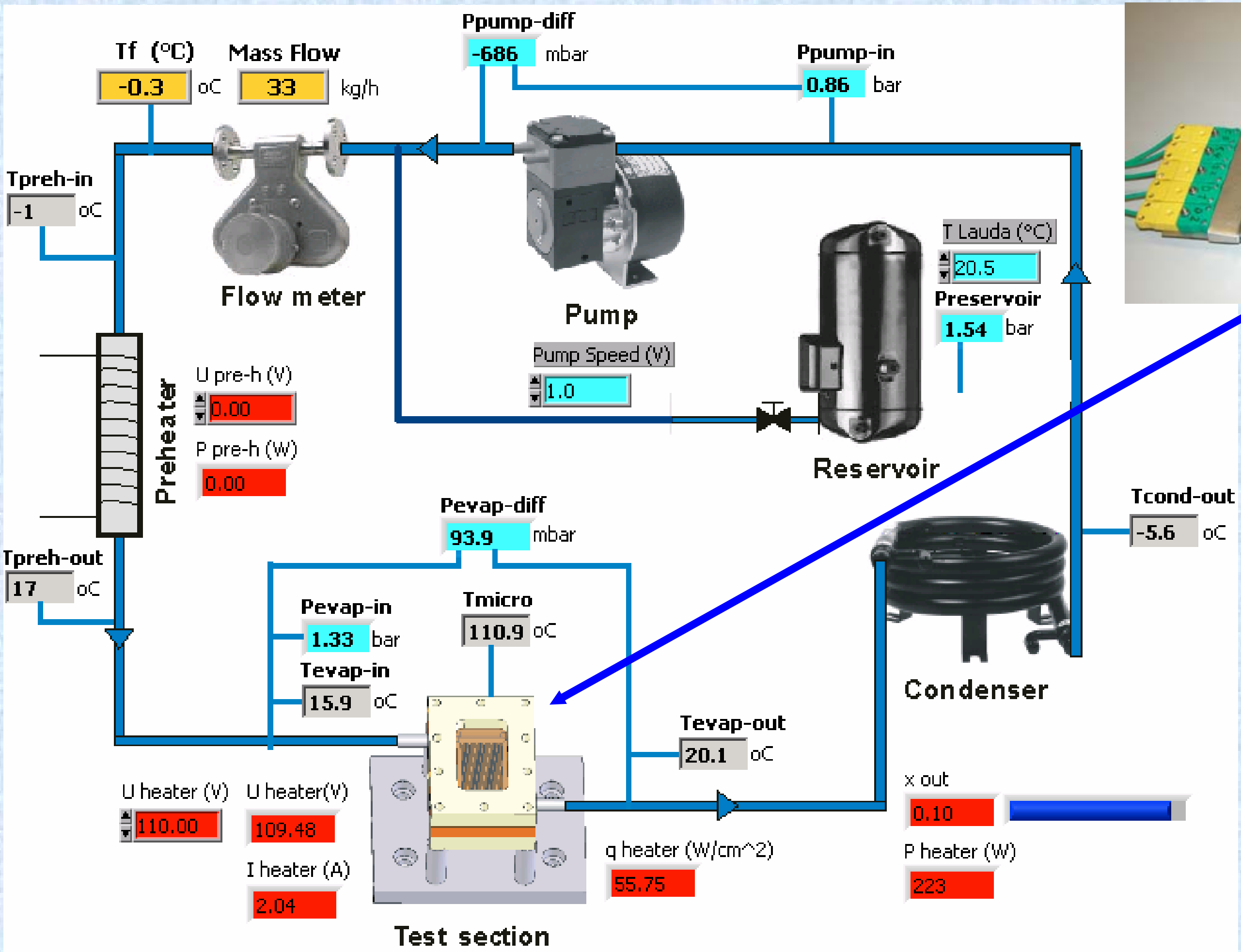
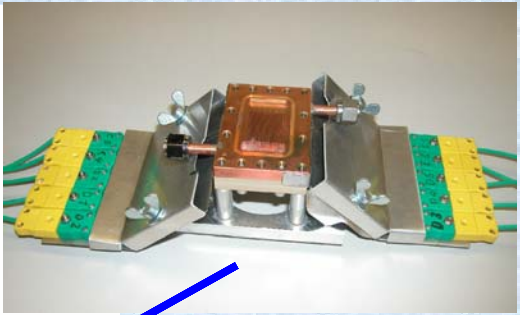
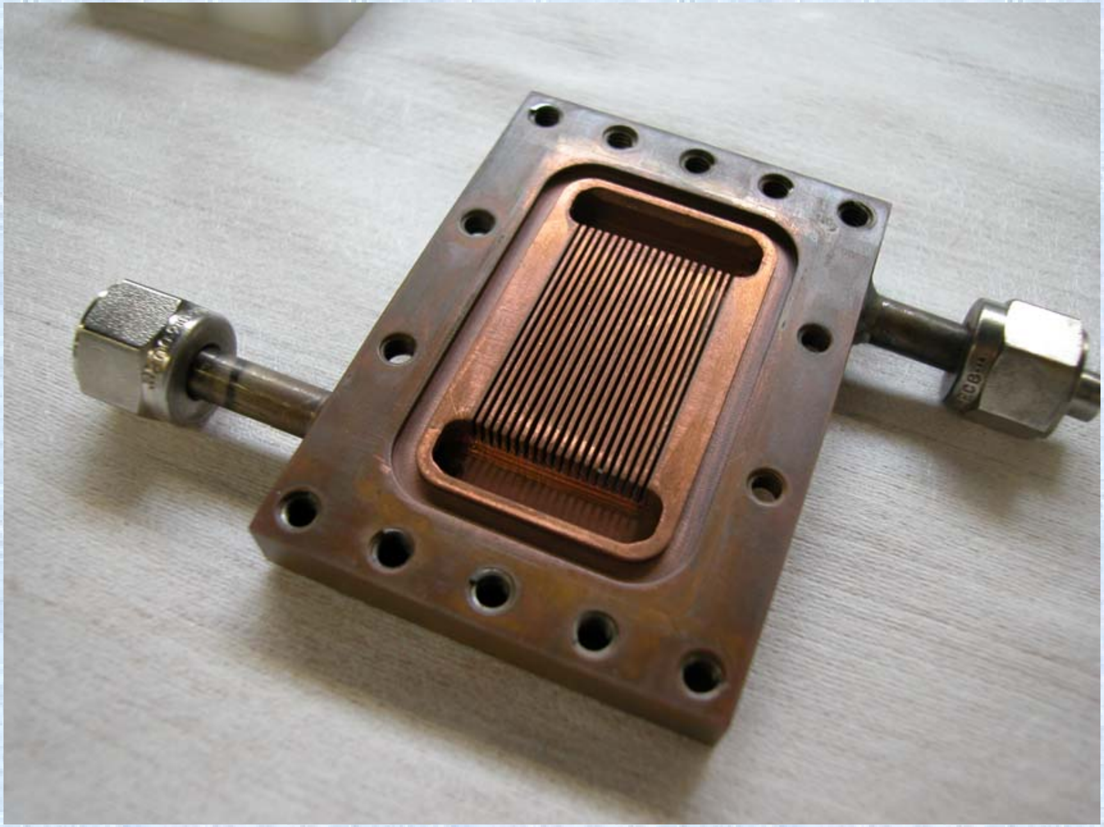
CPU Multi-Microchannel Flow Boiling Test Facility





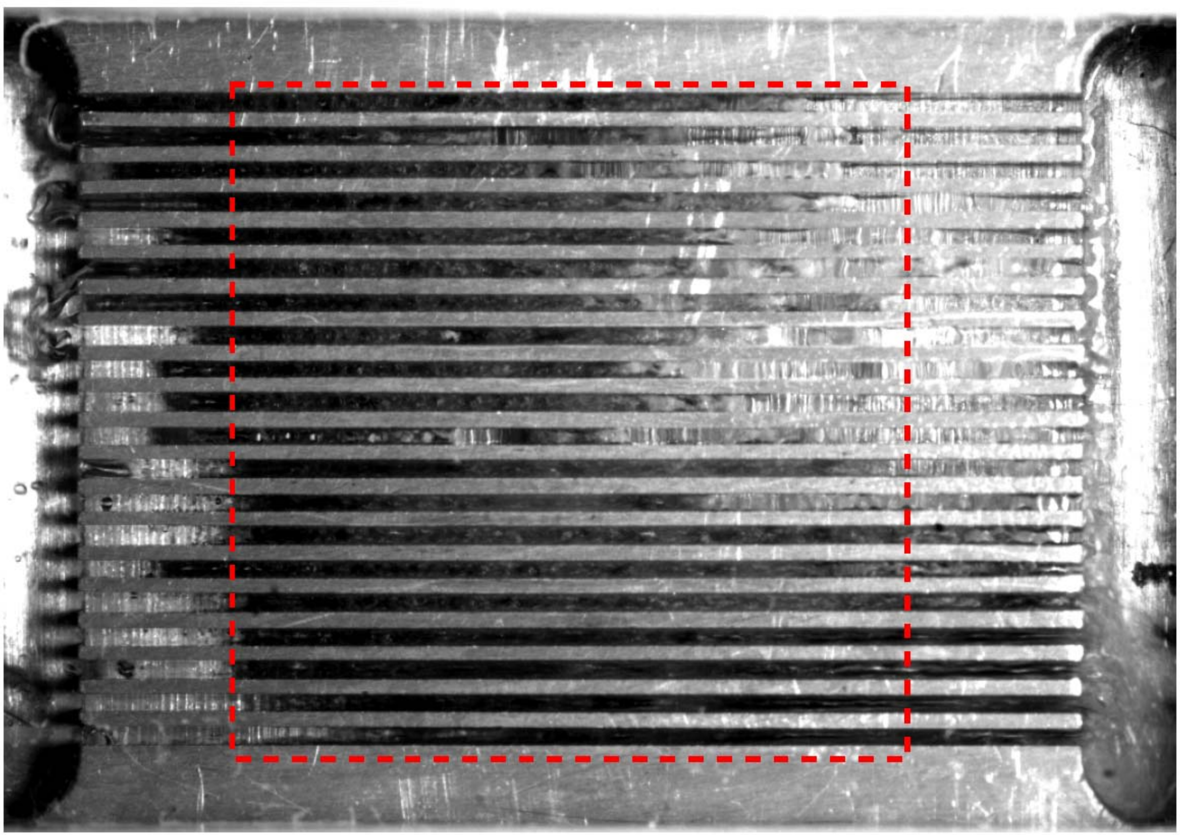
Flow loop for copper micro-channel array: currently used to measure CHF for R-134a, R-245fa and R-236fa with flow visualization using high speed camera. Diagram of Park (LTCM).

