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Session C3Or2B: Thermal-Fluid Transport and Properties for Fluids Boiling Above 50 K

“Simulation of a cryogenic capillary tube"

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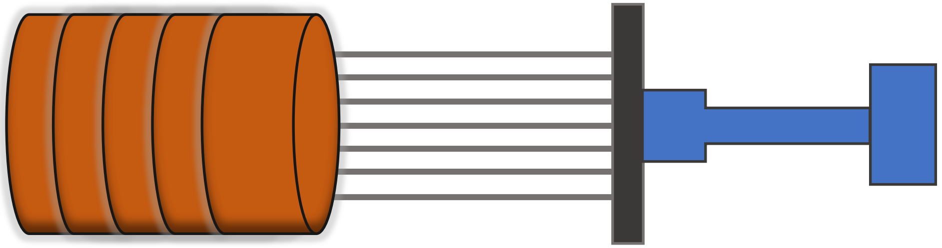
OVERVIEW

- Introduction
- Cryogenic Pulsating Heat Pipes
- Experimental results
- Reference case
- Description of the numerical model
- Numerical results
- Conclusions and future steps

INTRODUCTION

Develoment of **new cryogenic cooling technologies** for cooling **superconducting magnets**:
Cryogenic Pulsating Heat Pipes as a cooling solution



Magnet PHP thermal link Cryocooler

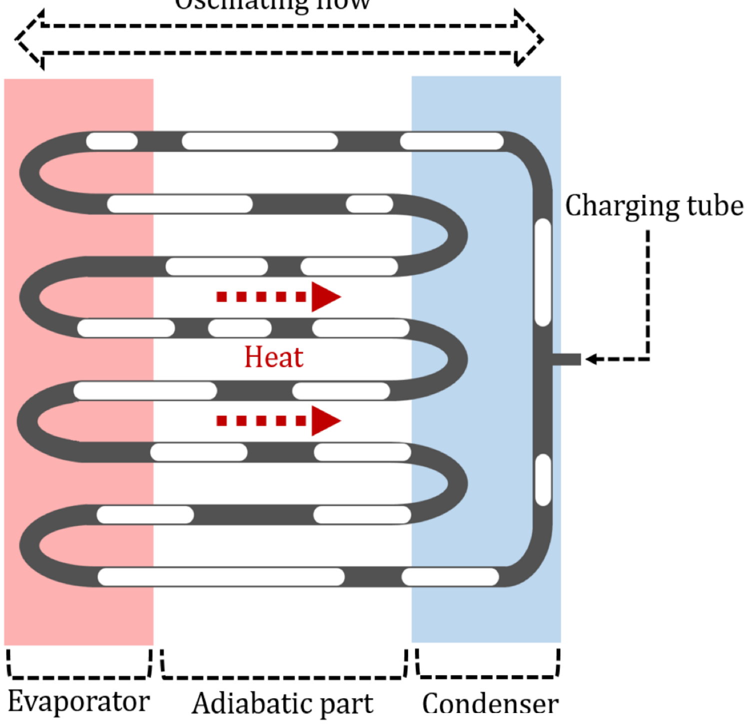
Goal:

To develop a **2D numerical model** based on a **simple geometry** able to represent the **fluid's behavior** in a **capillary tube**

Contribution to future 2D horizontal pulsating heat pipes simulations

CRYOGENIC PULSATING HEAT PIPES

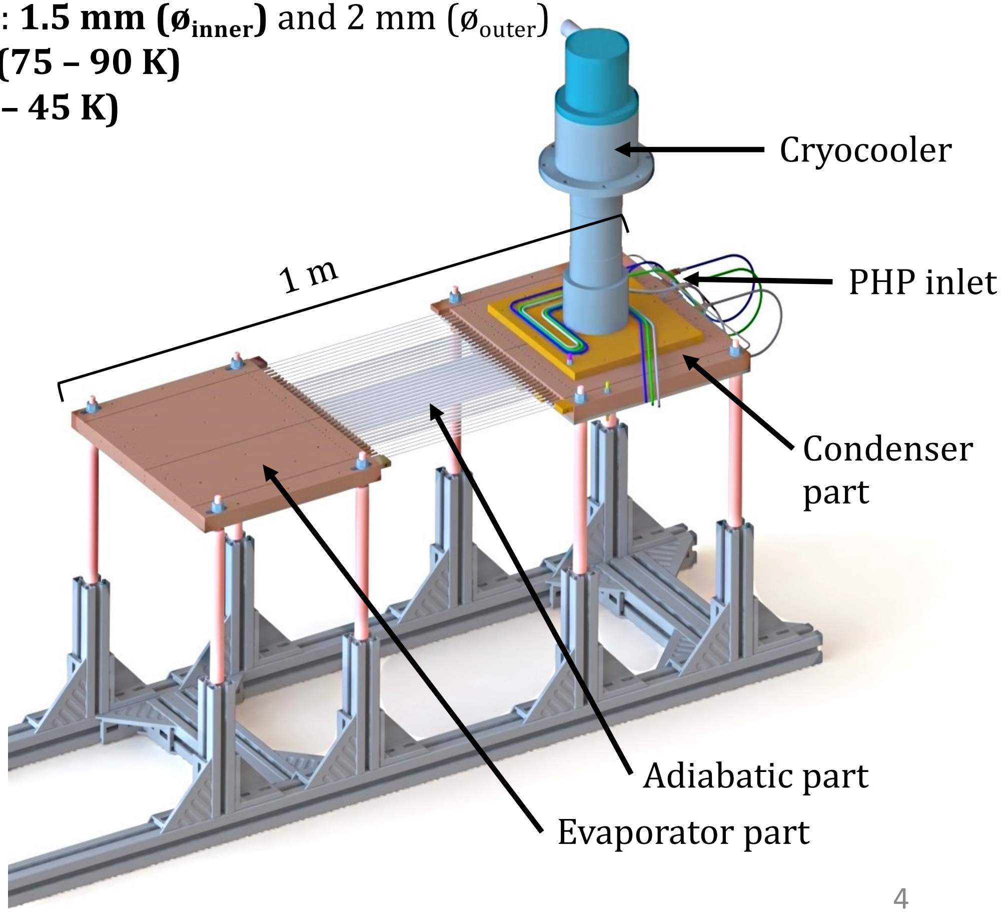
- **P**ulsating **H**eat **P**ipes (PHP): **two-phase thermal links**
- Working fluid close to phase-change conditions
- **Oscillating flow** of **liquid slugs** and **vapor plugs** (surrounded by a liquid film)