CPU Microchannel Flow Boiling Cooling at LTCM



Multi-microchannel evaporator in copper fabricated at LTCM: 20 channels (0.45 x 4.0 mm) and dissipates 340 W/cm^2 with a low pressure refrigerant as coolant (*LTCM PhD thesis of J.E. Park (2008*).

Video at High Heat Flux in Copper: Poor Flow Distribution



Maximum heat flux dissipation possible is only about 115 W/cm^2 (as a result of mal-distribution and back flow, there is over heating in liquid starved/ dry area!). Video of LTCM.

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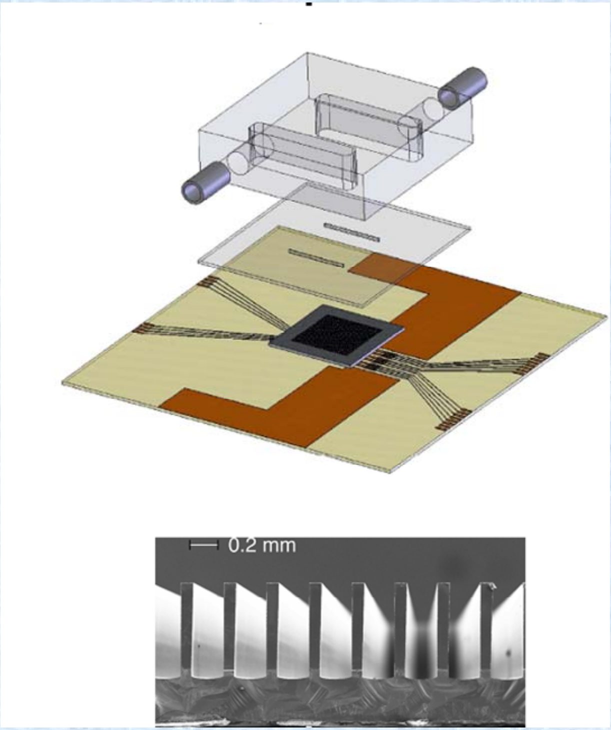
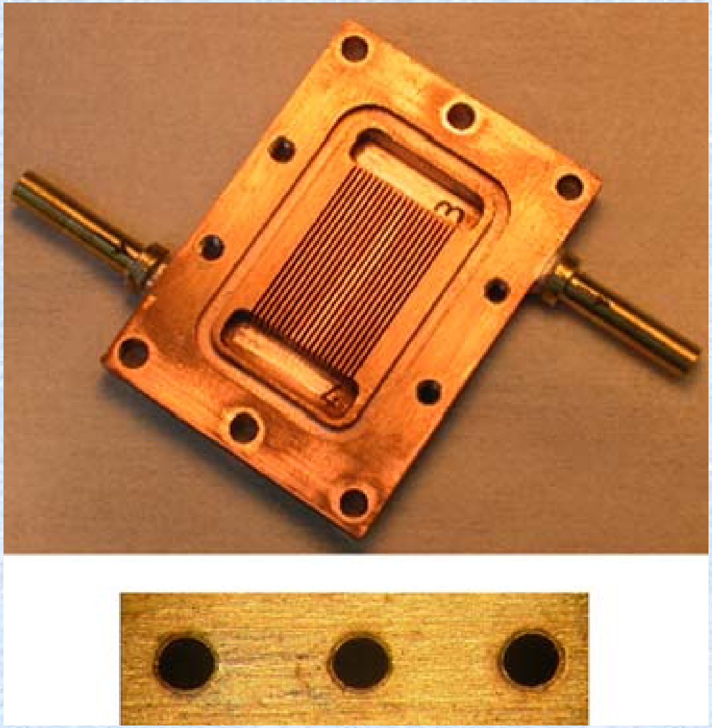
Micro-Evaporators: Basic Geometry and Flow Stabilization



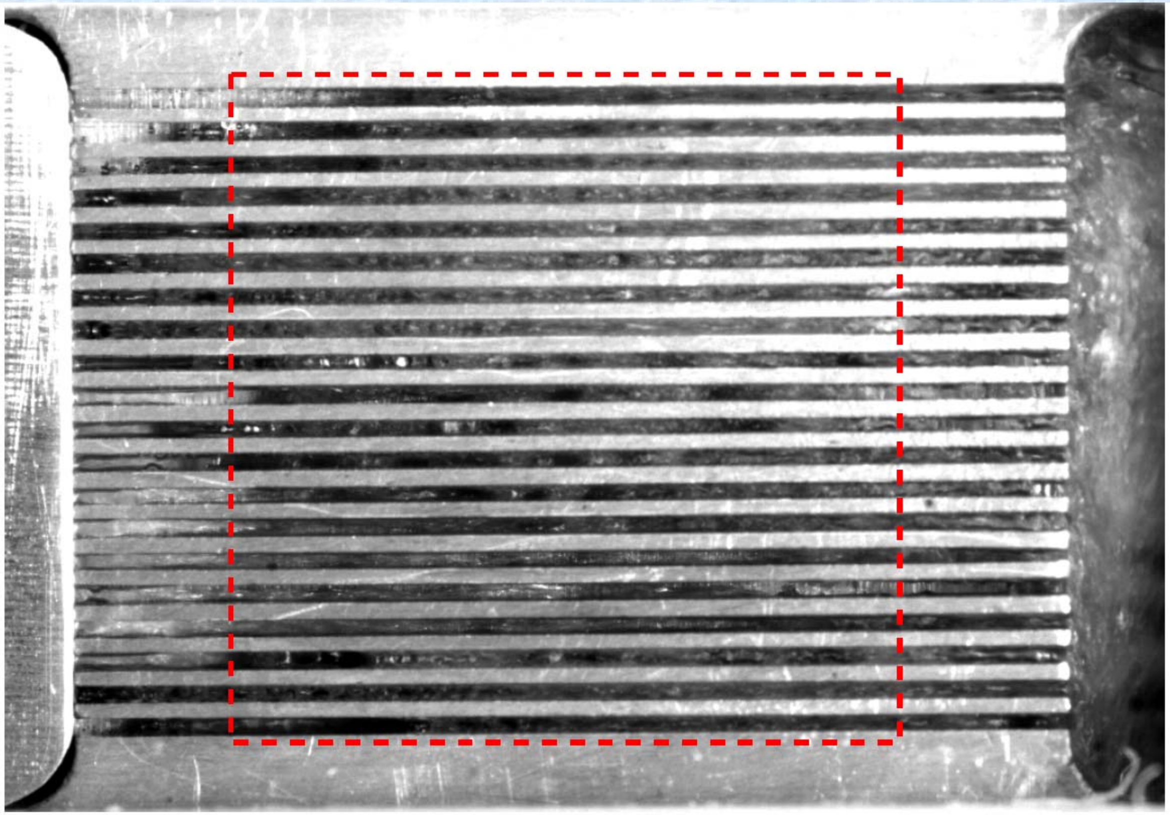
(a)



(b)

Examples of (a) silicon micro-evaporator with micro-channels etched on the chip silicon die and (b) copper micro-evaporator test section made by electro-erosion with an inlet orifice insert.

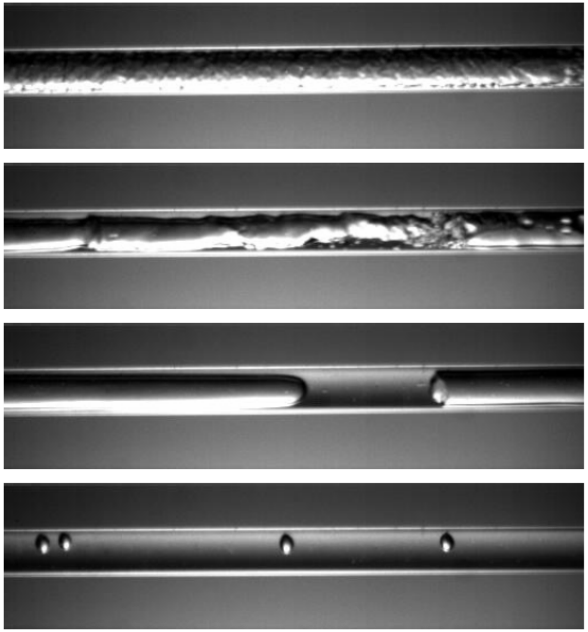
Video at High Heat Flux in Copper: Good Flow Distribution

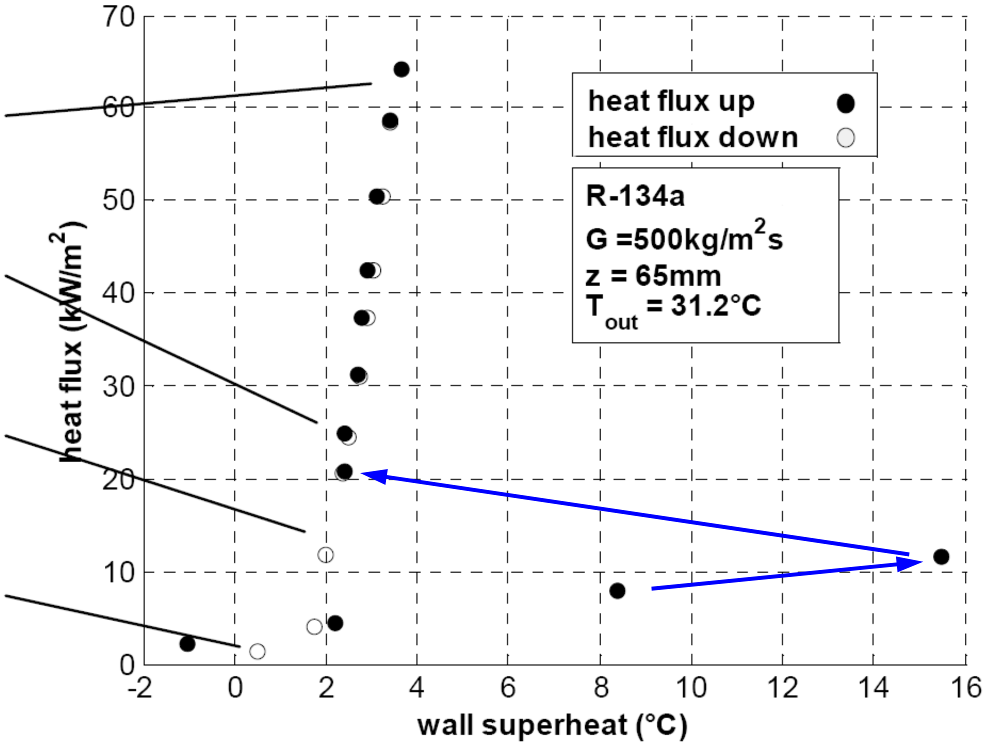


Maximum heat flux dissipation possible is at least 340 W/cm^2 using inlet flow restrictions to prevent back flow and create uniform flow distribution (shown here at 250 W/cm^2 at 2000 images/sec). Video of LTCM.

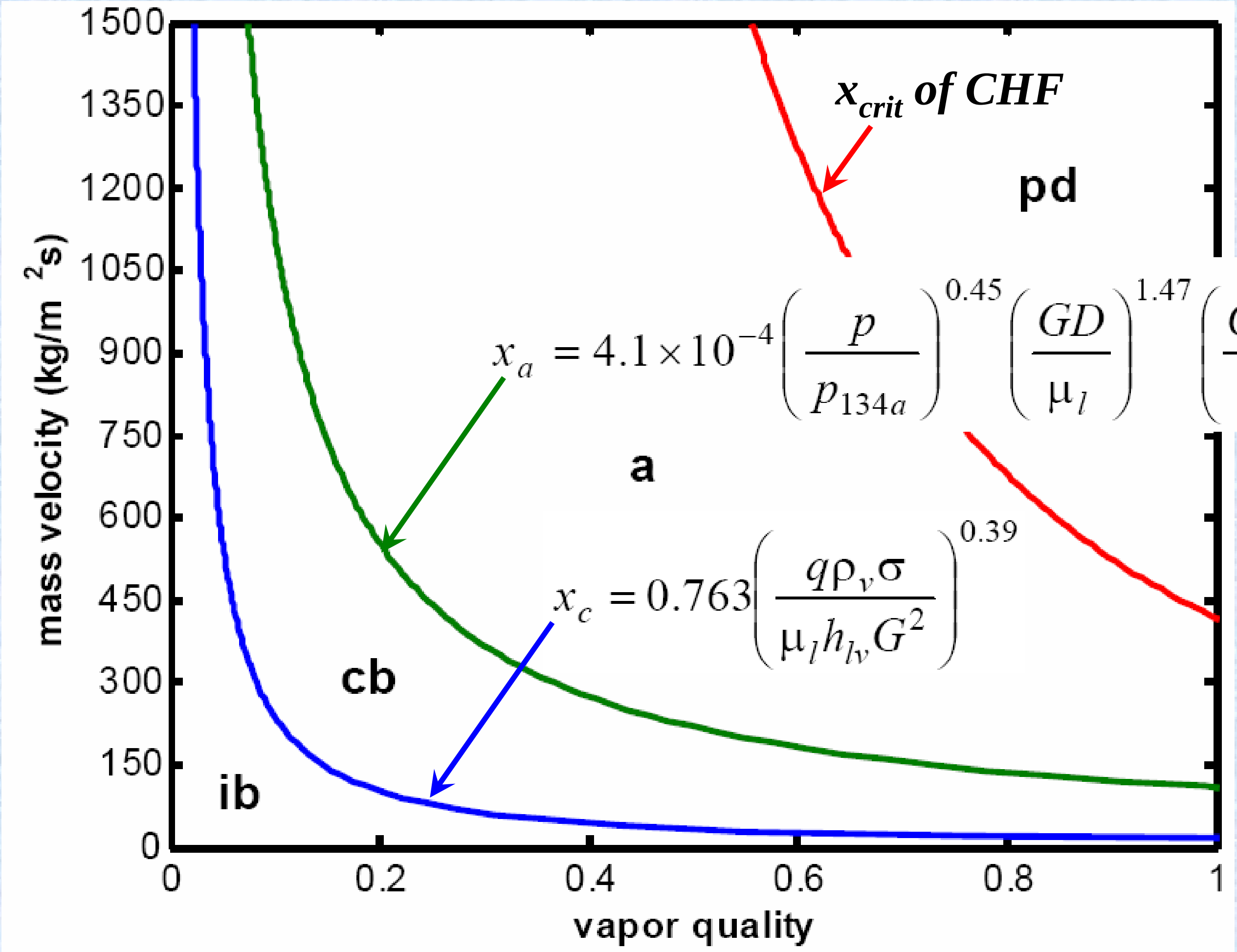
Microchannel Boiling Curve versus Flow Patterns

Flow Patterns in a 0.5 mm Diameter Channel: Videos at exit of microchannel & boiling data from Consolini Ph.D. thesis at EPFL (2008).





Microscale Map of Ong-Thome-Revellin 2009



$$x_a = 4.1\times10^{-4}\left(\frac{p}{p_{134a}}\right)^{0.45}\left(\frac{GD}{\mu_l}\right)^{1.47}\left(\frac{G^2 D}{\rho_l \sigma}\right)^{-1.23}$$

$$x_c = 0.763\left(\frac{q\rho_v \sigma}{\mu_l h_{lv} G^2}\right)^{0.39}$$

Microscale Map of Ong-Thome-Revellin 2009

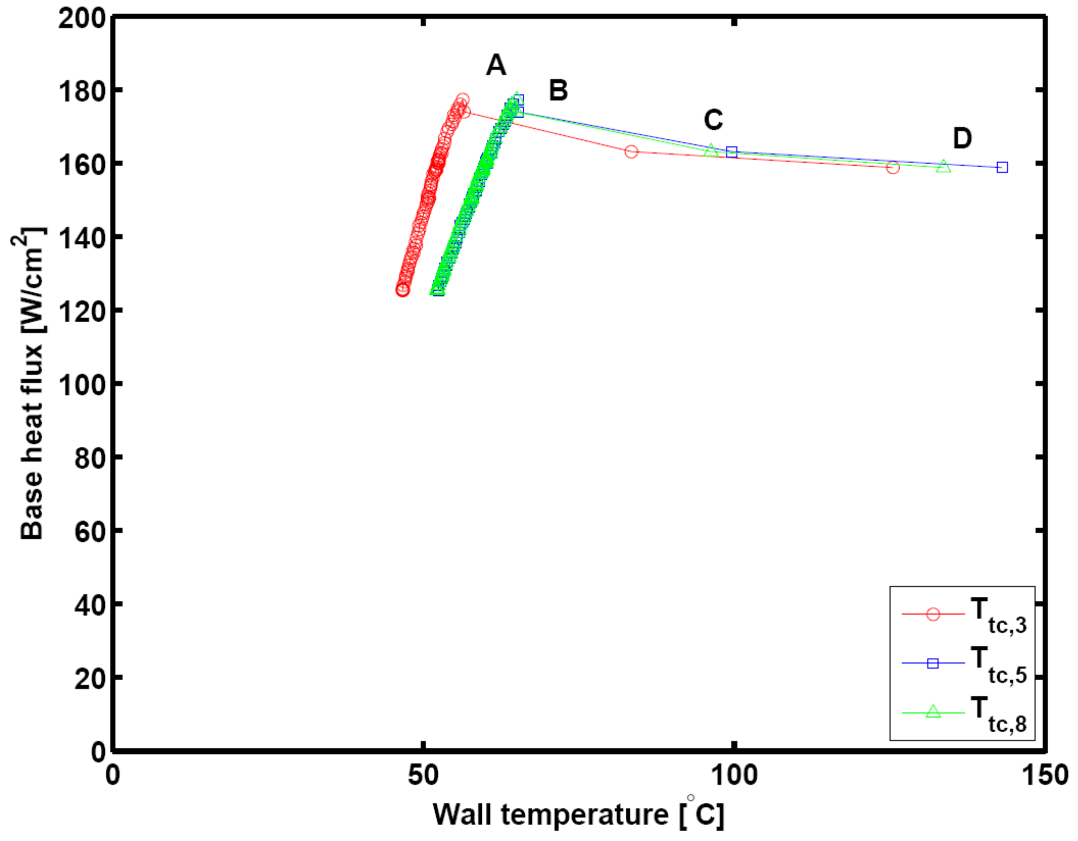
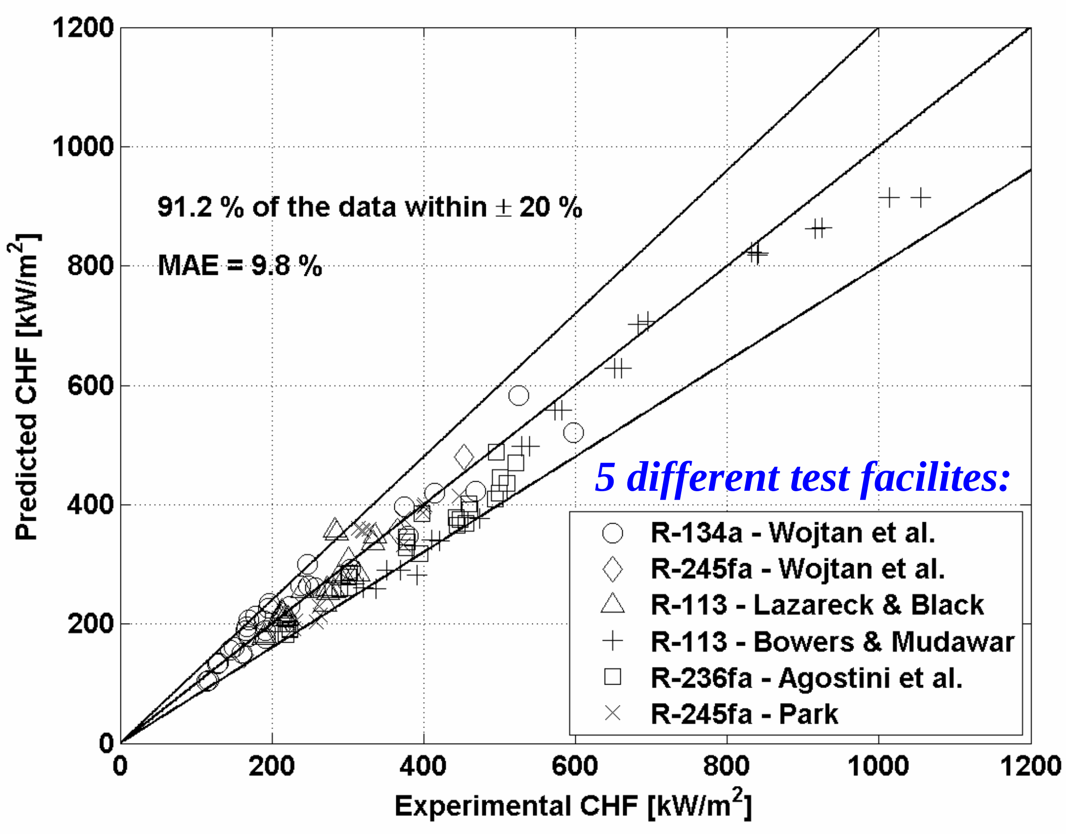
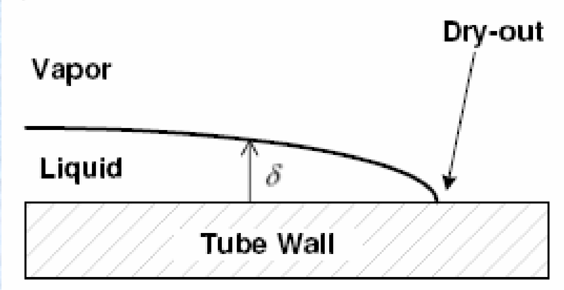


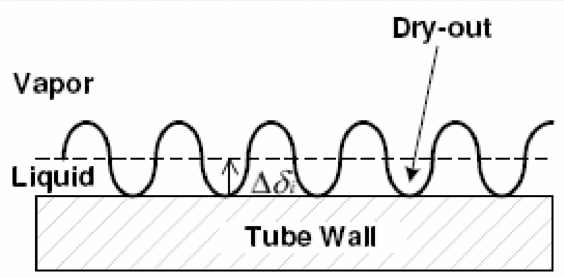
Figure 11: Flow boiling curve measured at three different positions along the channel showing the onset of critical heat flux from Park [13] for R134a.