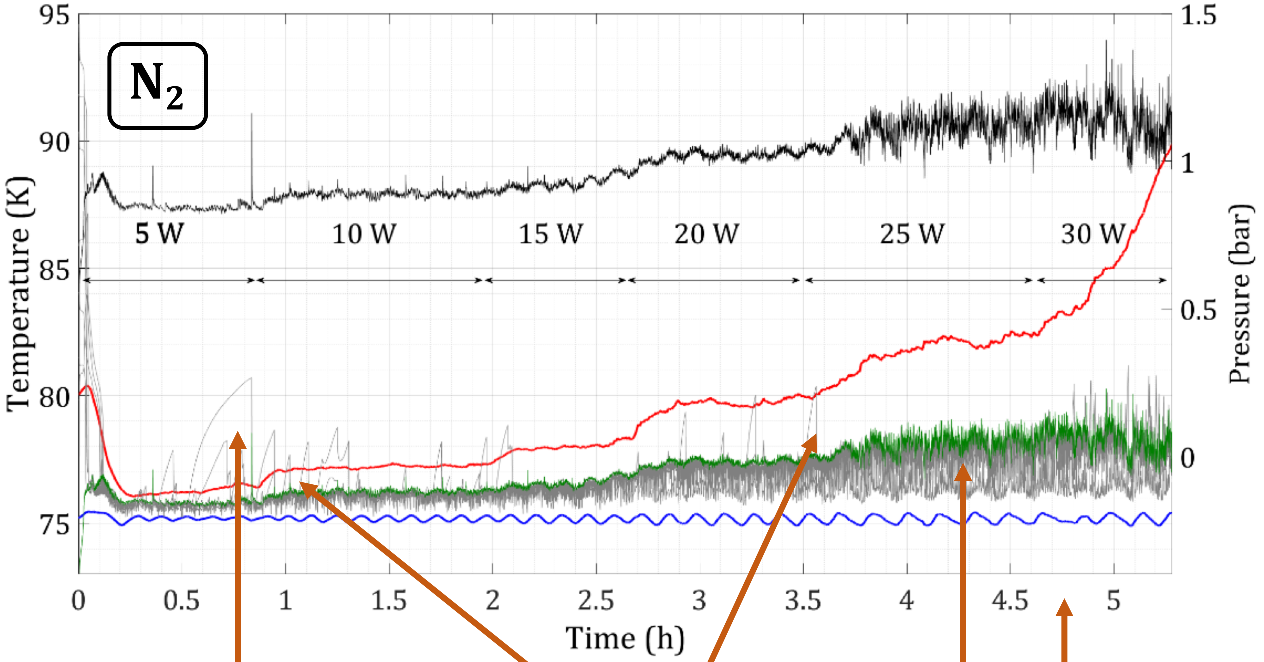
Maximum inner tube diameter (from the Bond number): $Bo = \frac{(\rho_l - \rho_v) g D^2}{\sigma} \leq 4 \qquad D_{crit} \leq 2\sqrt{\frac{\sigma}{g(\rho_l - \rho_v)}}$

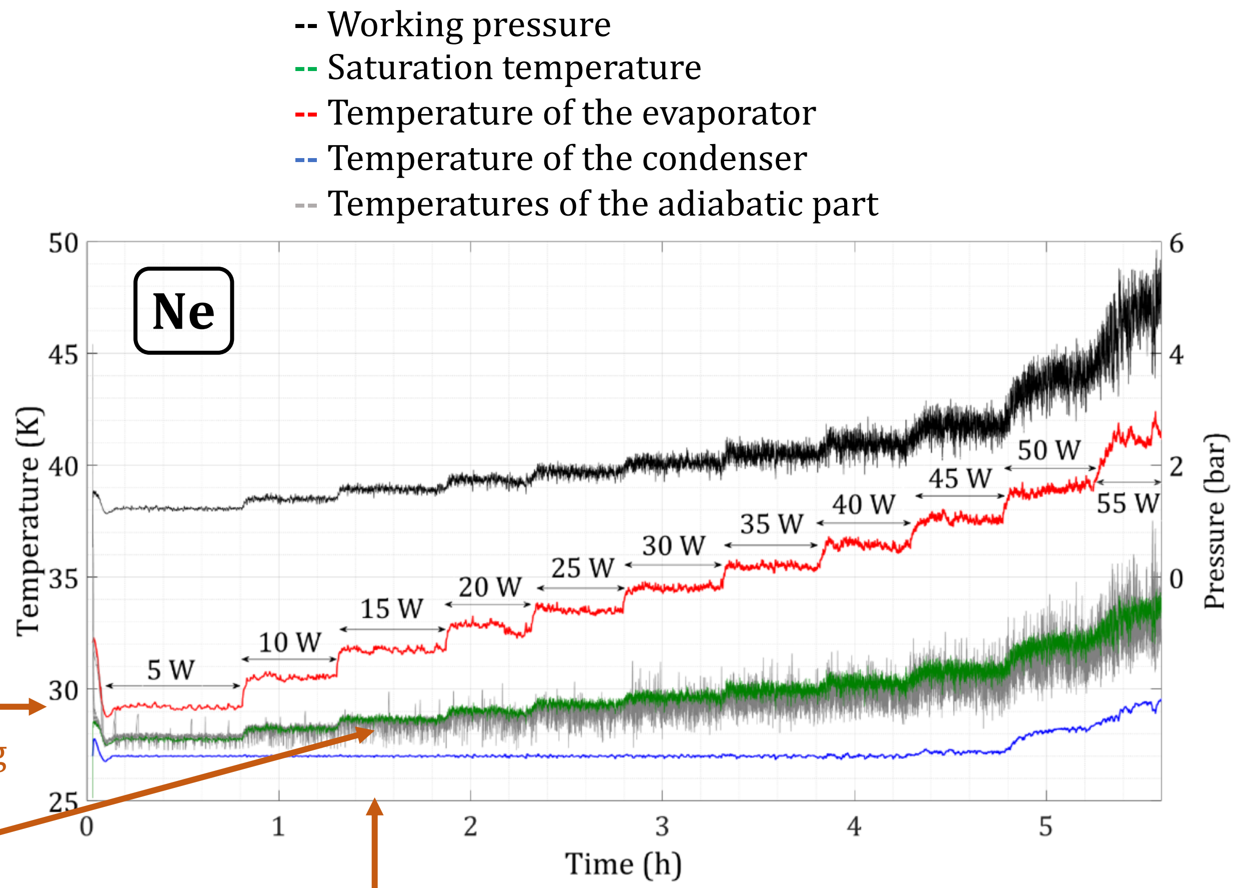
Cryogenic experimental facility:

- **Horizontal** position (closest configuration to zero-gravity)
- **1 m** long closed-PHP with 36 capillary tubes
- Diameters: **1.5 mm ($\boldsymbol{\o_{inner}}$)** and 2 mm ($\o_{outer}$)
- **Nitrogen (75 – 90 K)**
- **Neon (27 – 45 K)**



EXPERIMENTAL RESULTS





Working fluid exceeding evaporator's temperature (adiabatic compressions)

Local dry-outs at low and high heat loads (superheated vapor)