Revellin-Thome CHF Model for Microchannels: IJHMT'2007



Solving 1-d conservation equations for an annular flow and expression for wave height, solution gives location of CHF:



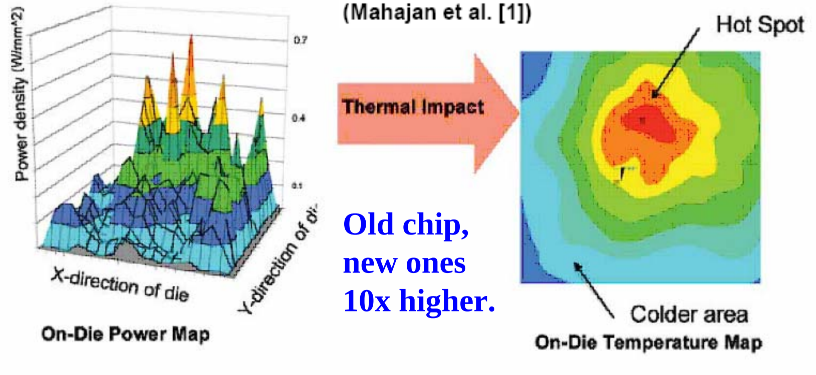


$$\frac{\Delta\delta_i}{R} = 0.088\left(\frac{u_V}{u_L}\right)^{-3/5}\left(\frac{(\rho_L-\rho_V)\,g\delta^2}{\sigma}\right)^{-1/5}$$

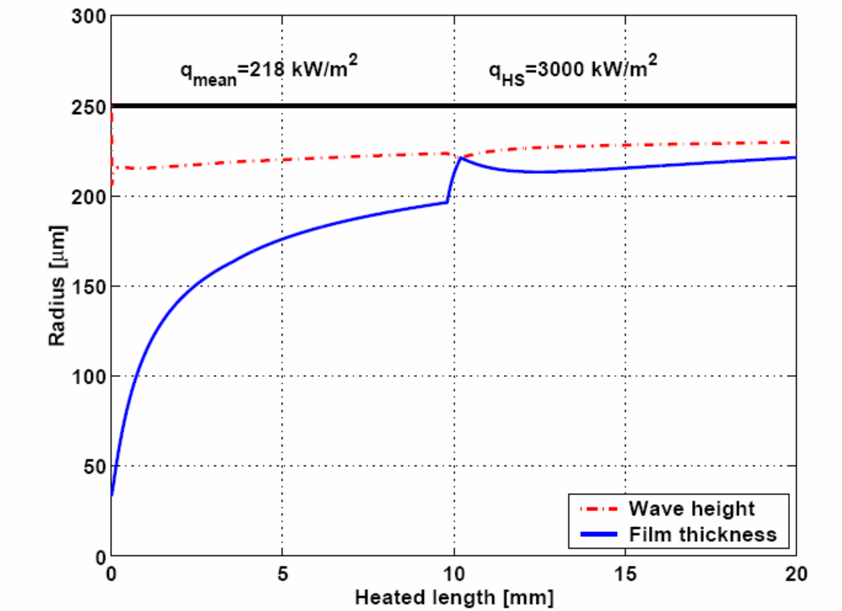
$$\Delta\delta_i = \delta$$

Analysis of CHF of Hotspots on Computer Chips

Heat flux/ temperature distribution on chip.



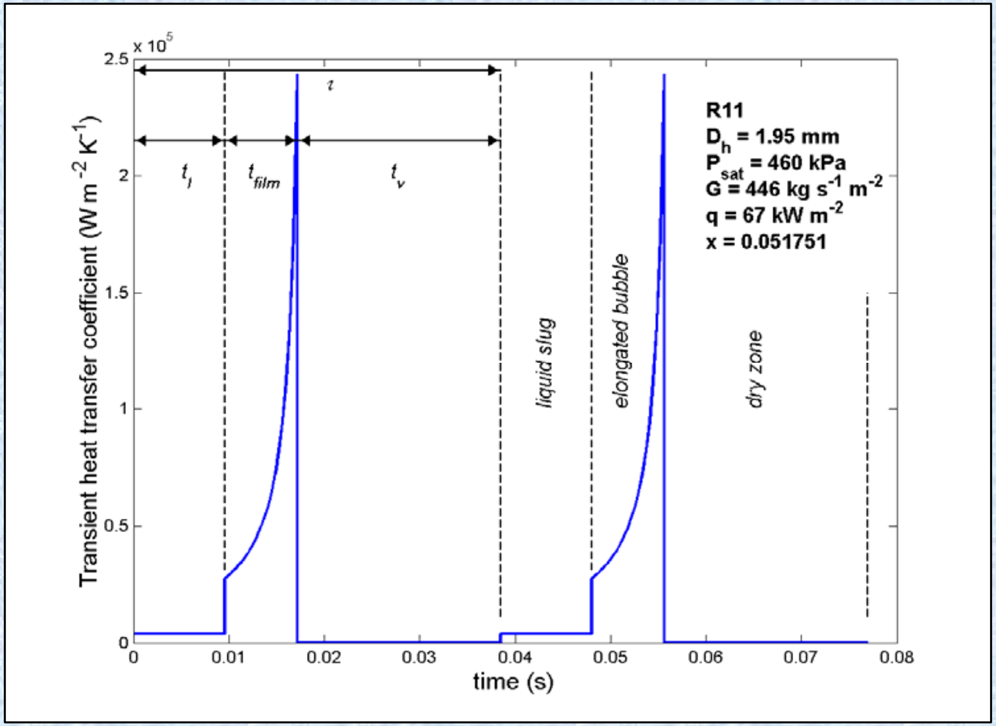
Revellin-Thome (2007): 1-d numerical model applied to prediction of local CHF at hotspots in micro-channels with micro-scale boiling.



(e) $q_{HS} = q_{HS,\,max} = 3000$ kW/m^2. Dryout occurs at the hot spot location.

Hot spot heat flux as a function of the hot spot size located at the midpoint along the circular microchannel for D=0.5 mm, G=500 kg/m^{2}s, T_{sat}=30°C, L_{MEV}=20 mm and the local hot spot situated at the midpoint along the microchannel.

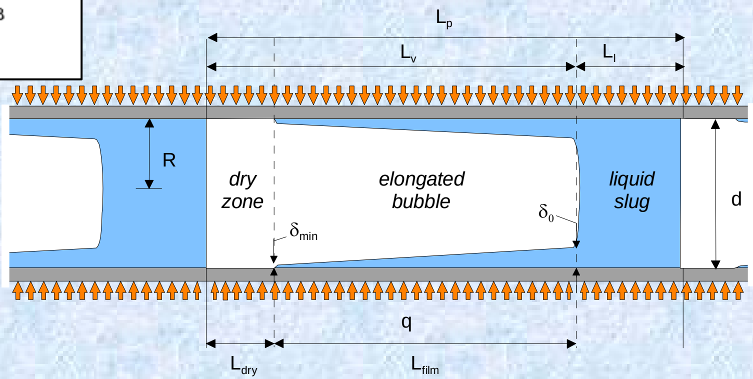
Local time averaged heat transfer flow boiling model



Three-zone flow boiling model of Thome, Jacobi and Dupont: time averaging of dynamic heat transfer coefficients gives local heat transfer coefficient (7 fluids from 7 labs in original data base, now 5 more fluids).

(in Int. J. Heat Mass Transfer, 2004)

$$h(z)=\frac{t_l}{\tau}h_l(z)+\frac{t_{film}}{\tau}h_{film}(z)+\frac{t_{dry}}{\tau}h_v(z)$$



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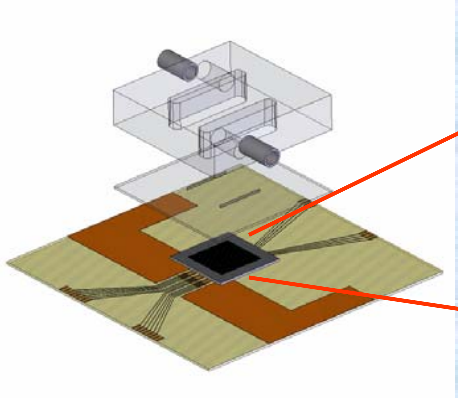
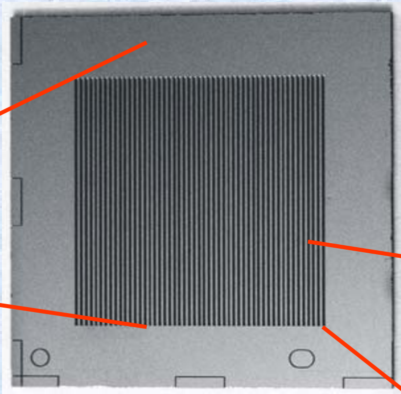
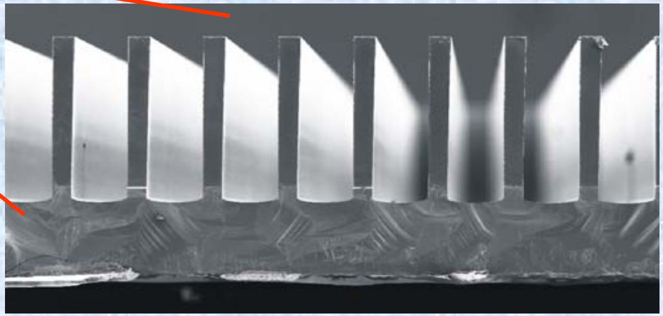
LTCM/IBM Micro-Evaporator Test Section in Silicon

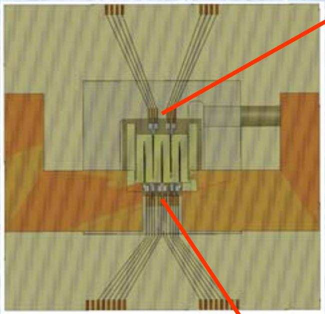


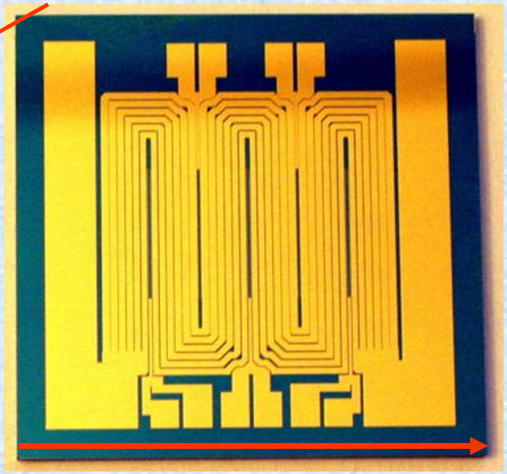


Si microchannel evaporator:

- **Channel widths: 200, 100, and 50 microns**
- **Manifolds: One inlet/one outlet or one inlet/two outlets**
- **Slit inlet avoids instabilities and generates bubbles to avoid temperature overshoot**
- **Can be bonded to top of microprocessor**







(red arrow) Channel Orientation on backside of chip

Cobra Heater on backside of silicon chip

- **Simulates heat dissipated by microprocessor or electronics**
- **Better measurement accuracy**
- **Lower thermal resistance to evaporation surface**
- **5 RTD sensors along channel length, span across 1/3rd of width**

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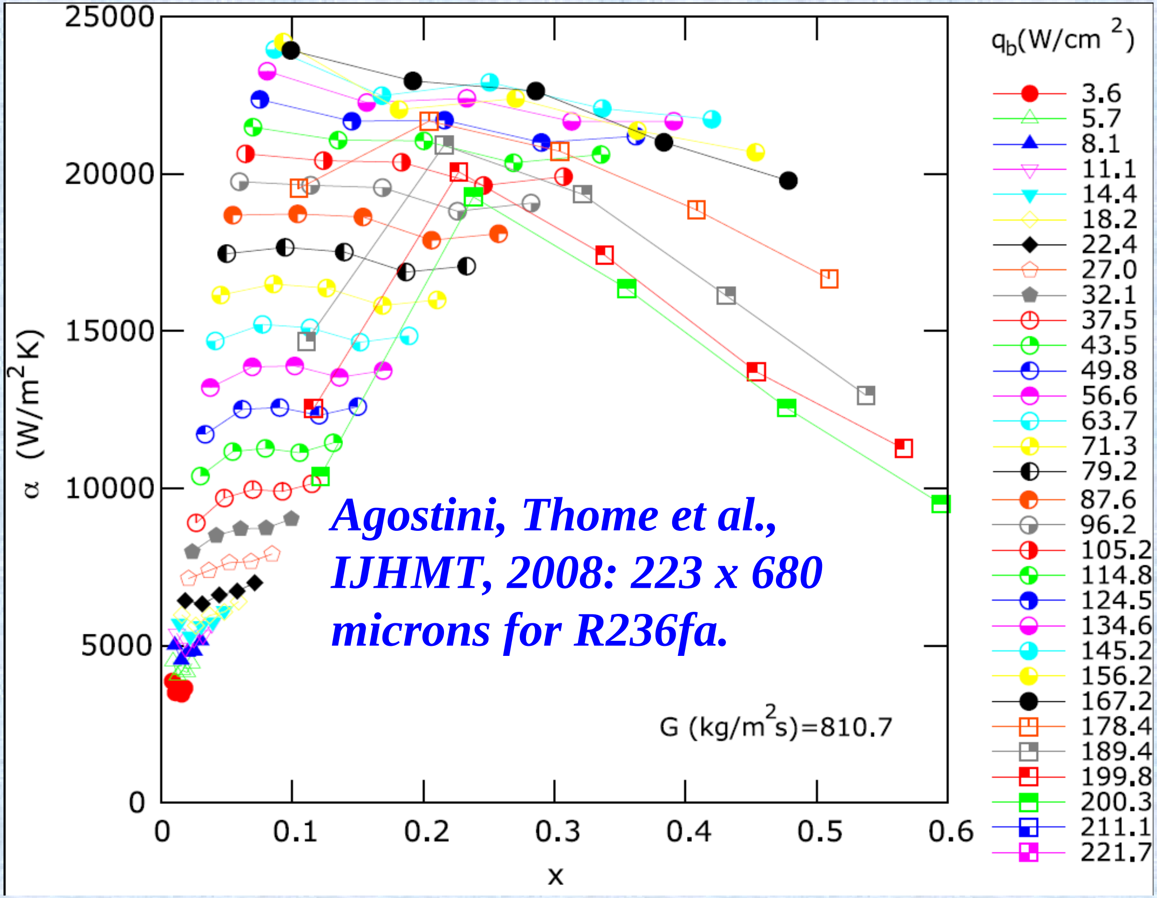
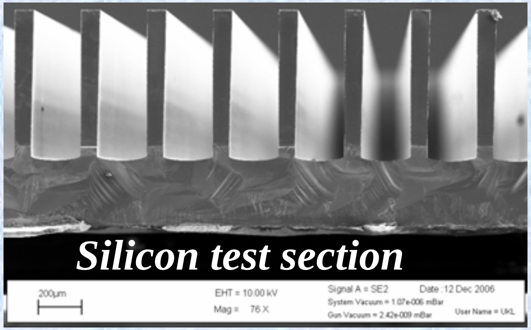
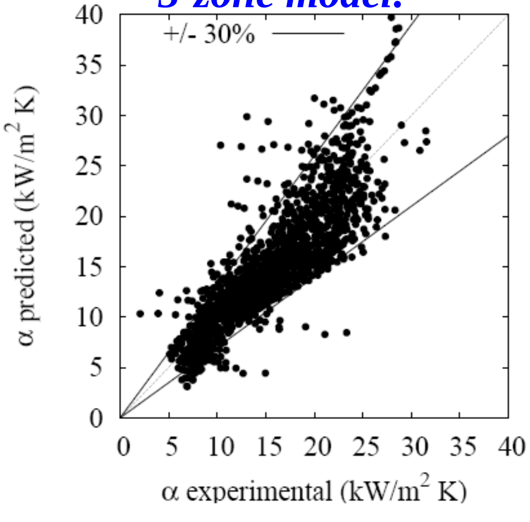
LTCM/IBM Silicon Test Section: Results/3-Zone Model



Agostini, Thome et al., IJHMT, 2008: 223 x 680 microns for R236fa.

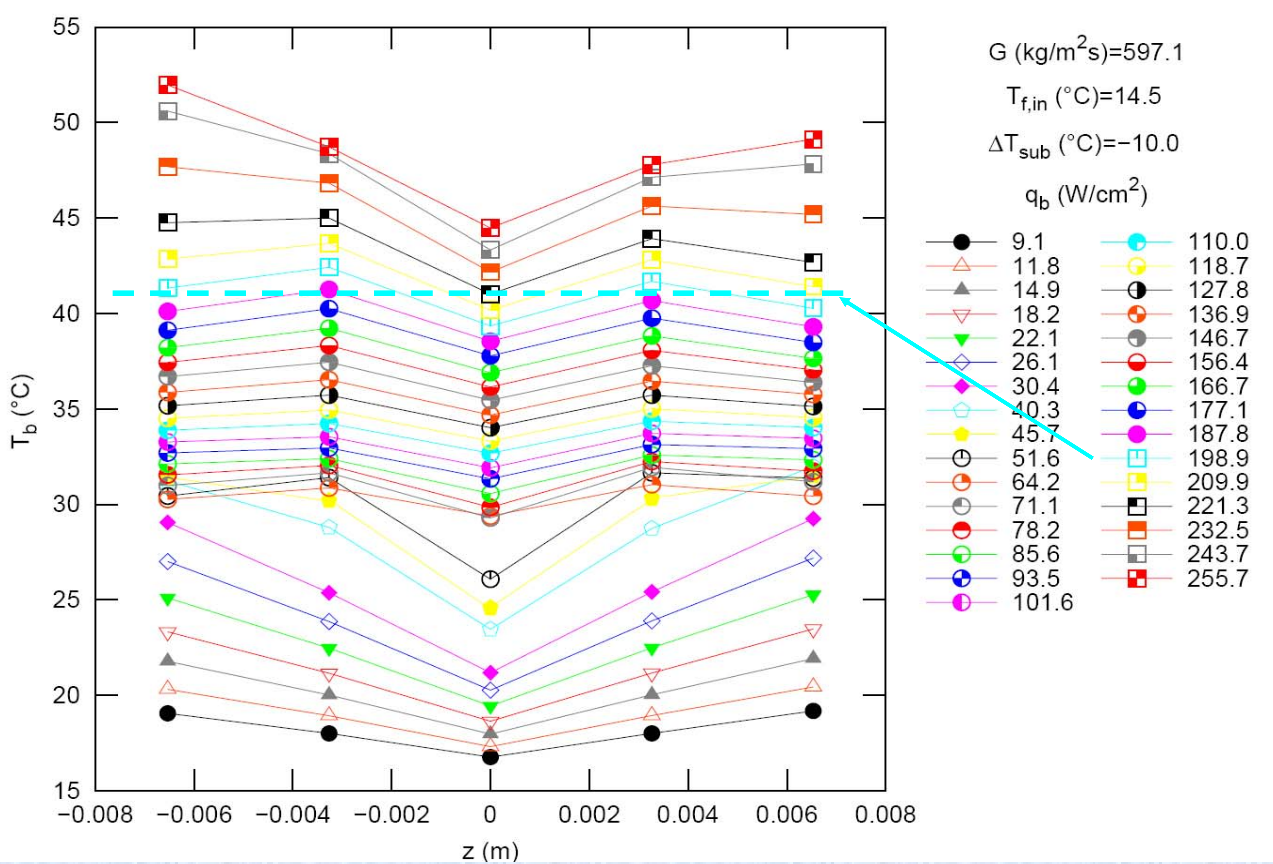


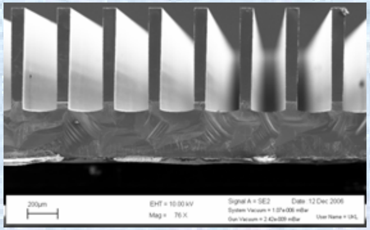
3-zone model:



Above graph shows some local heat transfer data based on effective area plotted with heat flux based on footprint area indicated (up to 2.11 MW/m^2)…footprint based boiling heat transfer coefficients go over 100’000 W/m^{2}k using a *refrigerant*, not *water*.