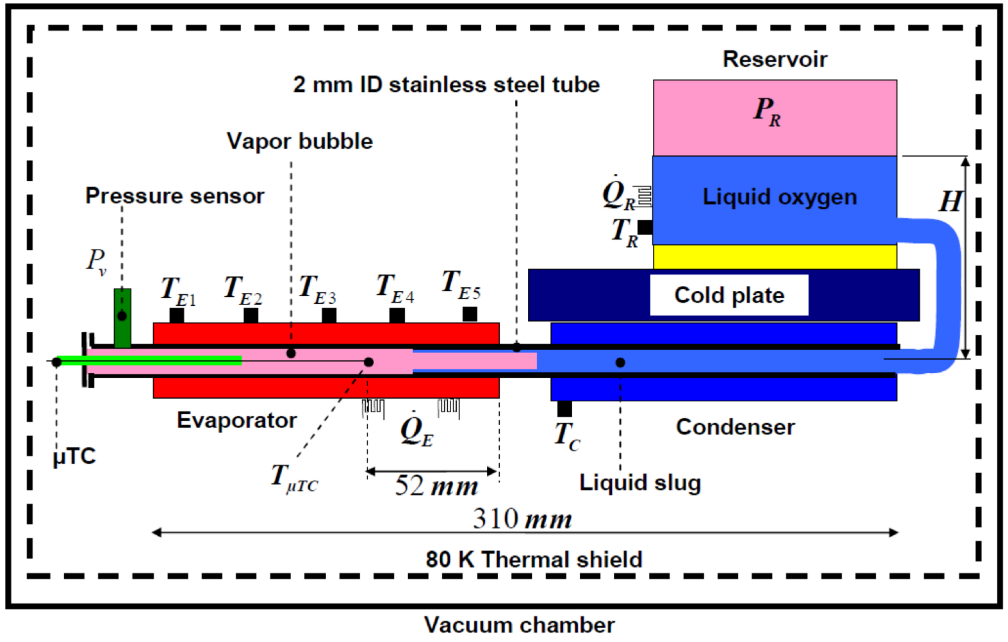
Stable oscillating phases

Subcooled state of the liquid parts

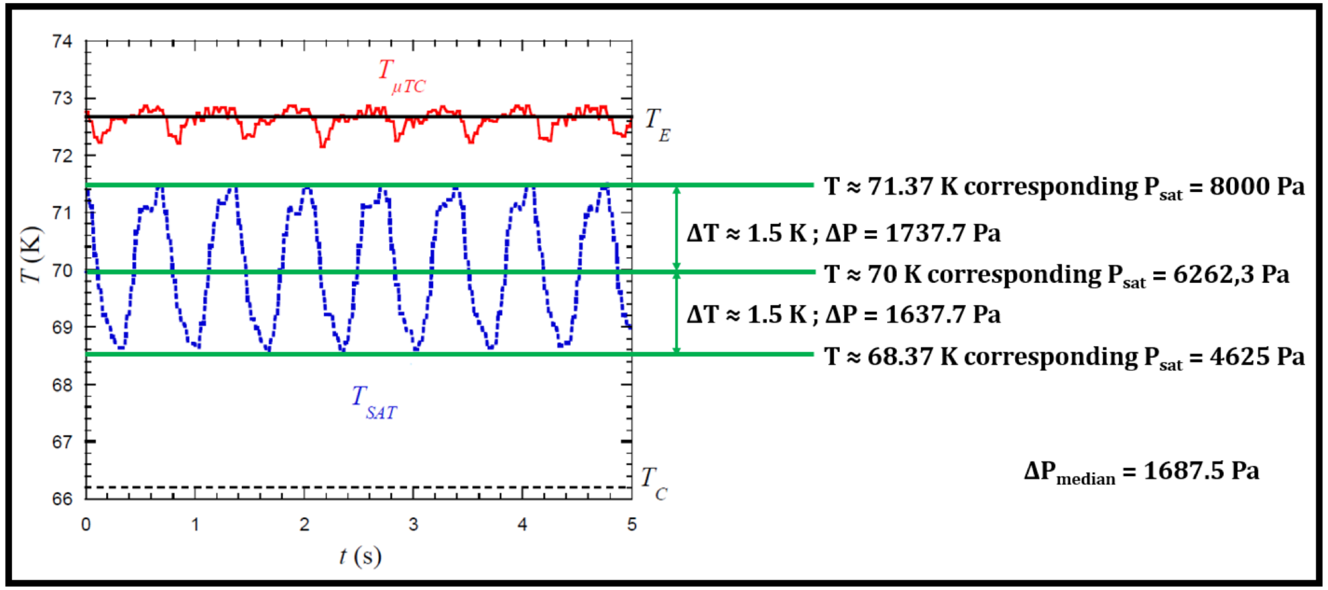
Inner diameter: 1.5 mm
Max theoretical inner diameter: ~ 1.25 mm
(larger than possible according to the most common criterion found in the literature, but impressive heat transfer capacity)

REFERENCE CASE*

2D numerical model with a simple geometry (cryogenic fluid, capillary dimensions, horizontal position)



Experimental results:



Characteristics:

- Evaporator and condenser copper plates
- 2 mm ($ø_{inner}$)
- Oxygen as working fluid (66 – 74 K)

*Gully P, Bonnet F, Nikoyalev V, Luchier N, Tran TQ
Evaluation of the thermodynamic state in PHP
17th International Heat Pipe Conference (2013)

DESCRIPTION OF THE NUMERICAL MODEL

Transient simulations performed with a **pressure-based ANSYS Fluent** solver using the **Volume of Fluid (VOF)** method in a **2D axisymmetric geometry.** The flow is assumed laminar and the gravity force is not considered.

VOF method: fluid-fluid interface modelling technique that allows to follow the evolution of the shape and position of the interface (conservative method):

- Volume fraction α_q of a given phase q, where $0 \leq \alpha_q \leq 1$ in each control volume
- Single momentum equation

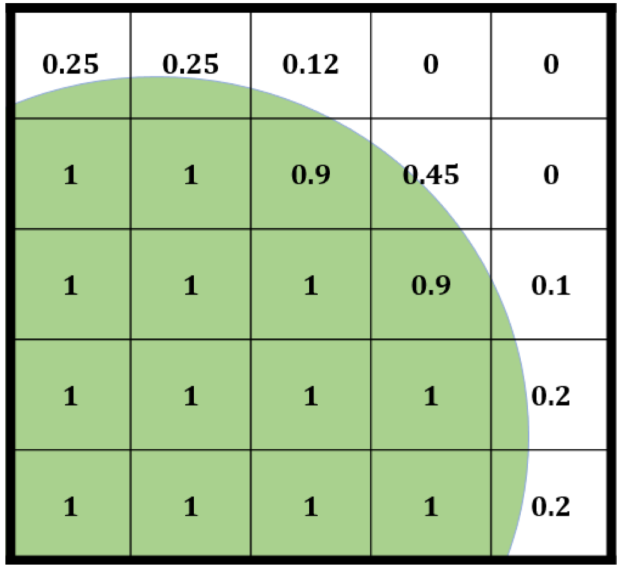
$$\frac{\partial(\rho\vec{v})}{\partial t} + \nabla\cdot(\rho\vec{v}\vec{v}) = -\nabla p + \nabla\cdot[\mu(\nabla\vec{v} + \nabla\vec{v}^T)] + \rho\vec{g} + \vec{F}$$

- Continuity and energy equations
- Averaged properties

$$p = \sum_{q=1}^{n} \alpha_q p_q$$ density for example, $\rho = \alpha_2\rho_2 + (1-\alpha_2)\rho_1$

- Energy and temperature (mass average variables)

$$E = \frac{\sum_{q=1}^{n} \alpha_q\rho_q E_q}{\sum_{q=1}^{n} \alpha_q\rho_q} \qquad T = \frac{\sum_{q=1}^{n} \alpha_q\rho_q T_q}{\sum_{q=1}^{n} \alpha_q\rho_q}$$



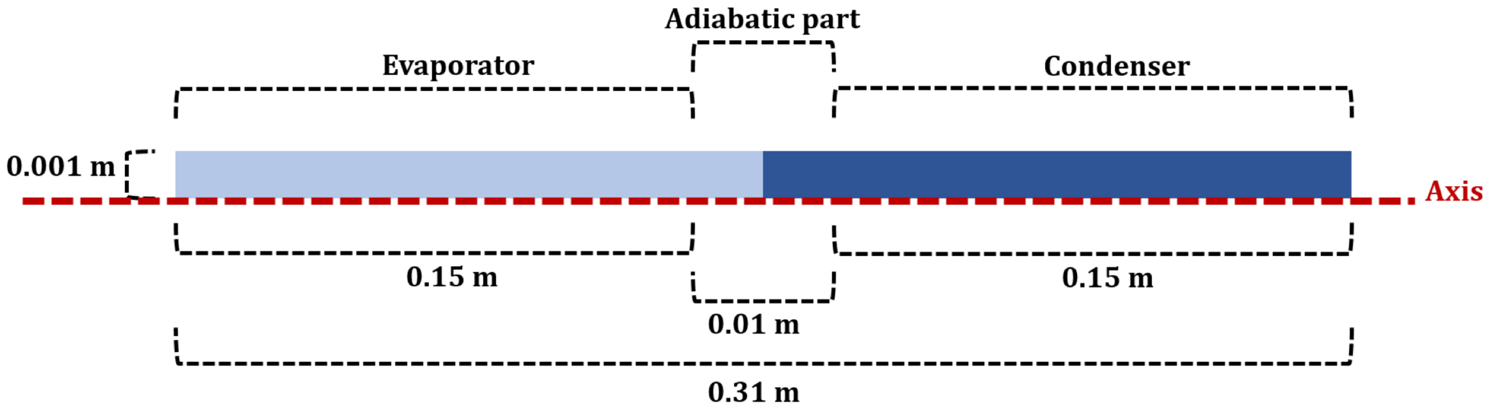
Solution methods: SIMPLE algorithm with a PRESTO! Loop (developed by ANSYS Fluent)
Geometry and mesh: generated with ANSYS ICEM CFD software and then imported into ANSYS Fluent