LTCM/IBM Silicon Test Section: Temperatures





***IEEE Trans. on Components and Packaging Technologies*, 2008: featured on cover of the journal.**

Above graph shows some local base temperatures at various heat fluxes with one inlet and two outlets (up to 256 W/cm^2). CHF not reached with this test section with 134 channels. Notice the nearly uniform, low temperatures that can be achieved.

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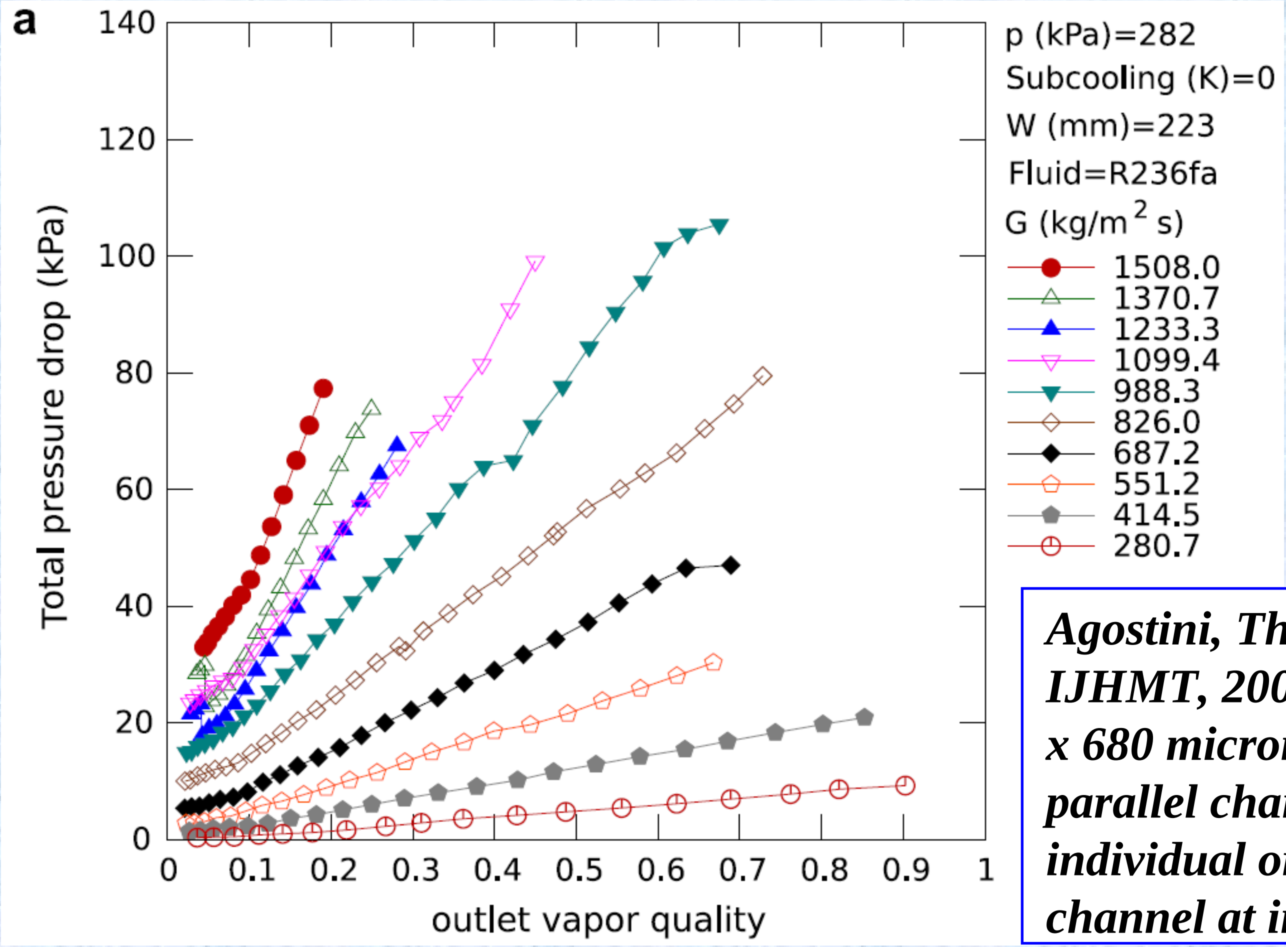
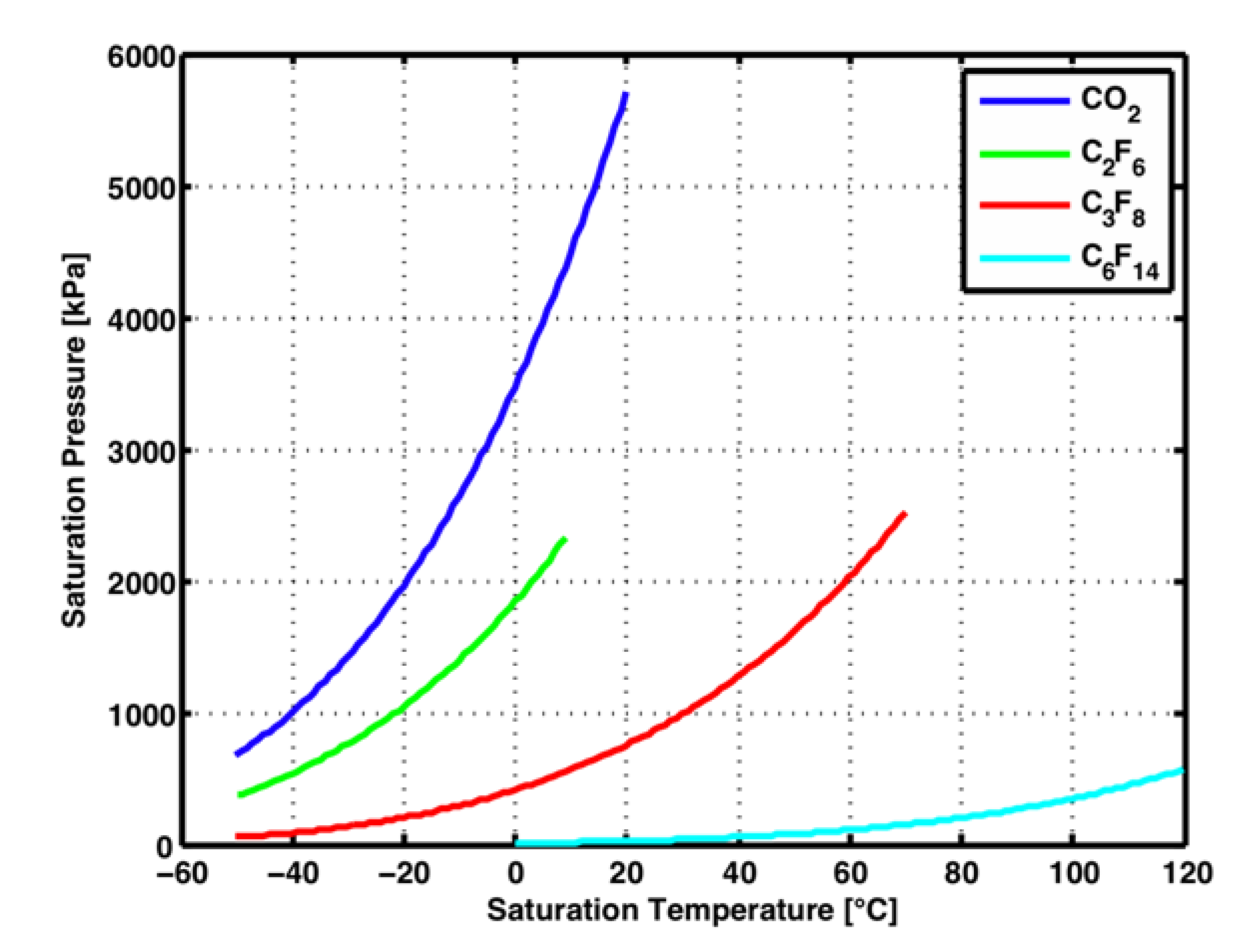
Two-Phase Pressure Drops in Parallel Channels



Agostini, Thome et al., IJHMT, 2008, Part 3: 223 x 680 micron channels (67 parallel channels) with individual orifices for each channel at inlet and outlet.

Saturation Pressure Curves of Radiation Hard Fluids

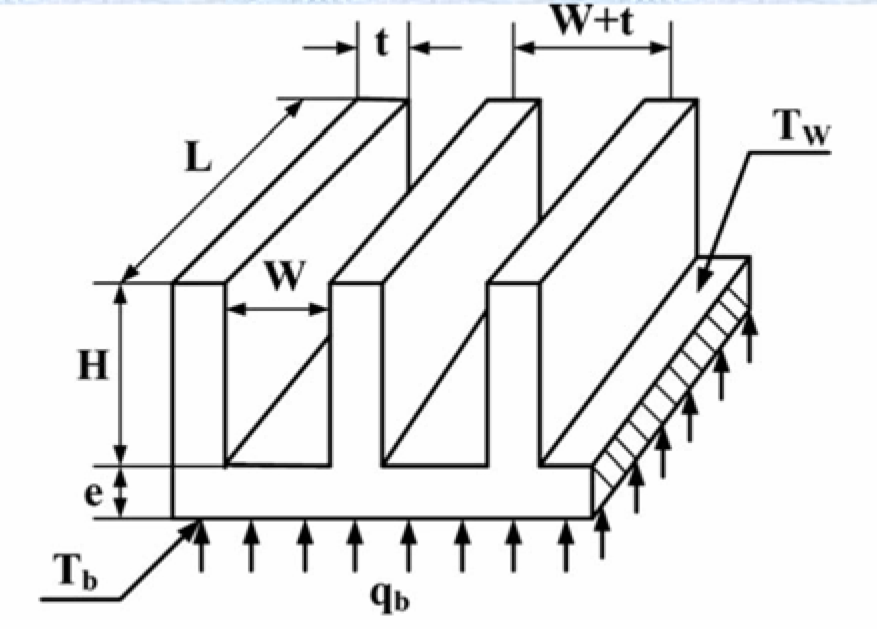


Simulations for Development of CERN GigaTracKer

Assumptions made for the present simulations are: (*1*) the evaporator is uniformly heated from the bottom with a base heat flux of q_b, (*2*) the flow through the cooler is uniformly distributed between all the channels, (*3*) the top of the cooler is adiabatic and (*4*) for two-phase flow, no inlet subcooling is used. The models used for single-phase heat transfer and pressure drop are those from Shah and London, while the three-zone model and homogeneous model were used for the two-phase heat transfer and pressure drop, respectively.

The fluid to be simulated will be radiation-hard fluorocarbons and CO_2. The choice of fluids depends on the desired operating condition:
*** For temperatures below -10˚C, the best choice of fluid would be between CO_2 and C_2F_6 but possibly also C_3F_8.**
*** The most common cooling fluid used at CERN is C_6F_{14} and is used in single-phase flows only. This fluid is not ideal for two-phase cooling as it is a low-pressure fluid, having a saturation temperature of 56˚C at atmospheric pressure, implying that the system would need to be under vacuum for lower temperatures. This has the disadvantage that one is limited by the allowable pressure drop within the cooling device, implying that channels should be relatively large. The potential of air also leaking into the system becomes greater.**
*** Thus, for two-phase cooling, CO_2, C_2F_6 and C_3F_8 will be compared.**

Simulation of Micro-Evaporator Performance: Geometry and Dimensions



Parametric study

In all cases, the base thickness, *e*, will be zero, thus showing a best-case scenario as any additional material added can be accounted for in separate calculations.

It is also stated that the GTK should not see a temperature difference of more than 5˚C while being kept as cold as possible (~ -30˚C).

Effect of Dimensions on Single-Phase Cooler

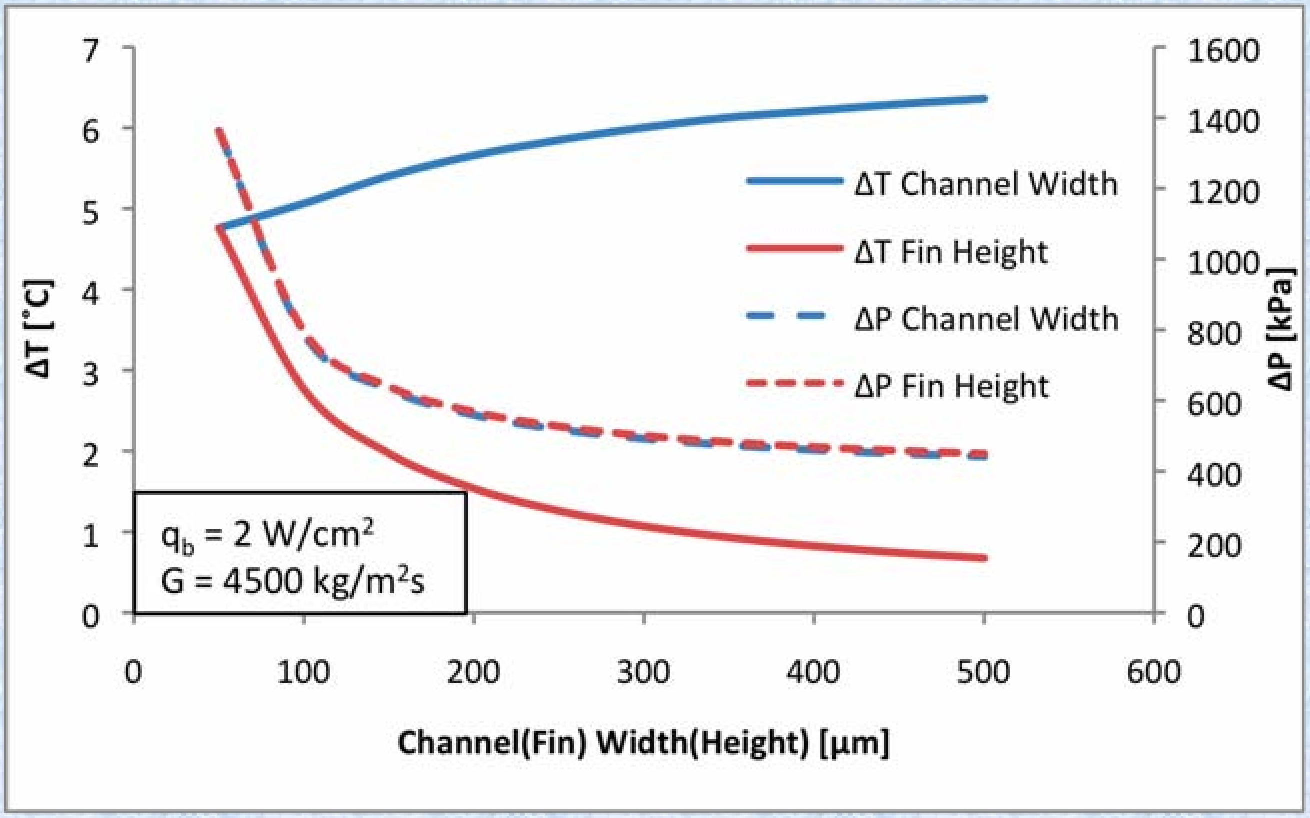


Figure 7: Effect of *channel width* and *fin height* on *maximum base temperature difference relative to the inlet* and on *pressure drop* for single-phase flow using C_6F_{14} relative to 50 microns width or height for fixed fin thickness of 25 microns.

Effect of Fin Height on Two-Phase Cooler

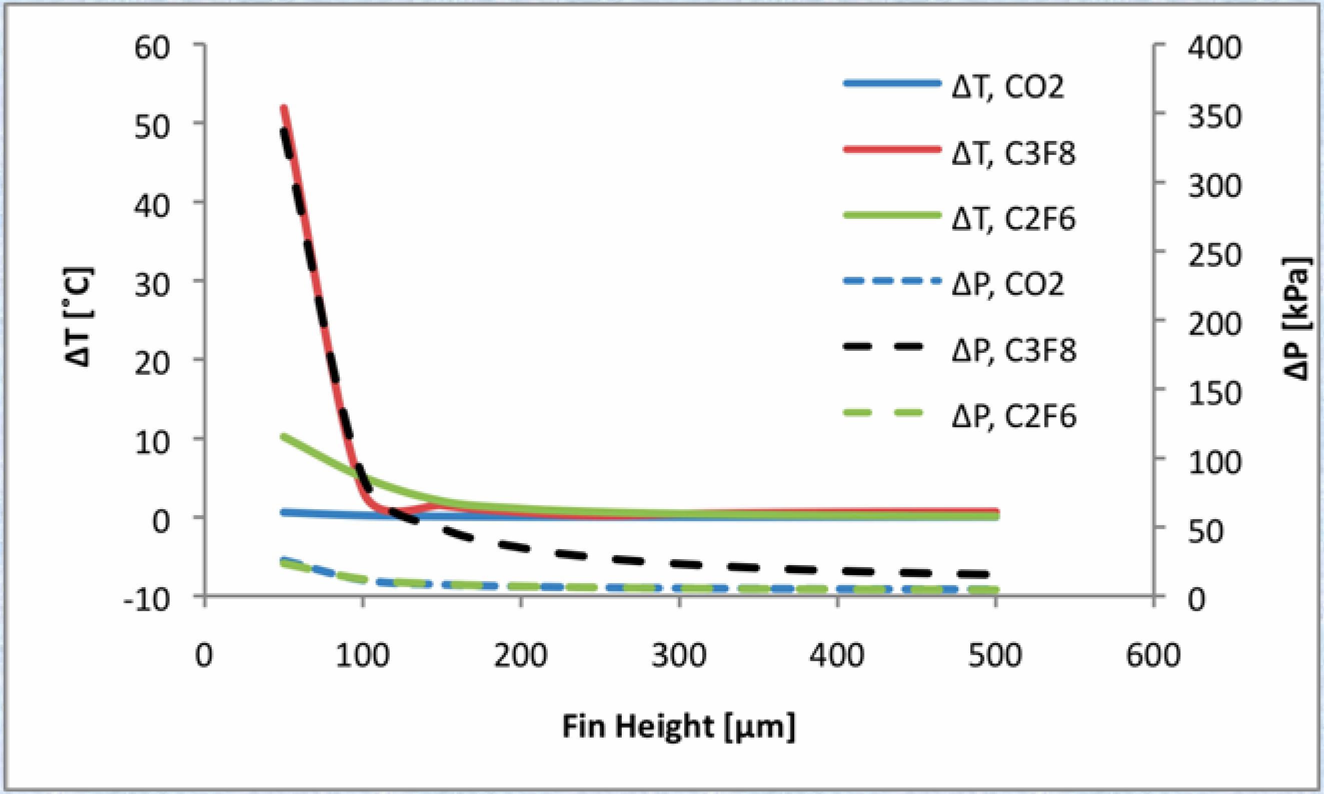
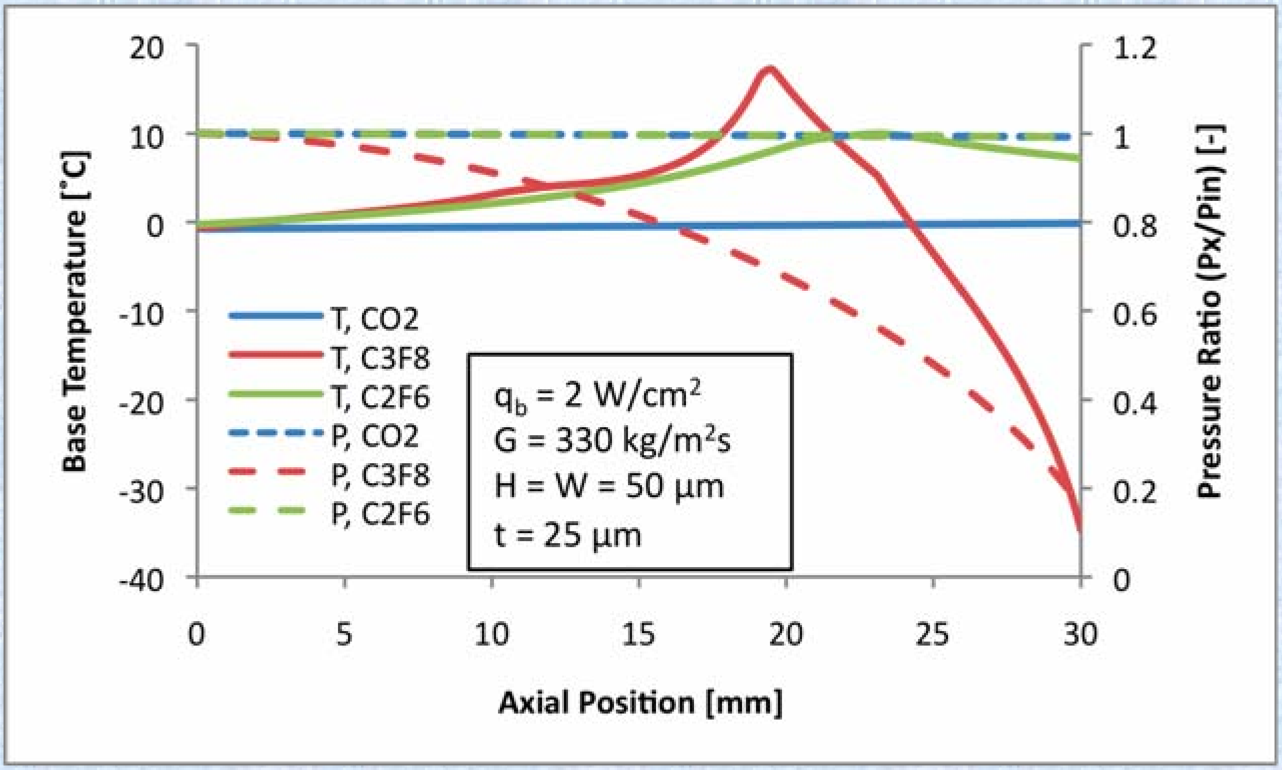