DESCRIPTION OF THE NUMERICAL MODEL

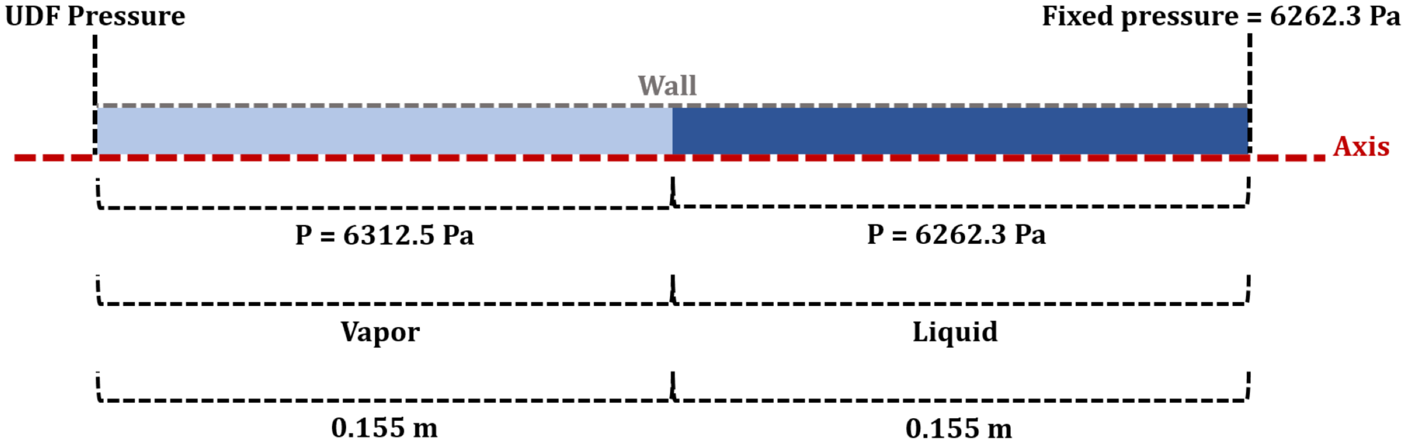
Dynamic model:

- Mesh: 117 743 square cells (25 in the y-axis)
- Time step: 10^{-5} s and, time calculation: 0.5 or 1 s

- Dimensions:



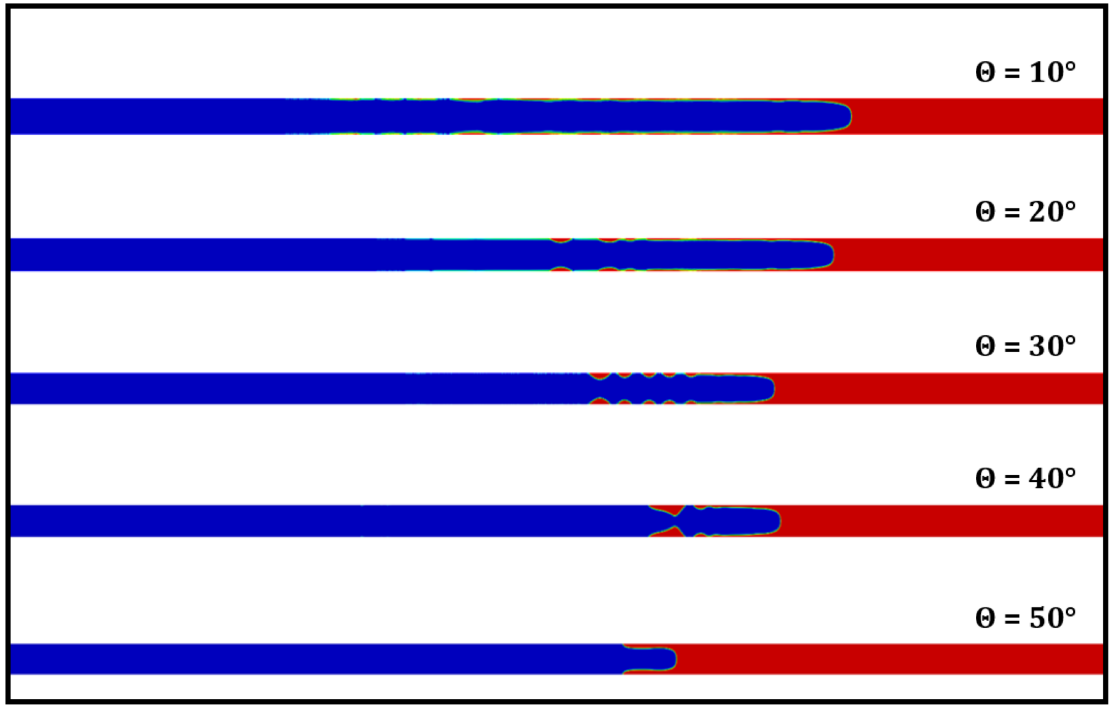
- Boundary and initial conditions:

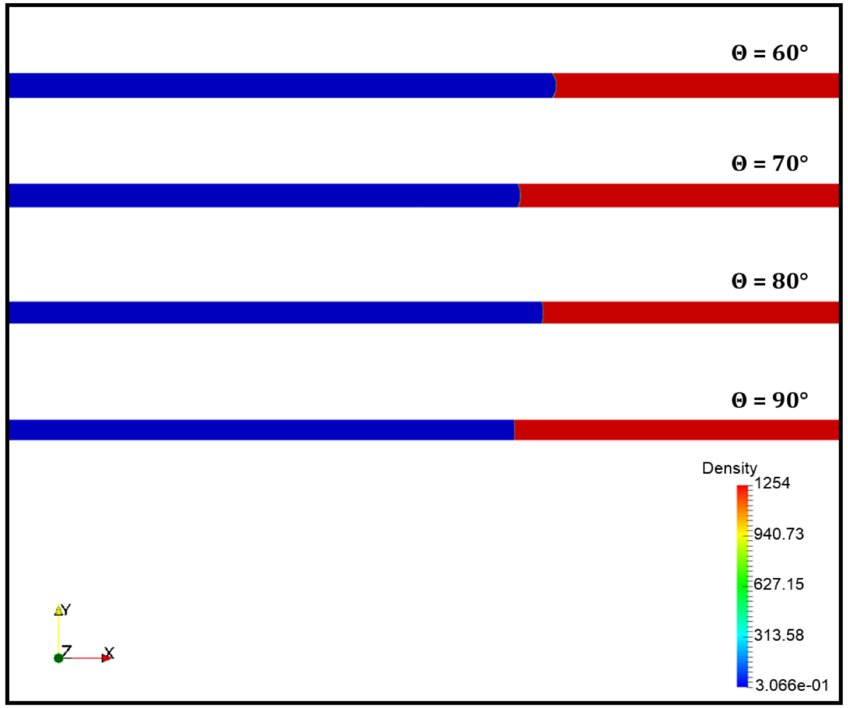


NUMERICAL RESULTS

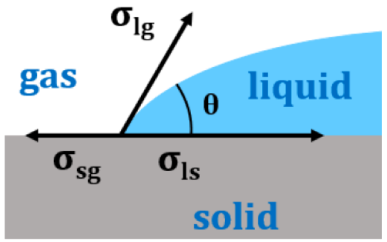
Remark: High influence of the contact angle in the shape of the liquid film and the formation of liquid droplets left behind the advancing liquid film. Several contact angles tested (from 10º to 90º)