Figure 9: Effect of *fin height* on *maximum base temperature difference relative to inlet* and on *pressure drop* for two-phase flow for channel width of 50 microns and fin thickness of 25 microns. Simulation shows that CO2 is the best candidate.

Temperature/Pressure Profiles in Two-Phase Cooler



The pressures are shown in terms of the ratio of the local pressure to the inlet pressure. The actual base temperature varies depending on the heat transfer/pressure drop/vapor pressure curve of the fluid. Smallest temp. variation is for CO_2. Pressure drops are also significantly less for CO_2 and C_2F_6 since their viscosities are about half that of C_3F_8.

Comparison of Single-Phase to Two-Phase Cooler

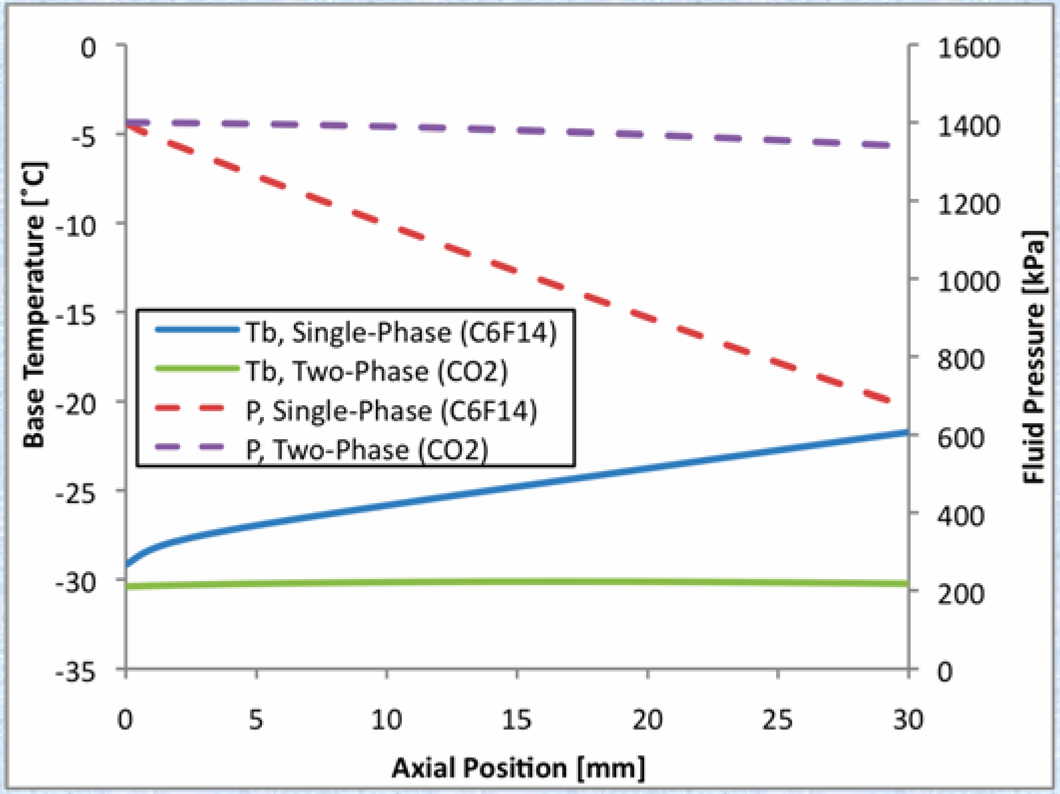


Figure 10 shows a comparison of single-phase to two-phase cooling. C_6F_{14} has an inlet temperature of -30˚C while the two-phase fluid CO_2 has an inlet saturation temperature of -30˚C. The fin height and channel width were 50 µm, while the fin thickness was 25 µm. A base heat flux of 2 W/cm² was applied. The actual junction/base temperature and fluid pressure along the channel are shown.

Comparison of Single-Phase to Two-Phase Cooler

For both the single-phase and two-phase results, the axial temperature difference is below 5K, although the increase in temperature for the two-phase fluid is much less than for the single-phase fluid (0.14K vs. 4.7K).

The difference in the fluids' pressure drops is even more significant:

*** The single-phase fluid requires a mass flux of 4500 kg/m^{2}s to obtain a temperature difference below 5K with a pressure drop of about 700 kPa!**

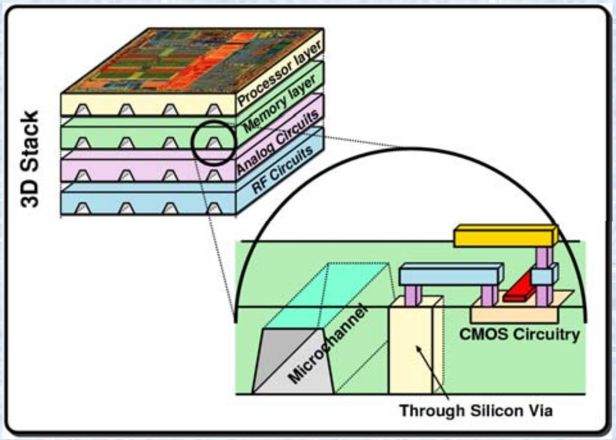
*** The two-phase fluid only required a mass flux of 250 kg/m^{2}s that resulted in a pressure drop of 60 kPa.**

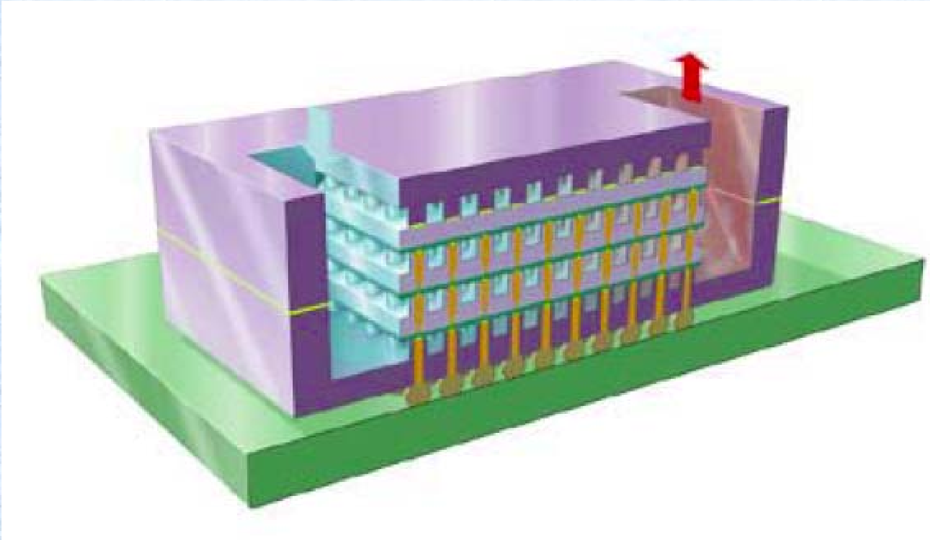
The power required to move the two fluids is 1984 mW and 28 mW, respectively.

Simulation implies that CO2 two-phase cooling relative to C6F14 single-phase cooling yields much lower axial temperature gradients in the GTK, lower pressure drops and pumping power consumption, and very high heat transfer coefficients but is more complex to implement.

CMOSAIC: 3D-IC Thermal Performance with Microscale Liquid/Evaporative Cooling

- **A 3D computer chip with integrated cooling system is expected to:**
  - ***Overcome the limits of air cooling***
  - ***Compress ~10^{12} nanometer sized functional units (1 Tera) into one cubic centimeter: nearing the equivalent in human brain***
  - ***Yield 10 to 100 fold higher connectivity***
  - ***Cut energy consumption***





A new $4million Swiss consortium project lead by Prof. Thome

Summary and Advantages:

Flow boiling in micro-evaporator elements is a convincing solution for cooling high energy physics targets and electronics rather than a viscous liquid because:

- **It yields much larger heat transfer coefficients,**
- **It makes low temperature difference operation possible,**
- **It has high critical heat fluxes for high W/cm^2 operation,**
- **It provides a near uniform temperature of cooled element,**
- **Hot spot cooling is self-compensated by boiling itself,**
- **It has lower pumping power vs. single-phase cooling,**
- **Evaporation at -20°C to -30°C is not a problem,**
- **But two-phase is more complex to implement.**

Microscale Flow and Heat Transfer Course:

FUNDAMENTALS OF MICROSCALE HEAT TRANSFER: BOILING, CONDENSATION, SINGLE- AND TWO-PHASE FLOWS

Date: June 7-11, 2010 at EPFL in Lausanne.

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***Lecturers*: Thome (EPFL), Michel (IBM Research), Celata (ENEA), Zun (Univ. of Ljubljana), Jacobi (Univ. of Illinois).**