Liquid film thicknesses analyzed and measured to choose a reasonable contact angle:
at **30º** average liquid film thickness around **80 µm**
(same order of magnitude than literature for the same inner diameter*)





Equilibrium between surface tension forces in the triple line



$$\sigma_{sl} + \sigma_{lg}\cos(\theta) = \sigma_{sg}$$

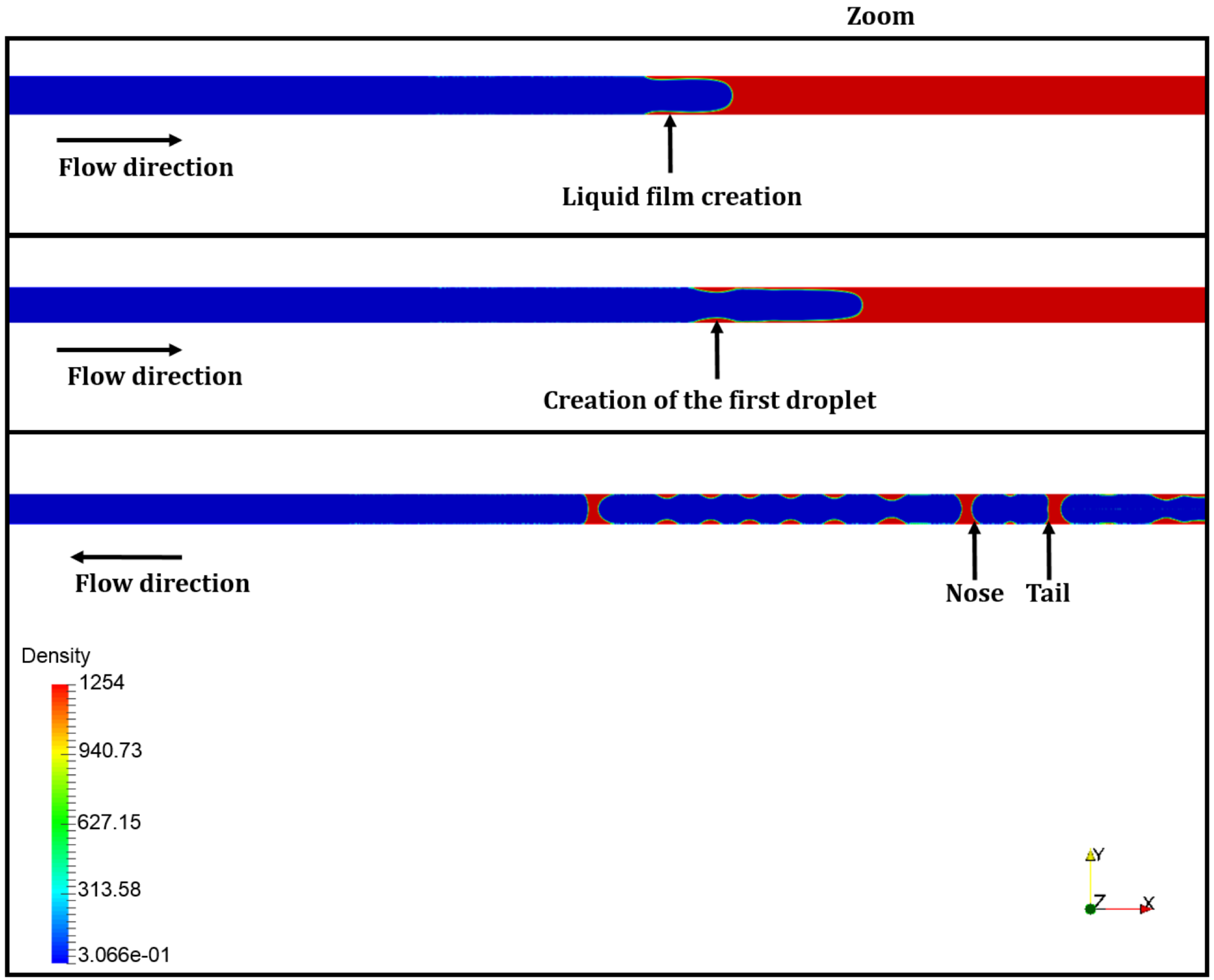
*Nekrashevych I, Nikoyalev V (2017) *Effect of tube heat conduction on the pulsating heat pipe start-up* Applied Thermal Engineering (Vol. 117, Pages 24-29)

NUMERICAL RESULTS

Interface visualization with ParaView: different instants of the creation of the liquid film at the beginning of the oscillations

- Liquid film droplets created and left behind the liquid film (Laplace Instabilities)
- Vapor plug adopts a different shape depending on the flow direction (nose and tail) in capillary tubes
  (also found in other visualization works**)

Optimization: including more information about the triple line evolution (dynamic contact angle)



***Tong BY, Wong TN, Ooi KT (2001)**
Closed-loop pulsating heat pipe
Applied Thermal Engineering (Vol. 21, Pages 1845-1862)

***Khandekar S (2004)**
Thermo-hydrodynamics of closed-loop pulsating heat pipes
PhD thesis, Institut für Kernenergetik und Energiesysteme der Universität Stuttgart

DESCRIPTION OF THE NUMERICAL MODEL

Thermal model: **Lee model** (evaporation and condensation processes)

Processes governed by the transport equation:

$$\frac{\partial \alpha_v \rho_v}{\partial t} + \nabla \cdot (\alpha_v \rho_v \vec{V_v}) = \dot{m}_{lv} - \dot{m}_{vl}$$

Evaporation process ($T_l > T_{sat}$):

$$\dot{m}_{lv} = Coef_l \alpha_l \rho_l \frac{T_l - T_{sat}}{T_{sat}}$$

Condensation process ($T_{sat} > T_v$):

$$\dot{m}_{vl} = Coef_v \alpha_v \rho_v \frac{T_{sat} - T_v}{T_{sat}}$$

Assumptions (used by other authors**):

- During evaporation/condensation process: $\mathbf{T_{cell} = T_{sat}}$ **and constant** (acceptable due to small size of cells)
  Temperature variation between two cells close to each other: **not significant**
- "excess of energy" of the cell making possible to **exceed the saturation temperature** used in the **evaporation/condensation process**

Mass transfer general formulation:

$$\dot{m} = \frac{(T - T_{sat})\rho\alpha Cp}{L_v}$$

Evaporation/condensation coefficients:

$$Coef_l = \frac{Cp_l T_{sat}}{L_v} \qquad Coef_v = \frac{Cp_v T_{sat}}{L_v}$$

*Lee H.W. (1979)
A pressure iteration scheme for two-phase flow modelling
Energy Division. Los Alamos Scientific Laboratory.

*Bruce R, Pascali S, Vendramini C, Baudouy B (2015)
Implementation of the thermodynamic and phase transition equations of superfluid helium in CFD software
IOP Conference Series – Materials Science and Engineering (Vol. 101)

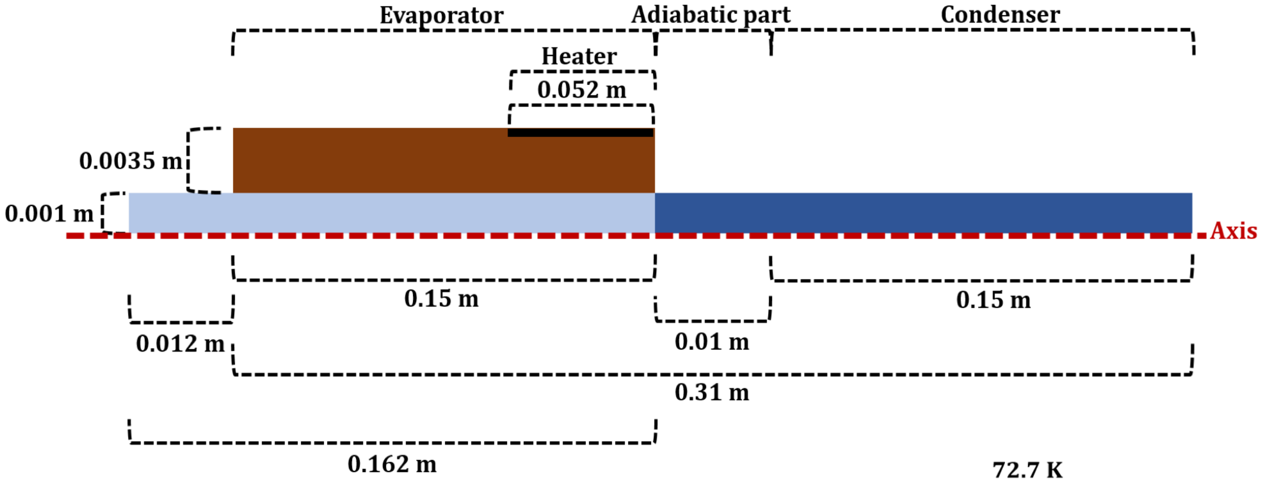
*Wrobel KM (2006)
Simulation of Flows with Evaporation and Condensation
PhD thesis, University of Twente

DESCRIPTION OF THE NUMERICAL MODEL

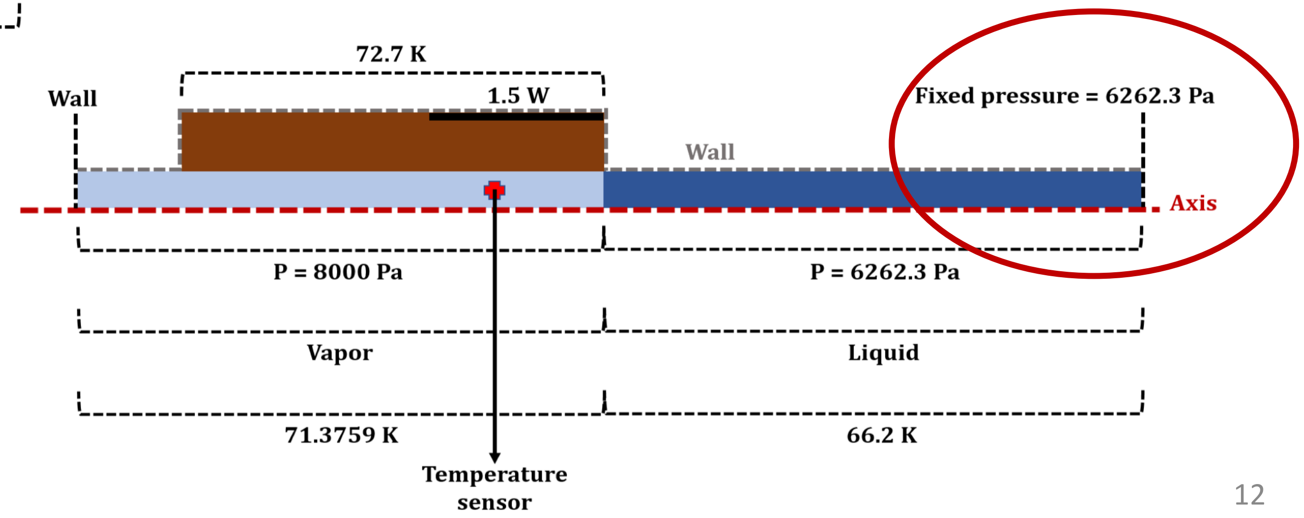
First thermal simulation:

- Mesh: 145 080 square cells
- Time step: 10^{-5} s and, time calculation: 2.5 s

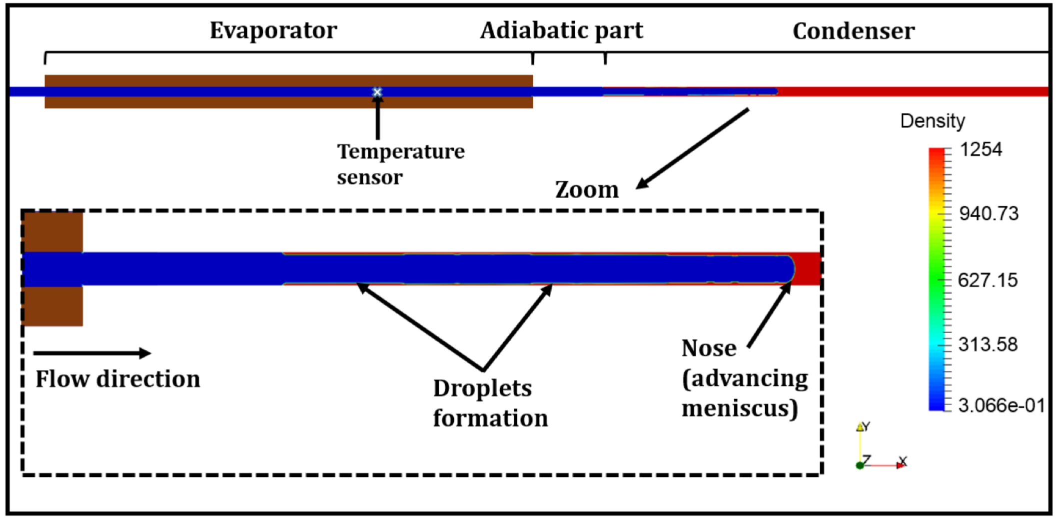
- Dimensions:

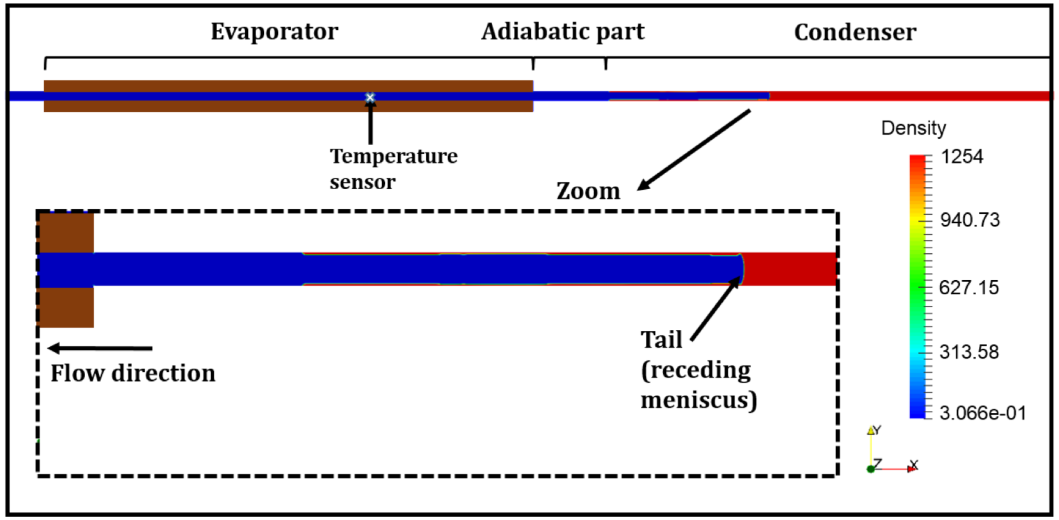


- Boundary and initial conditions:



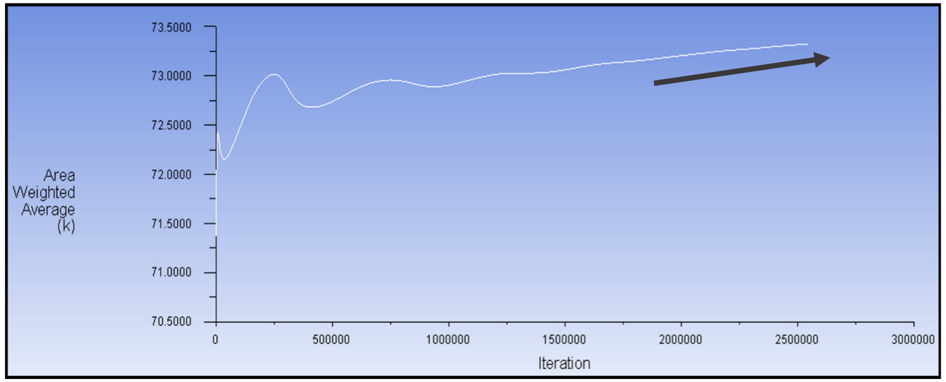
NUMERICAL RESULTS





- Droplets formation due to Laplace instabilities less pronounced (liquid film condensation)

- Shape of the advancing and receding meniscus also depends on the flow direction



- After a few oscillations due to the initial thermal instabilities, the two-phase interface stops to oscillate

- Liquid part do not enter into the evaporator (poor heat transfer) and temperature of the evaporator increases

Conclusion: Necessity of additional forces or instabilities to maintain an oscillations in horizontal capillary tubes (as in the reference case with the reservoir)

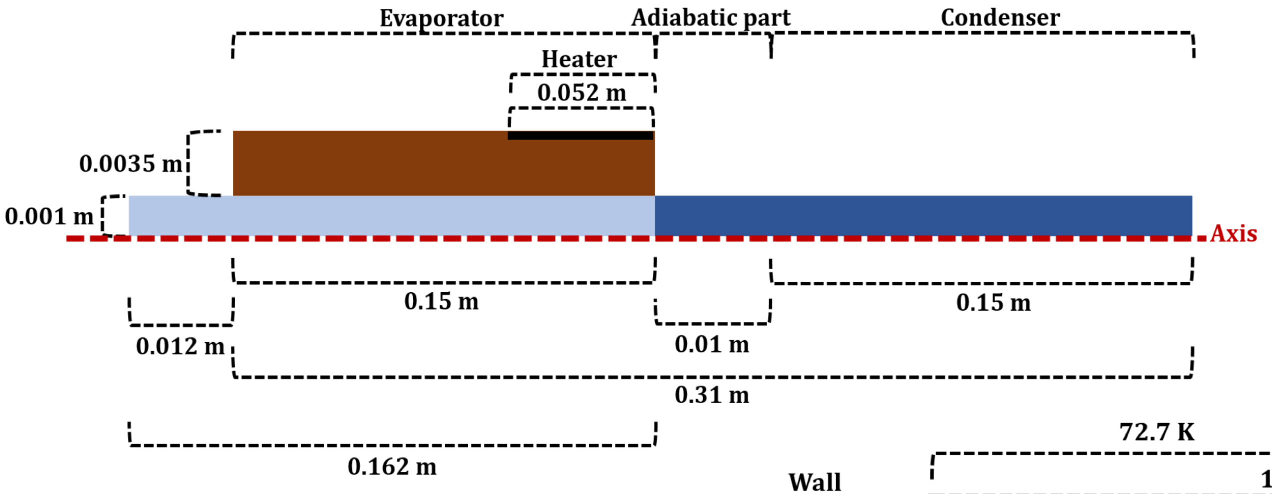
In pulsating heat pipes (pressure drop in U-turns, alternating heating/cooling zones, etc)

DESCRIPTION OF THE NUMERICAL MODEL

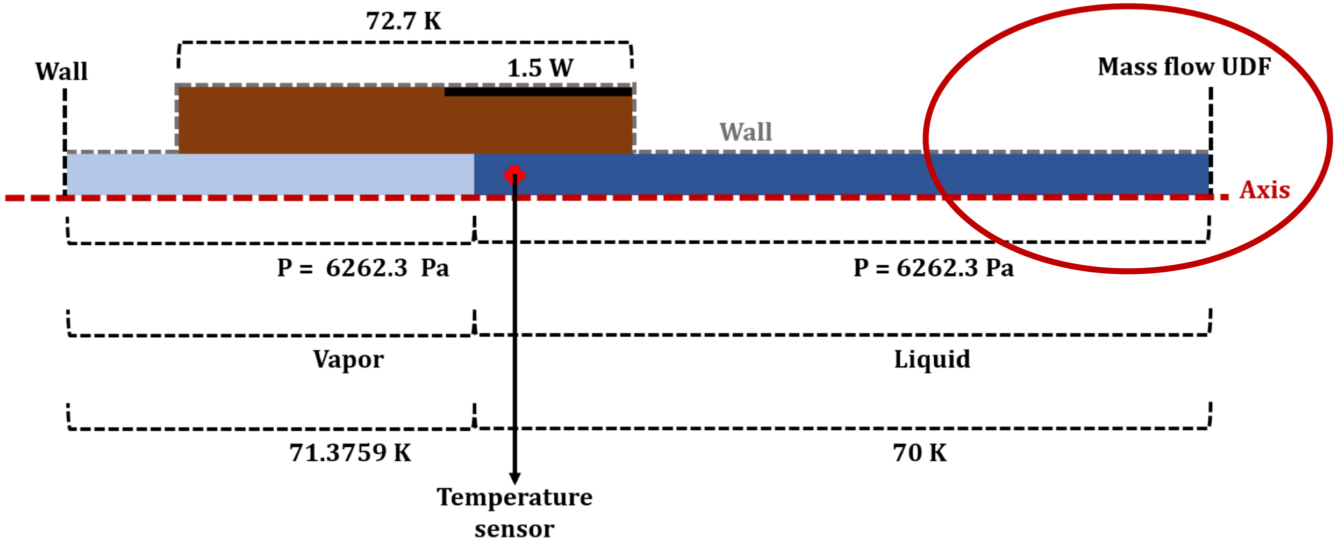
Second thermal simulation:

- Mesh: 145 080 square cells
- Time step: 10^{-5} s and, time calculation: 2.5 s (not finished)

- Dimensions:



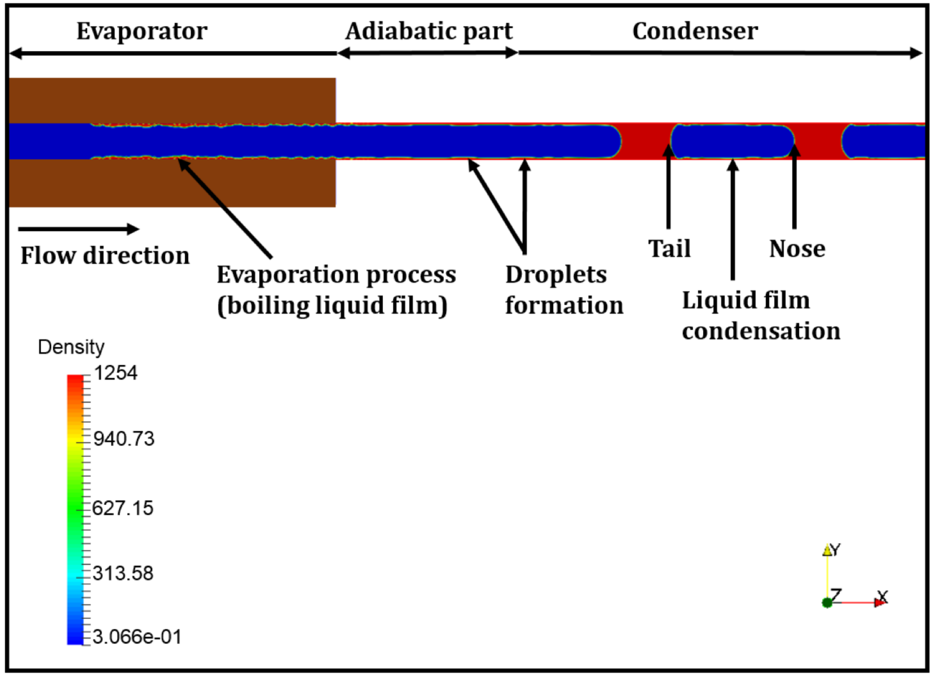
- Boundary and initial conditions:

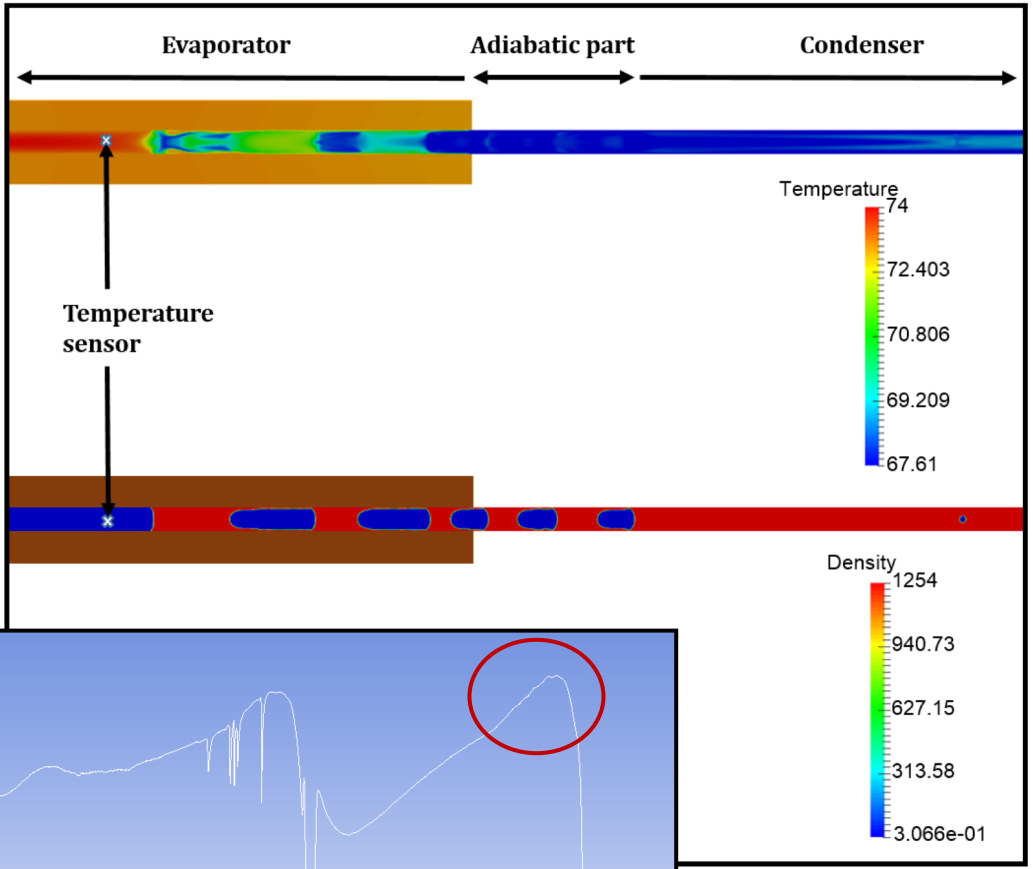


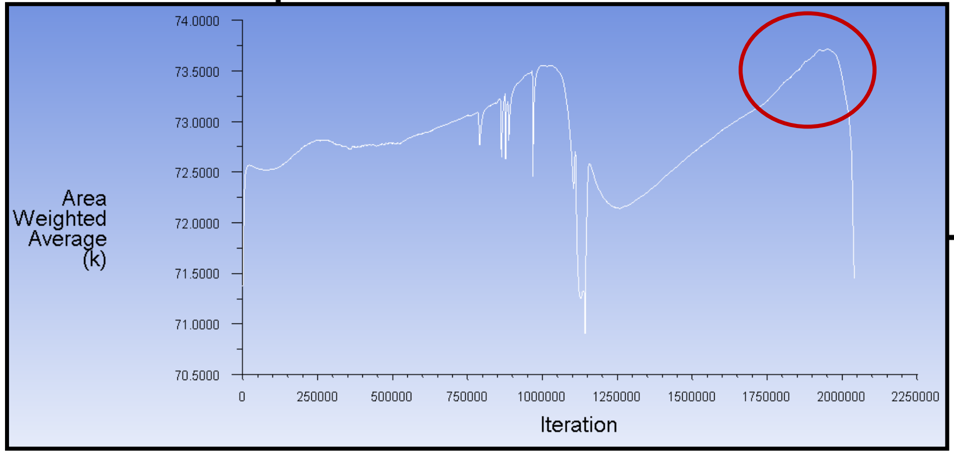
NUMERICAL RESULTS

Unfortunately, the calculation had to be stopped to include the last results in the this presentation, and the total time calculation is only 2.1 s (0.5 s of calculation takes more than one week).

- Evaporation and condensation processes of the liquid film observed in the evaporator and the condenser
- Temperature peaks registered by the temperature sensor when liquid parts re-enter in the evaporator
- Satisfactory heat transfer (evaporator stable at 73 K)
- Vapor part exceeding the evaporator's temperature (adiabatic compression)







CONCLUSIONS AND FUTURE STEPS

CONCLUSIONS:

From numerical results of the dynamic model:

- **influence of the contact angle** in the liquid film thickness and droplets formation due to Laplace instabilities
- vapor plug adopts a **different shape** depending on the **flow direction** in capillary tubes (also found in other visualization works)

Nevertheless, the model can still be optimized including more information about the **triple line evolution**

From numerical results of the thermal model:

- **necessity of additional instabilities** (force or pressure variation) that contributes to global flow oscillations (apart from thermodynamic instabilities)
- satisfactory **heat transfer** between the evaporator solid part and the fluid, and the condenser part and the fluid (calculation still running)

Even if this type of **CFD 2D axisymmetric simulation is still at its early stages**, it has been shown that it can be developed for future 2D pulsating heat pipes simulations

FUTURE STEPS

- Modification of the geometry (alternating heating and cooling sections, including a U-turn)

- Comparison with our experimental results (nitrogen and neon as working fluids)

- Study of variable inner diameter (specially with neon)

Thank you!